compressed air compendium

Contents

Chapter Page

Part 1 Fundamentals of compressed air

1.1 The history of compressed air 1
1.1.1 The origin of compressed air 1
1.1.2 The first applications of compressed air 2
1.2 Units and formula symbols 6
1.2.1 Basic units 6
1.2.2 Compressed air units 6
1.3 What is compressed air ? 7
1.3.1 The composition of air 7
1.3.2 The properties of compressed air 7
1.3.3 How does compressed air behave? 7
1.4 Physical fundamentals 8
1.4.1 Temperature 9
1.4.2 Volume 9
1.4.3 Pressure 10
1.4.4 Volume flow 11
1.5 Compressed air in motion 13
1.5.1 Flow behaviour 13
1.5.2 Types of flow 13

Part 2 Applications for pneumatics

2.1 The advantages of compressed air 14
2.2 Pressure ranges 17
2.3 Possible applications for compressed air 18
2.3.1 Tensioning and clamping with compressed air 18
2.3.2 Conveyance by compressed air 18
2.3.3 Pneumatic drive systems 19
2.3.4 Spraying with compressed air 19
2.3.5 Blowing and flushing with compressed air 19
2.3.6 Testing and inspection with compressed air 20
2.3.7 Using compressed air for process control 20
2.4 Examples of specialised applications 21

Part 3 Compressed air generators

3.1 Compressors (compactors) 24
3.1.1 Dynamic compressors (Turbo-compressors) 24
3.1.2 Displacement compressors 24
3.2 Types of compressor 25
3.2.1 Standard compressors 26
3.2.2 Piston (reciprocating) compressor 27
3.2.3 Diaphragm compressor 29
3.2.4 Free piston compressor 30
3.2.5 Rotary vane compressor 31
3.2.6 Liquid ring compressor 32
3.2.7 Screw compressor 33
3.2.8 Roots compressor 34
3.2.9 Axial compressor 35
3.2.10 Radial compressor 36

Chapter		Page

3.3 Piston compressors **37**
3.3.1 General 37
3.3.2 Suction capacity - output 38
3.3.3 Cooling 39
3.3.4 Coolant 40
3.3.5 Control of reciprocating piston compressors 40
3.3.6 Advantages of reciprocating piston compressors 40
3.3.7 Components of a piston compressor 41

3.4 Screw compressors **42**
3.4.1 General 42
3.4.2 Compression process 42
3.4.3 Method of operation 43
3.4.4 Oil circuit 44
3.4.5 Pneumatic circuit 45
3.4.6 Heat reclamation 46
3.4.7 Intake control 46
3.4.8 Advantages of screw compressors 46
3.4.9 Main components of a screw compressor 47

3.5 Components **48**
3.5.1 Drive motor 48
3.5.2 Drive belts 48
3.5.3 Belt tensioning 48
3.5.4 Inlet and pressure valves 49
3.5.5 Safety valve 49
3.5.6 Intake filter 49

3.6 Compressor lubricants and coolants **50**

Part 4
Control of compressors

4.1 Pressure definitions **51**

4.2 Operating status **52**
4.2.1 Stopped (L_0) 52
4.2.2 Idle (L_1) 52
4.2.3 Part-load 53
4.2.4 Operating load (L_2) 53

4.3 Controlling individual compressors **54**
4.3.1 Intermittent control 54
4.3.2 Idle mode control 54
4.3.3 Delayed intermittent control 55
4.3.4 Part-load control 56
4.3.4.1 Proportional regulation 56
4.3.4.2 Frequency control 57

4.4. The ARS control concept **59**
4.4.1 Autotronic 60
4.4.2 Ratiotronic 60
4.4.3 Supertronic 61

Chapter Page

4.5 Control of several compressors ... 62
4.5.1 MCS 1 and MCS 2 ... 62
4.5.2 MCS 3 ... 63
4.5.3 MCS 4 ... 64
4.5.4 MCS 5 ... 65
4.5.5 MCS 6 ... 66
4.5.6 MCS 7 ... 67

Part 5
Compressed air treatment

5.1 Why treatment ? ... 68
5.1.2 Planning information ... 69
5.1.3 Consequences of poor treatment ... 70
5.1.3 Impurities in the air ... 71
5.2 Water in the compressed air ... 72
5.2.1 Atmospheric humidity ... 72
5.2.2 Dew points ... 73
5.2.3 Air moisture content ... 73
5.2.4 Quantity of condensate during compression ... 74
5.2.5 Example for calculating quantities of condensate ... 75
5.2.6 Quantity of condensate on a humid Summer day ... 76
5.2.7 Determining the pressure dew point ... 77
5.2.8 Pressure dew point after removal of pressure ... 78
5.3 Compressed air quality ... 79
5.3.1 Quality classes defined in DIN ISO 8573-1 ... 79
5.4 Methods of drying ... 80
5.4.1 Operating conditions ... 81
5.4.2 Condensation by high pressure ... 82
5.4.3 Condensation by refrigeration drying ... 83
5.4.4 Diffusion by membrane drying ... 84
5.4.5 Sorption by Absorption ... 85
5.4.6 Sorption by Adsorption ... 86
5.4.6.1 Heatless regeneration ... 87
5.4.6.2 Internal heat regeneration ... 88
5.4.6.3 External heat regeneration ... 89
5.4.6.4 Vacuum regeneration ... 90
5.4.7 Arrangement of the refrigeration compressed air dryer ... 91
5.4.7.1 Dryer before the compressed air receiver ... 91
5.4.7.2 Dryer behind the compressed air receiver ... 92
5.5 Compressed air filters ... 93
5.5.1 Basic terminology of filters ... 93
5.5.1.1 Filter separation rate η [%] ... 93
5.5.1.2 Pressure drop Δp ... 94
5.5.1.3 Operating pressure ... 94
5.5.2 Dust separators ... 95
5.5.3 Pre-filters ... 96
5.5.4 Microfilters ... 96
5.5.5 Active carbon filters ... 99
5.5.6 Active carbon adsorbers ... 100
5.5.7 Sterile filters ... 101

	Chapter		Page
Part 6 Disposal of condensate	**6.1**	**Condensate**	**102**
	6.2	**Condensate drains**	**103**
	6.2.1	Condensate drains with manual valves	104
	6.2.2	Condensate drains with float control	104
	6.2.3	Condensate drains with timer operated solenoid valves	105
	6.2.4	Condensate drains with electronic level control	106
	6.2.5	Condensate drains with float operated level control	107
	6.3	**Condensate treatment**	**108**
	6.3.1	Oil-water separators	109
Part 7 Compressed air requirement	**7.1**	**Consumption of compressed air by pneumatic devices**	**110**
	7.1.1	Consumption of nozzles	110
	7.1.1.1	Compressed air consumption of cylindrical nozzles	111
	7.1.1.2	Compressed air consumption of paint spray guns	112
	7.1.1.3	Compressed air consumption of jet nozzles	113
	7.1.2	Compressed air consumption of cylinders	114
	7.1.3	Compressed air consumption of tools	115
	7.2	**Determining compressed air requirement**	**117**
	7.2.1	Average operation time	117
	7.2.2	Simultanity factor	118
	7.2.3	Defining compressed air requirement	119
	7.2.3.1	Automatic consumer devices	119
	7.2.3.2	General consumer devices	120
	7.2.3.3	Total compressed air consumption	120
	7.2.4	Allowances for losses and reserves	121
	7.2.5	FAD Required L_B	121
	7.3	**Compressed air loss**	**122**
	7.3.1	Costs of compressed air loss	122
	7.3.2	Quantifying leakage	123
	7.3.2.1	Quantifying leakage by emptying the receiver	123
	7.3.2.2	Quantifying leakage by measuring working time	124
	7.3.3	Limits for leakage	125
	7.3.4	Measures for minimising compressed air loss	125
	7.3.5	Reconstructing a pneumatic network	126
Part 8 Determining the size of the compressor station	**8.1**	**The type of compressor**	**127**
	8.1.1	Screw compressors	127
	8.1.2	Piston compressors	127
	8.2	**Maximum pressure P_{max}**	**128**
	8.2.1	Factors influencing cutout pressure P_{max}	128
	8.3	**Determining the volume of a compressed air receiver**	**129**
	8.3.1	Recommendations for the volume of compressed air receivers	129
	8.3.2	Norm series and operating pressures for sizes of compressed air receivers	129
	8.3.3	Volumes of compressed air receivers for compressors	130

Chapter | Page

8.4 Compressor cycle intervals 131
8.4.1 Compressor idle times 131
8.4.2 Compressor running times 131
8.4.3 Determining the motor cycle speed 132
8.5 Examples for compressor configuration 133
8.5.1 Samples calculation for piston compressors 133
8.5.1.1 Determining the maximum pressure P_{max} 133
8.5.1.2 Determining compressor size 134
8.5.1.3 Volume of the compressed air receiver 134
8.5.1.4 Compressor cycle interval 135
8.5.1.5 Motor cycling rate of compressor 136
8.5.2 Samples calculation for screw compressors 137
8.5.2.1 Example for determining the maximum pressure P_{max} 137
8.5.2.2 Determining compressor size 137
8.5.2.3 Dimensioning the compressed air receiver 138
8.5.2.4 Compressor cycle interval 138
8.5.3 Summary on compressor selection 139
8.6 Information on compressor configuration 140
8.6.1 Performance and working pressure 140
8.6.2 Varying working pressure of consumer devices 141
8.6.3 Combined compressor systems 141

Part 9
The pneumatic system

9.1 The compressed air receiver 142
9.1.1 Storing compressed air 142
9.1.2 Pulsation damping 142
9.1.3 Condensate collection 143
9.1.4 Operation of compressed air receivers 143
9.1.5 Installation of compressed air receivers 143
9.1.6 Safety rules for compressed air receivers 144
9.1.6.1 Registration and inspection obligations 144
9.1.6.2 Approved inspection authorities and authorized personnel 144
9.1.6.3 Inspection prior to commissioning 145
9.1.6.4 Registration 145
9.1.6.5 Repetitive inspections 145
9.1.7 Fittings on the compressed air receivers 147
9.1.7.1 Safety valve 148
9.2 The compressed air circuit 149
9.2.1 The structure of a compressed air circuit 149
9.2.1.1 The main line 149
9.2.1.2 The distribution line- ring line 150
9.2.1.3 The distribution line- stub line 151
9.2.1.4 The connection line 151
9.2.1.5 Connecting to a collective line with multiple systems 152

Chapter **Page**

9.3 Tips for planning pipe systems 153
9.3.1 General planning tips 153
9.3.2 Pipeline without compressed air dryer 154
9.3.3 Pipeline system with compressed air dryer 155

9.4 Pressure loss Δp 156
9.4.1 Type of flow 156
9.4.2 The Reynolds number Re 156
9.4.3 Pressure loss in the pipe system 157

9.5 Dimensioning pipelines 158
9.5.1 Maximum pressure drop Δp 158
9.5.2 Nominal width of pipelines Comparison [DN – Inch] 159
9.5.3 Equivalent pipe length 160
9.5.4 Determining the inside diameter d_i of the pipe by calculation 161
9.5.5 Determining the inside diameter of the pipe d_i by graphics 162
9.5.6 Determining the inside diameter of the pipe d_i with the aid of a bar graph 163

9.6 Choosing the material for pipelines 164
9.6.1 Threaded pipes 164
9.6.2 Seamless steel pipes 165
9.6.3 Stainless steel pipes 165
9.6.4 Copper pipes 166
9.6.5 Plastic pipes 167

9.7 Marking pipelines 168

Part 10
The Installation Room

10.1 Cooling the compressor 169

10.2 Compressor installation 170
10.2.1 General information regarding the installation room 170
10.2.2 Admissible ambient temperature 170
10.2.3 Fire safety rules for installation rooms 171
10.2.4 Disposal of condensate 171
10.2.5 Compressor installation instructions 172
10.2.6 The space requirement of a compressor 172
10.2.7 Conditions for installing compressed air receivers 173

10.3 Ventilation of a compressor station 174
10.3.1 Factors influencing the flow of cooling air of a $\dot{V}_c$ of a compressor 174
10.3.2 Definition of the factors influencing the flow of cooling air $\dot{V}_c$ to and from a compressor 175
10.3.3 General information for ventilation of compressor rooms 176
10.3.4 Natural ventilation 177
10.3.4.1 Outlet air aperture required for natural ventilation 177
10.3.5 Artificial ventilation 178
10.3.5.1 Required ventilator output with artificial ventilation 178
10.3.5.2 Required inlet air aperture with artificial ventilation 179

Chapter Page

10.3.5.3 Example of artificial ventilation of a compressor station ... 180
10.3.6 Circulation of cooling-air with inlet and outlet ducts 181
10.3.6.1 Air inlet ducts 181
10.3.6.2 Extraction of air through a cool-air duct 182
10.3.6.3 Required flow of cooling-air $\dot{\mathbf{V}}_d$ and cross-section of duct $\mathbf{A}_d$ when using a cool-air duc 182
10.3.6.4 Information concerning ventilation by ducting 183
10.3.6.5 Dimensioning the air inlet aperture when using an outlet duct 184
10.3.6.6 Variations of duct-type ventilation 185
10.4 Example installation plans 186
10.4.1 Installation of a screw-type compressor: an example 186
10.4.2 Installation of piston-type compressor: an example 187

Part 11
Heat recovery

11.1 The heat balance of a compressor station 188
11.2 Room heating 189
11.2.1 Room heating through ducting 189
11.2.2 Operation of room heating 190
11.2.3 Economy of room heating 190
11.3 The Duotherm heat exchanger 191
11.3.1 Duotherm BPT 191
11.3.2 Duotherm BSW 192
11.3.3 How much energy is it possible to save? 193
11.4 Closing remarks concerning heat recovery 194

Part 12
Sound

12.1 The nature of sound 195
12.1.1 Sound perception 195
12.2 Important terminology in acoustics 196
12.2.1 Sound pressure 196
12.2.2 Sound level 196
12.2.3 Sound intensity 196
12.3 Human perception of sound 197
12.3.1 The sound intensity level 197
12.3.2 Assessed sound level dB (A) 197
12.3.3 Loudness in comparison 198
12.4 Behaviour of sound 199
12.4.1 Distance from the sound source 199
12.4.2 Reflection and Absorption 199
12.4.3 Damping sound 200
12.4.5 Dessemination of sound in pipes and ducts 200
12.4.6 Sound pressure level from many sound sources 201
12.4.6.1 Several sound sources with the same level 201
12.4.6.2 Two sound sources with different levels 201
12.5 The effects of noise 202
12.6 Noise measurement 203
12.7 Silencing on compressors 203

	Chapter	Page
Part 13 Costs of compressed air	**13.1 Composition of compressed air costs**	**204**
	13.1.1 Cost factor ratios	204
	13.2 Cost-effectiveness calculation for energy costs	**205**
Part 14 Appendix	**A.1 Symbols**	**206**
	A.1.1 Picture symbols defined by DIN 28004	206
	A.1.2 Symbols for contact units and switching devices as per ISO 1219	208
Part 15 Index	**15.1 Conversion Table**	**212**

1. Fundamentals of compressed air

1.1 The history of compressed air

Compressed air, together with electricity, is the most frequently used carrier of energy in industry and the crafts today. But whereas we learn to use electricity and electrical appliances from a very early age, the possibilities, advantages and essentials of compressed air are far less understood.

People's comprehension of compressed air grew parallel to their understanding in other technical fields. Its development was only furthered where it was seen to have advantages over other technologies. But compressed air was always being used, and so clever people were always thinking about how to put it to better use.

1.1.1 The origin of compressed air

The first compressor - the lung

Many technical applications originate from the earliest days of mankind. The first use of compressed air was blowing on tinder to fan a flame. The air used for blowing was compressed in the lungs. Indeed, the lung could be called a kind of **natural compressor**. The capacity and performance of this compressor is extremely impressive. The human lung can process 100 l/min or 6 m^3 of air per hour. In doing so it generates a pressure of 0,02 - 0,08 bar. In a healthy condition, the reliability of the human compressor is unsurpassed and it costs nothing to service.

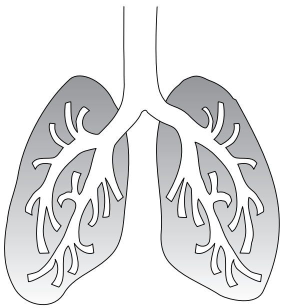


Fig. 1.1:
The first compressor - the lung

The further development of the „lung“

However, the lung proved to be wholly inadequate when people began to smelt pure metals such as gold, copper, tin and lead more than 5000 years ago. And when they started to make high grade metals, such as iron from ore, further development of compressed air technology was essential. More powerful aids than the lung were needed to generate temperatures of over 1000° C. At first they used the high winds on uplands and the crests of hills. Later, Egyptian and Sumerian goldsmiths made use of the blast pipe. This brought air directly into the embers, which increased the temperature decisively. Even today, goldsmiths all over the world use a similar device. However, this is only useful for melting small quantities of metal.

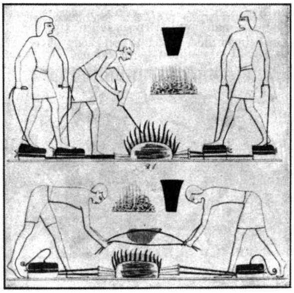


Fig. 1.2:
Picture of the foot-powered bellows in ancient Egypt

The first mechanical compressor - the bellows

The first mechanical compressor, the hand-powered bellows, was developed in the middle of the third millennium BC. The much more powerful foot-powered bellows was invented around 1500 BC. This progress was necessary when the alloying of copper and tin to make bronze developed into a stable manufacturing process. The development can be seen in a wall-painting of an ancient Egyptian grave. It was the birth of compressed air as we know it today.

1.1.2 The first applications of compressed air

Recognising the properties of compressed air

Hydraulic organ
Storage and suppression of pulsation

The first deliberate exploitation of energy in the air is handed down to us by the Greek Ktesibios (ca. 285 to 222 BC). He built a hydraulic organ and used compressed air for the **storage** and **reduction of vibration**.

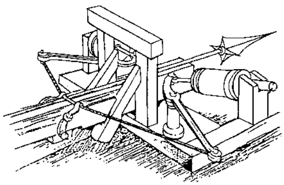


Fig. 1.3:
The catapult of Ktesibios

Catapult
Storage of energy

Ktesibios used another property of compressed air, **stored energy**, for his catapult. With the aid of air compressed in a cylinder, the Greek's catapult generated enough tension to propel missiles.

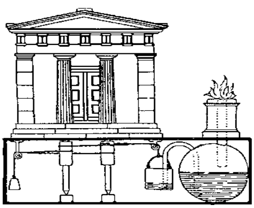


Fig.1.4:
The temple doors of Heron

Temple doors
Expansion and the performance of work

Heron, an engineer living in Alexandria in the first century BC, found a way to open the doors of a temple automatically by keeping the flame at the altar inside the building permanently alight. The secret was to use the **expansion** of hot air to force water out of one container and into another. Heron recognised, even if unwittingly, that it was possible to **perform work** by changing the condition of air.

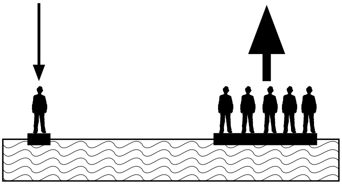


Fig. 1.5 :
Compressed air to increase energy

Pascal's law
Increasing energy

It was only in the 17th century that a series of learned people began to study the physical laws applicable to compressed air. In 1663 Blaise Pascal published an essay on **increasing energy** by using liquids (hydraulics), that was also valid for the technology of compressed air. He found that the energy exerted by one man at one end of a closed container of water was equivalent to the energy exerted by 100 men at another end.

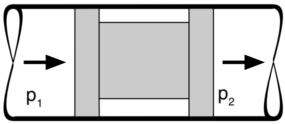


Fig. 1.6 :
Compressed air as a means of transport

Transporting objects through pipes
Pneumatic conveyance

Taking up where Heron left off, the French physicist Denis Papin described in 1667 a method of transporting objects through pipes. He exploited the slight difference in pressure inside a pipe. In doing so he found out that energy was generated at an object inside the pipe. This was recognition of the advantage of the high work speeds obtainable by using air. Papin thus laid the foundation stone for **pneumatic conveyance.**

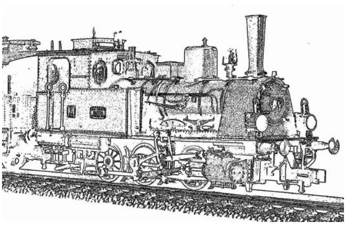


Fig. 1.7 :
Pneumatic brakes in a train ca. 1870

Pneumatic brakes
Power transmission

As early as around 1810, trains were being powered by compressed air. In 1869 Westinghouse introduced his pneumatic brake. His brake motor followed three years later. In this system the brakes were applied **by over-pressure** i.e., the full braking effect is obtained if there is a drop in pressure e.g., by the bursting of a hose.

This was the first use of a fail-safe system. Brake systems based on this principle are still used in HGVs today.

Pneumatic post

Conveyance by compressed air

The idea of trains powered by compressed air was not forgotten. In 1863, Latimer Clark together with an engineer named Rammel built a pneumatic conveyance system in London. It featured small trolleys moving completely inside conveyor tubes and was designed to transport postal bags and parcels. This system was much more flexible than the heavy, atmospheric railways of 1810, and led eventually to the introduction of pneumatic post.

Pneumatic post networks soon sprung up in Berlin, New York and Paris. The Paris network reached its peak length of 437 km in 1934. Even today, pneumatic post systems are still used in large industrial operations.

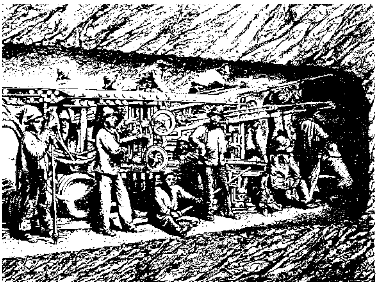


Fig. 1.8 :
Pneumatic drills in tunnel construction

Pneumatic tools

Transporting energy

When the tunnel through Mont Cenis was being built in 1857, the new technology was used in a pneumatically-powered hammer drill to cut through the rock. From 1861 they used pneumatically-powered percussion drills, these being supplied with compressed air from compressors at both ends of the tunnel. In both cases the compressed air was transported over long distances.

When in 1871 the breakthrough in the tunnel was achieved, there were over 7 000 m of pipelines on both sides. Thus, for the first time, the **transportability of energy** was demonstrated and made known to a wide public as one of the advantages of compressed air. And from here on, pneumatic tools of even greater performance and versatility were developed.

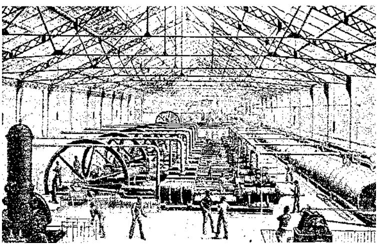


Fig. 1.9 :
Compressed air station in Paris 1888

Pneumatic networks

Central generation of compressed air and signal transmission

The experience gained using networks of pneumatic lines and the development of more powerful compressors led to a pneumatic network being installed in the sewage canals of Paris. It was put into commission in 1888 with a **central compressor output** of 1 500 kW. By 1891 its output rating had already reached 18 000 kW.

The all-round success of the pneumatic network was underlined by the invention of a clock, the minute hand of which was moved on every sixty seconds by an impulse from the compressor station. People had not only seen the possibility of transporting energy, but also of moving **signals over great distances through a pneumatic network**.

The pneumatic network in Paris is unique to this day, and is still in use.

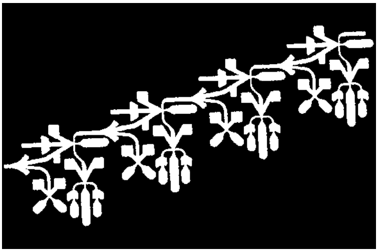


Fig. 1.10 :
Four-stage adding device with wall radiation elements

Signal processing
Compressed air for the transmission and processing of signals

In the 1950s in the USA the high flow speed of compressed air was first used for the transmission and processing of signals. **Low-pressure pneumatics**, also known as **fluidics or pneumonics (pneumatic logic)**, allow the integration of logical switching functions in the form of fluidic elements in a very small area at pressures of 1.001 to 1.1 bar.
The high operating precision of the fluidic logic elements under extreme conditions allowed them to be used in the space and defence programmes of the USA and the USSR. Immunity to electromagnetic radiation from exploding nuclear weapons gives fluidics a special advantage in several sensitive areas.

Even so, over the course of time fluidics has largely been superseded by electrical and microelectronic technology in the fields of signal and information processing.

1.2 Units and formula symbols

The **SI-units** (Système International d'Unités) were agreed at the 14th General Conference for Weights and Measures. They have been generally prescribed since 16.10.1971.

1.2.1 Basic units

The **basic units** are defined independent units of measure and form the basis of the **SI-system**.

Basic unit	Formula symbol	Symbol	Name
Length	l	[m]	Metre
Mass	m	[kg]	Kilogramme
Time	t	[s]	Second
Strength of current	I	[A]	Ampere
Temperature	T	[K]	Kelvin
Strength of light	I	[cd]	Candela
Qty of substance	n	[mol]	Mol

1.2.2 Compressed air units

Engineering uses measures derived from the basic units. The following table shows the most frequently used units of measure for **compressed air**.

Unit	Formula symbol	Symbol	Name
Force	F	[N]	Newton
Pressure	p	[Pa] [bar]	Pascal Bar **1 bar = 100 000 Pa**
Area	A	[m^2]	Square metre
Volume	V	[m^3] [l]	Cubic metre Litre **1 m^3 = 1 000 l**
Speed	v	[m / s]	Metre per Second
Mass	m	[kg] [t]	Kilogramme Tonne **1 t = 1 000 kg**
Density	ρ	[kg / m^3]	Kilogramme per cubic metre
Temperature	T	[°C]	Degree Celsius
Work	W	[J]	Joule
Energy	P	[W]	Watt
Tension	U	[V]	Volt
Frequency	f	[Hz]	Hertz

1.3 What is compressed air ?

1.3.1 The composition of air

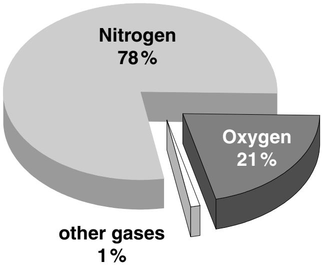


Fig. 1.11:
The composition of air

The air in our environment, the atmosphere, consists of:

78 % Nitrogen

21 % Oxygen

1 % other gases
(e.g.. carbon-dioxide and argon)

1.3.2 The properties of compressed air

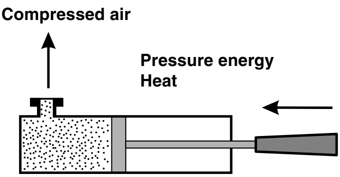


Fig. 1.12:
Air compression

Compressed air is compressed atmospheric air.

Compressed air is a carrier of heat energy.

Compressed air can bridge certain distances (in pipelines), be stored (in compressed air receivers) and perform work (decompress).

1.3.3 How does compressed air behave?

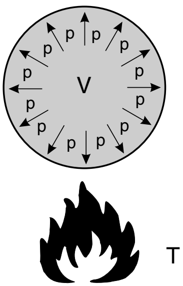


Fig. 1.13:
Air in a closed container

As with all gases, the air consists of molecules. The molecules are held together by molecular force. If the air is enclosed in a tank (constant volume), then these molecules bounce off the walls of the tank and generate **pressure p**.

The higher the **temperature**, the greater the movement of air molecules, and the higher the **pressure** generated.

Volume (V)	=	constant
Temperature (T)	=	is increased
Pressure (p)	=	rises

Boyle and Mariotte carried out experiments with enclosed volumes of gas independently of each other and found the following interrelationship:

The volume of gas is inversely proportional to pressure.
(Boyle-Mariotte's Law)

1.4 Physical fundamentals

The condition of compressed air is determined by the 3 measures of thermal state:

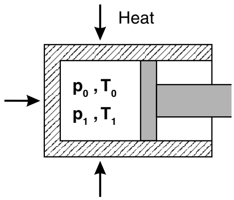
T = Temperature

V = Volume

p = Pressure

$$\frac{p \times V}{T} = \text{constant}$$

This means:

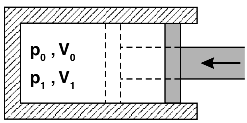


constant volume
isochore compression

Volume constant (isochore)
Pressure and temperature variable

When the temperature is increased and the volume remains constant, the pressure rises.

$$\frac{p_0}{p_1} = \frac{T_0}{T_1}$$

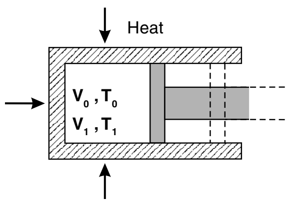


constant temperature
isotherm compression

Temperature constant (isotherm)
Pressure and volume variable

When the volume is reduced and the temperature remains constant, the pressure rises.

$$p_0 \times V_0 = p_1 \times V_1 = \text{constant}$$



constant pressure
isobar compression

Pressure constant (isobar)
Volume and temperature variable

When the temperature is increased and the pressure remains constant, the volume increases.

$$\frac{V_0}{V_1} = \frac{T_0}{T_1}$$

1.4.1 Temperature

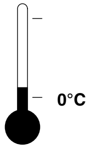
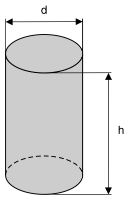


Fig.1.14:
Showing temperature

The temperature indicates the heat of a body and is read in °C on thermometers or converted to Kelvin (K).

$$\mathbf{T\ [K] = t\ [°C] + 273{,}15}$$

1.4.2 Volume



Volume (V)

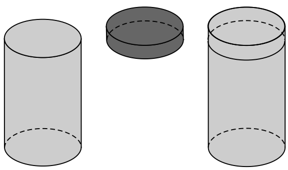
Volume V [l, m³]
Compressed air in expanded state, open air

The volume is determined, for example, by the size of a cylinder. It is measured in l or m³ and relative to 20 °C and 1 bar.

The numbers in our documentation always refers to compressed air in its expanded state.

$$V_{Cyl} = \frac{d^2 \times \pi}{4} \times h$$

V_{Cyl}	=	Volume	[m³]
d	=	Diameter	[m]
h	=	Height	[m]

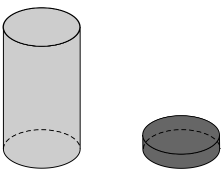


Norm volume 0°C + 8% = Volume 20 °C

Normal volume V_{Norm} [Nl, Nm³]
Compressed air in expanded state under normal conditions

The normal volume refers to the physical normal state as specified in DIN 1343. It is 8% less than the volume at 20 °C.

760 Torr = 1,01325 bar_{abs} = 101 325 Pa
273,15 K = 0 °C



0 bar_{abs} 8 bar_{abs}

Operating volume V_{operat} [Bl, Bm³]
Compressed air in compressed state

The volume in operating state refers to the actual condition. The temperature, air pressure and air humidity must be taken into account as reference points.

When specifying the operating volume the pressure must always be given, e.g., 1 m³ at 7 bar means that 1 m³ expanded (relaxed) air at 7 bar = 8 bar abs. is compressed and only occupies 1/8 of the original volume.

1.4.3 Pressure

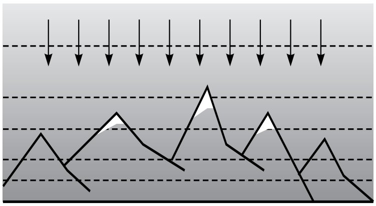
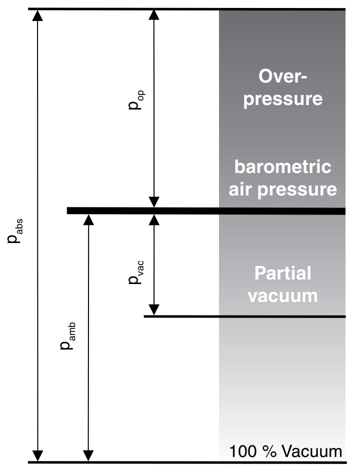


Fig.1.15:
Atmospheric pressure



p_{amb} = Atmospheric pressure
p_{op} = Over-pressure
p_{vac} = Partial vacuum
p_{abs} = Absolute pressure

Fig.1.16:
Illustration of different pressures

Atmospheric pressure p_{amb} [bar]

Atmospheric pressure is caused by the weight of the air that surrounds us. It is dependent on the density and height of the atmosphere.

At sea level, 1 013 mbar = 1,01325 bar
= 760 mm/Hg [Torr]
= 101 325 Pa

Under constant conditions atmospheric pressure decreases the higher the measuring location is.

Over-pressure p_{op} [bar_{op}]

Over-pressure is the pressure above atmospheric pressure. In compressed air technology, pressure is usually specified as over-pressure, and in bar without the index „op“.

Absolute pressure p_{abs} [bar]

The absolute pressure $\mathbf{p_{abs}}$ is the sum of the atmospheric pressure $\mathbf{p_{amb}}$ and the over-pressure $\mathbf{p_{op}}$.

$$\boxed{p_{abs} = p_{amb} + p_{op}}$$

According to the **SI-System** pressure is given in Pascal [Pa]. In practice, however, it is still mostly given in „bar“. The old measure atm (1 atm = 0,981 bar-op) is no longer used.

$$\text{Pressure} = \frac{\text{Force}}{\text{Area}} \qquad p = \frac{F}{A}$$

$$1\ \text{Pascal} = \frac{1\ \text{Newton}}{1\ \text{m}^2} \qquad 1\ \text{Pa} = \frac{1\ \text{N}}{1\ \text{m}^2}$$

1 bar = 10195 mmWH [mm water head]

1.4.4 Volume flow

Working volume flow
Induction rate

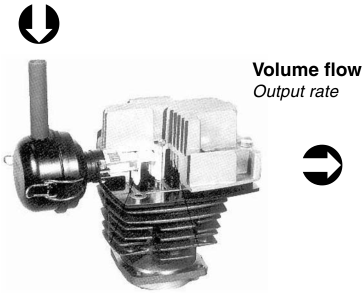


Fig. 1.17:
Working volume flow and volume flow

Volume flow $\dot{V}$ [l/min, m³/min., m³/h]

The volume flow describes the volume (l or m³) per unit of time (minute or hour).

A distinction is made between the working volume flow (induction rate) and the volume flow (output rate) of a compressor.

Working volume flow $\dot{V}_{Wor}$ [l/min, m³/min., m³/h]
Induction rate

The working volume flow is a calculable quantity on piston compressors. It is the product of the cylinder size (piston capacity), compressor speed (number of strokes) and the number of cylinders working. The working volume flow is given in l/min, m³/min or m³/h.

$$\dot{V}_{Wor} = A \times s \times n \times c$$

$\dot{V}_{Wor}$	=	Working volume flow	[l/min]
A	=	Cylinder area	[dm²]
s	=	Stroke	[dm]
n	=	Number of strokes (compressor speed)	[1/min]
c	=	Number of working cylinders	

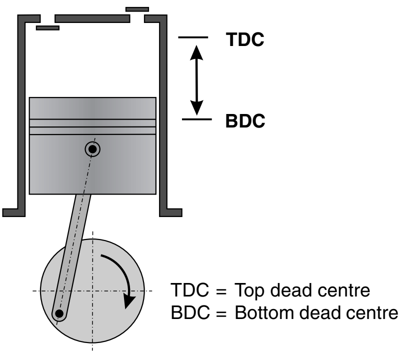


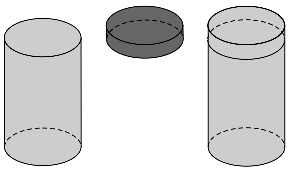
Fig. 1.18:
Cylinder movement

Volume flow $\dot{V}$ [l/min, m³/min, m³/h]
Output rate

The output rate of a compressor is normally declared as the volume flow.

In contrast to the working volume flow, the volume flow is not a calculated value, but one **measured** at the pressure joint of a compressor and calculated back to the induction state. The volume flow is dependent on the final pressure relative to the induction conditions of pressure, temperature and relative humidity. This is why when calculating the induction state the measured volume flow to induction pressure must be „relaxed“ and to induction temperature it must be „re-cooled“ and „dryed“ to a relative humidity of 0 %.

The volume flow is measured according to VDMA 4362, DIN 1945, ISO 1217 or PN2 CPTC2 and given in l/min, m³/min or m³/h. The effective volume flow, i.e., the output that can actually be used, is an important consideration for the design of a compressor. Volume flows can only usefully be compared when measured under the same conditions. This means that the induction temperature, pressure, relative air humidity and measured pressure must match.



Norm volume flow + 8 % = Volume flow
0°C 20°C

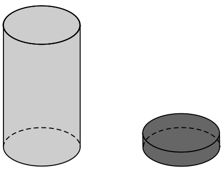
Fig. 1.19:
Norm volume flow

Norm volume flow $\dot{V}_{Norm}$ [Nl/min, Nm³/min, Nm³/h]

As with the volume flow, the norm volume flow is also **measured**.

However, it does not refer to the induction state, but to a theoretical comparative value. With the physical norm state the theoretical values are:

Temperature	=	273,15 K	(0 °C)
Pressure	=	1,01325 bar	(760 mm HG)
Air density	=	1,294 kg/m³	(dry air)
rel. humidity	=	0 %	



1 bar_{abs} 8 bar_{abs}

Fig. 1.20:
Operating volume flow

Operating volume flow $\dot{V}_{Operat}$ [Ol/min, Om³/min, Om³/h]

The operating volume flow gives the effective volume flow of compressed air.

To be able to compare the operating volume flow with the other volume flows, the pressure of he compressed air must always be given in addition to the dimension Ol/min, Om³/min or Om³/h.

1.5 Compressed air in motion

Different laws apply to compressed air in motion than to stationary compressed air.

1.5.1 Flow behaviour

The volume flow is calculated from area and speed.

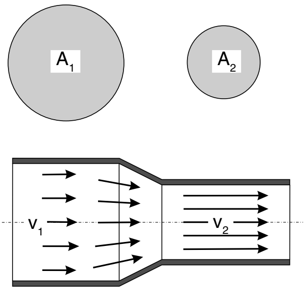


Fig. 1.21:
Flow behaviour

$$\dot{V} = A_1 \times v_1 = A_2 \times v_2$$

$$\frac{A_1}{A_2} = \frac{v_2}{v_1}$$

$\dot{V}$ = Volume flow
A_1, A_2 = Cross section
v_1, v_2 = Speed

The result of the formula is that:

The speed of flow is inversely proportional to the cross section.

1.5.2 Types of flow

Flow can be laminar or even (Ideal),
or turbulent (with backflow and whirling).

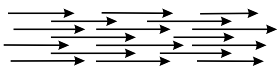


Fig. 1.22:
Laminar flow

Laminar flow (even flow)

low drop in pressure
slight heat transition

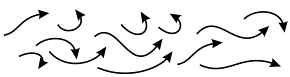


Fig. 1.23:
Turbulent flow

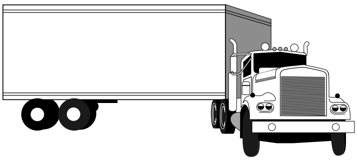
Turbulent flow (whirl flow)

high drop in pressure
great heat transition

2. Applications for pneumatics

2.1 The advantages of compressed air

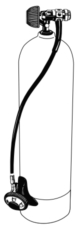
Pneumatics faces increasing competition from mechanical, hydraulic and electrical appliances on all fronts. But pneumatic devices have fundamental advantages over the other technologies:



Easily transported

Air is available everywhere, and there is plenty of it. Since outlet air escapes into the open, there is no need for return lines. Electrical and hydraulic systems need a return line to the source.

Compressed air can be transported over great distances in pipelines. This allows the installation of central generation stations that can supply points of consumption via ring mains with a constant working pressure. The energy stored in compressed air can be widely distributed in this way.



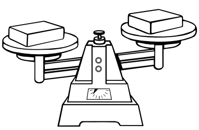
Easily stored

It is easy to store compressed air in purpose-built tanks. If there is a storage tank integrated in a pneumatic network, the compressor only needs to work when the pressure drops below a critical level. And because there is always a cushion of pressure, a work cycle can be completed even if the power network fails.

Transportable compressed air bottles can also be used at locations where there is no pipe system (e.g., under water).

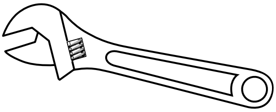
Clean and dry

Compressed air does not cause soiling or leave drops of oil if the lines are defective. Cleanliness in fitting and operation are extremely important factors in many sectors of industry, e.g., food, leather, textiles, and packing.



Lightweight

Pneumatic devices are usually much lighter than comparable equipment and machinery with electrical power units. This makes a big difference with manual and percussion tools (pneumatic screwdrivers and hammers).



Safe to use

Compressed air works perfectly even when there are great temperature fluctuations and the temperatures are extreme. It can also be used where there are very high temperatures, e.g., for operating forge presses and blast furnace doors.

Pneumatic devices and lines that are untight are no risk to the safety and serviceability of the system.

Pneumatic systems and components in general wear very little. They therefore have a long working life and a low failure rate.

Accident-proof

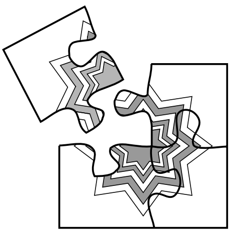
Pneumatic elements are very safe with regard to fire, explosion and electrical hazards. Even in areas where there is a risk of fire, explosion and extreme weather conditions, pneumatic elements can be used without large and expensive safety apparatus. In damp-rooms or outdoors too, there is no danger with pneumatic equipment.

Rational and economical

Pneumatics is 40 - 50 times more economical than muscle power. This is a major point, particularly in mechanisation and automation.

Pneumatic components are cheaper than the equivalent hydraulic components.

There is no need for regular medium changes, as with hydraulic equipment, for instance. This reduces costs and the servicing requirement, and increases operating times.



Simple

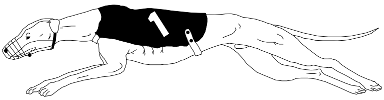
The design and operation of pneumatic equipment is very simple. For this reason it is very robust and not susceptible to malfunctioning. Pneumatic components are easy to install and can be re-used later without difficulty. Installation times are short because of the simple design. The fitters require no expensive special training.

Straight-line movements can be executed without extra mechanical parts such as levers, cams, eccentric disks, screw spindles and the like.

Overload-proof

Compressed air equipment and pneumatic working parts can be loaded until they stop without being damaged. This is why they are considered to be overload-proof.

In contrast to electrical systems, the output of a pneumatic network can be overloaded without risk of danger. If the pressure drops too much, the work can not be done, but there will be no damage to the network or its working elements.



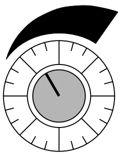
Fast work medium

The very high flow speeds allow rapid completion of work cycles. This provides short cut-in times and fast conversion of energy into work.

Compressed air can achieve flow speeds of over 20 m/s. Hydraulic applications only manage 5 m/s.

The pneumatic cylinders reach linear piston speeds of 15 m/s.

Maximum control speeds in signal processing lie between 30 and 70 m/s at operating pressures of between 6 and 8 bar. With pressures of less than 1 bar it is even possible to obtain signal speeds of 200 to 300 m/s.



Fully adjustable

Travel speeds and exerted force are fully and easily adjustable. Both with linear and rotary movement, force, torque and speeds can be fully adjusted without difficulty by using throttles.

2.2 Pressure ranges

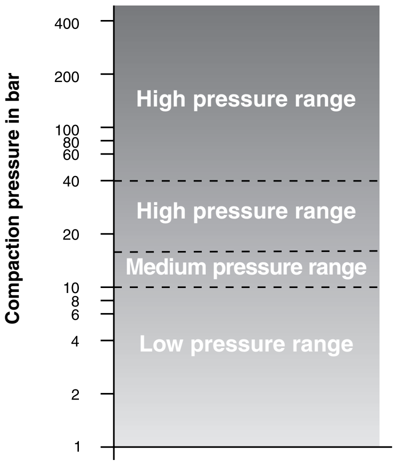


Fig 2.1 :
Pressure ranges

Low pressure range to 10 bar

Most pneumatic applications in industry and the crafts lie in the low pressure range of 10 bar and below.

Compressors used :

– one and two-stage piston compressors
– single-stage screw compressors with oil-injection cooling
– two-stage compressors
– rotary compressors

Medium pressure range to 15 bar

HGV and other heavy vehicle tyres are filled with compressed air from 15 bar compressors. There are also other special machines that operate with such pressures.

Compressors used :

– two-stage piston compressors
– single-stage screw compressors (up to 14 bar) with oil-injection cooling

High pressure range to 40 bar

The compressors in this pressure range are generally used for starting large diesel engines, testing pipelines and flushing plastic tanks.

Compressors used :

– two and three-stage piston compressors
– multi-stage screw compressors

High pressure range to 400 bar

One example of the use of compressed air in the high pressure range is the storage of breathing air in diving bottles. High pressure compressors are used in power stations, rolling mills and steel works and for leak testing. Compressors of this type are also used for compressing utility gases, such as oxygen.

Compressors used:

– three and four-stage piston compressors

2.3 Possible applications for compressed air

Compressed air is used intensively in all sectors of industry, the crafts, and everyday life. The range of possible applications is diverse and all-embracing. Some of the technical uses are mentioned and explained briefly below.

In view of the versatility of this medium it is only possible to outline a few of the possible applications. The arrangement of the chapter can not be unambiguous since the criteria for assessment and differentiation are too varied.

2.3.1 Tensioning and clamping with compressed air

Tensioning and clamping with compressed air is mainly used in applications involving mechanisation and automation. Pneumatic cylinders or motors fix and position the tools needed for work processes. This can be done by linear and rotary movement, and also by swivel movement. The energy in the compressed air is converted directly into force and movement through the exertion of pressure. The amount of tensioning force required must be dispensed with precision.

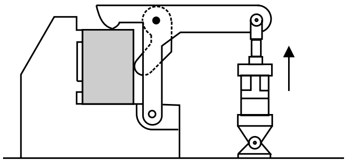


Fig. 2.2:
Pneumatic-mechanical clamp

2.3.2 Conveyance by compressed air

Conveyance by compressed air is found in mechanisation and automation. In these applications, motors and cylinders are used for timed or untimed conveyance, or according to work processes. Automated storage and receipt also belongs in this category, as does the turn-around of tools and other items on longer conveyor belts.

Another variation of pneumatic transport is the conveyance of bulk material and liquids through pipes. With this method, granulates, corn, powder and small parts can be quickly and comfortably conveyed over relatively long distances. The pneumatic post concept also belongs in this category.

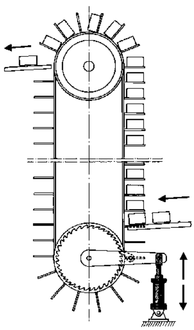


Fig. 2.3:
Bridging the heights with a pneumatically powered elevator

2.3.3 Pneumatic drive systems

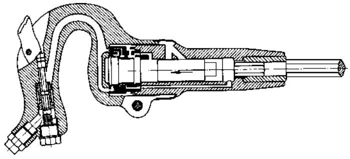


Fig. 2.4:
Valveless pneumatic hammer

Pneumatic drive systems are found in all areas of industry and the crafts. These can perform rotary and linear movements. Linear movement with the aid of cylinders in particular is seen as a highly economical and rational application. The utility work is performed by dropping the pressure and changing the volume of the compressed air.

Pneumatic percussion machinery and tools (e.g., pneumatic hammers) are of great importance in this category. The energy in the compressed air is converted into kinetic energy for a moving piston. Vibrators and jolting devices belong to this category.

Pneumatic power is also used by a multitude of valves and slides, tools, adjustment devices, feed systems and vehicles.

2.3.4 Spraying with compressed air

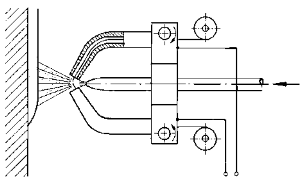


Fig. 2.5:
Arc-type metal spraying system

With **Spraying** applications, the energy of the expanding compressed air is used to force materials or liquids through a spray nozzle. This procedure is used to apply or atomise various substances.

Surface treatment processes, such as sand and gravel blasting, shot peening and painting with spray-guns belong to this category. Concrete and mortar are also applied using this method.

If high temperatures are also used, compressed air can be utilised for applying liquid metals. Arc-type spraying is an example worthy of mention here..

Another application is the atomisation of liquids through spray nozzles, e.g., for spraying weedkillers and insecticides.

2.3.5 Blowing and flushing with compressed air

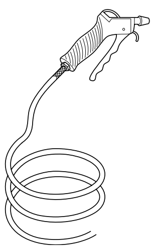


Fig. 2.6:
Air gun with spiral hose

When **blowing and flushing** the compressed air itself is the work medium and tool. The flow speed generated by dropping pressure and/or the expanding volume performs the utility work.

Examples of this type of work are blowing out glass or plastic bottles, blowing out and cleaning tools and moulds, fixing light tools for processing or conveyance and flushing out metal chips and residue. Compressed air in this form can also be used to let off heat.

2.3.6 Testing and inspection with compressed air

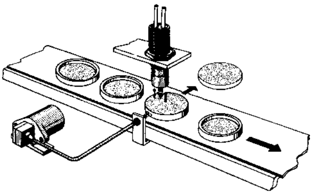


Fig. 2.7:
Reflex nozzle with impulse emitter

In **pneumatic testing and inspection procedures,** the changes in pressure at the measuring point are used to determine spacings, weights and changes in shape. This allows passing articles to be counted, correct positioning to be checked and the presence of workpieces to be ascertained.

This process is an integral part of many sorting, positioning and processing systems.

2.3.7 Using compressed air for process control

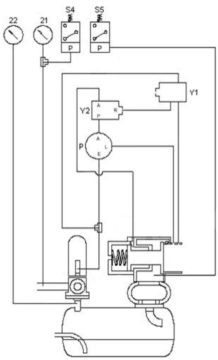


Fig. 2.8:
Diagram of a BOGE screw compressor, air-cooled version with fully-adjustable output control

All pneumatic applications must be controlled by some means. They must receive instructions.

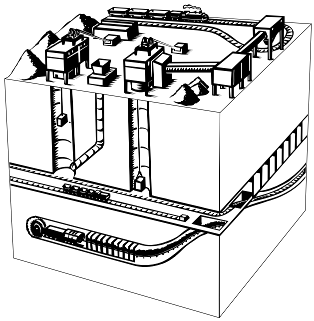
In general this is done by press-switches, direction valves and so forth. These control mechanisms are in turn actuated in many different ways, e.g., by mechanical switches, cams, or by hand. Electrical and magnetic switches are also in widespread use. The results determined by pneumatic process control systems can be used directly by direction valves or press-switches.

Pneumatics is of great importance for checking flow processes with liquids and gases. It is used for the remote control of valves, slides, and flaps in large industrial installations.

Pneumatics (fluidics) is also used for information processing and logical switching. These logic plans are comparable with integrated electronic circuits. They require much more space, but are characterised by high operating precision in certain applications. If the demands on the logic elements are not too high, fluidics can offer an alternative.

2.4 Examples of specialised applications

The following list will give the reader an idea of the many applications of compressed air in industry, the crafts and everyday life. Obviously, it is not possible to list all the possibilities for pneumatics since new areas appear and old ones become disused in the course of development and progress. This can therefore only be an incomplete summary of typical applications to be found in the various sectors of the economy.
A list of the typical applications in general mechanical engineering has not been included, since pneumatics touches practically every area, and mentioning all would be beyond the scope of this manual.

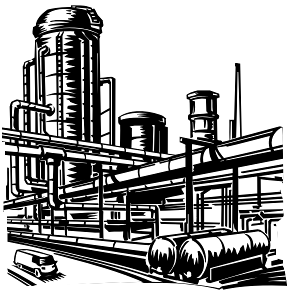


Construction trade

- Drill and demolition hammers (hand rams)
- Concrete compactors
- Conveyor systems for brickworks and artificial stone factories
- Conveyor systems for concrete and mortar

Mining

- Rock drilling hammers and carriage systems
- Loading machinery, shuttle and demolition cars
- Pneumatic hammers and chisels
- Ventilation systems

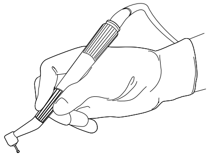


Chemicals industry

- Raw material for oxidation processes
- Process control
- Remote-controlled valves and slides in process circuits

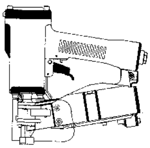
Energy industry

- Inserting and withdrawing reactor rods
- Remote-controlled valves and slides in steam and coolant circuits
- Ventilation systems for boiler houses



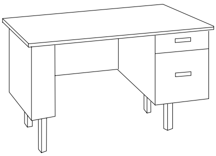
Health system

– Power packs for dentists' drills
– Air for respiration systems
– Extraction of anaesthetic gases



The crafts

– Staplers and nail guns
– Paint spray-guns
– Drills and screwdrivers
– Angle grinders

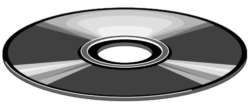


Wood processing industry

– Roller adjustment for frame saws
– Drill feed systems
– Frame, glue and veneer presses
– Contact and transport control of wooden boards
– Removal of chips and sawdust from work areas
– Automatic pallet nailing

Steel mills and foundries

– Carbon reduction in steel production
– Jolt squeeze turnover machines
– Bundling machinery for semi-finished products
– Coolants for hot tools and systems



Plastics industry

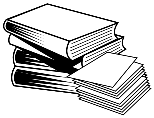
– Transport of granulate in pipes
– Cutting and welding equipment
– Blowing workpieces from production moulds
– Locking mechanisms for casting moulds
– Shaping and adhesive stations

Agriculture and forestry

– Plant protection and weed control
– Transport of feed and grain to and from silos
– Dispensing equipment
– Ventilation systems in glasshouses

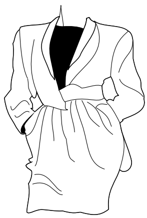
Food and semi-luxury food industry

– Filling equipment for drinks
– Closing and checking devices
– Bulk packing and palleting machinery
– Labelling machines
– Weighing equipment



Paper-processing industry

– Roller adjustment and feed machinery
– Cutting, embossing and pressing machinery
– Monitoring of paper reels



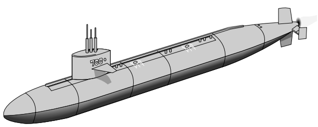
Textiles industry

– Thread detectors
– Clamping and positioning equipment in sewing machines
– Sewing needle and system cooling
– Stacking devices
– Blowing out residual material and dust from sewing

Environmental technology

– Forming oil barriers in the water
– Enriching water with oxygen
– Keeping lock gates free of ice
– Slide actuation in sewage plants
– Increasing pressure in the drinking water supply
– Mammoth pump for submarine applications

Traffic and communications



– Air brakes in HGVs and rail vehicles
– Setting signals, points and barriers
– Road-marking equipment
– Starting aids for large diesel engines
– Blowing out ballast tanks in submarines

3. Compressed air generators

Compressors (compactors)

are engines used for pumping and compressing gases to any pressure.

Ventilators

are flow machines that pump nearly atmospheric air.

With ventilators only slight changes to density and temperature occur.

Vacuum pumps

are machines that induct gases and steam in order to create a vacuum.

3.1 Compressors (compactors)

3.1.1 Dynamic compressors (Turbo-compressors)

Dynamic compressors are for instance turbo-compressors, by which running wheels equipped with blades accelerate the gas to be compressed. Fixed direction gear on the blades converts speed energy into pressure energy.

Dynamic compressors are to be preferred for **large quantities of medium** and **low medium pressures**.

3.1.2 Displacement compressors

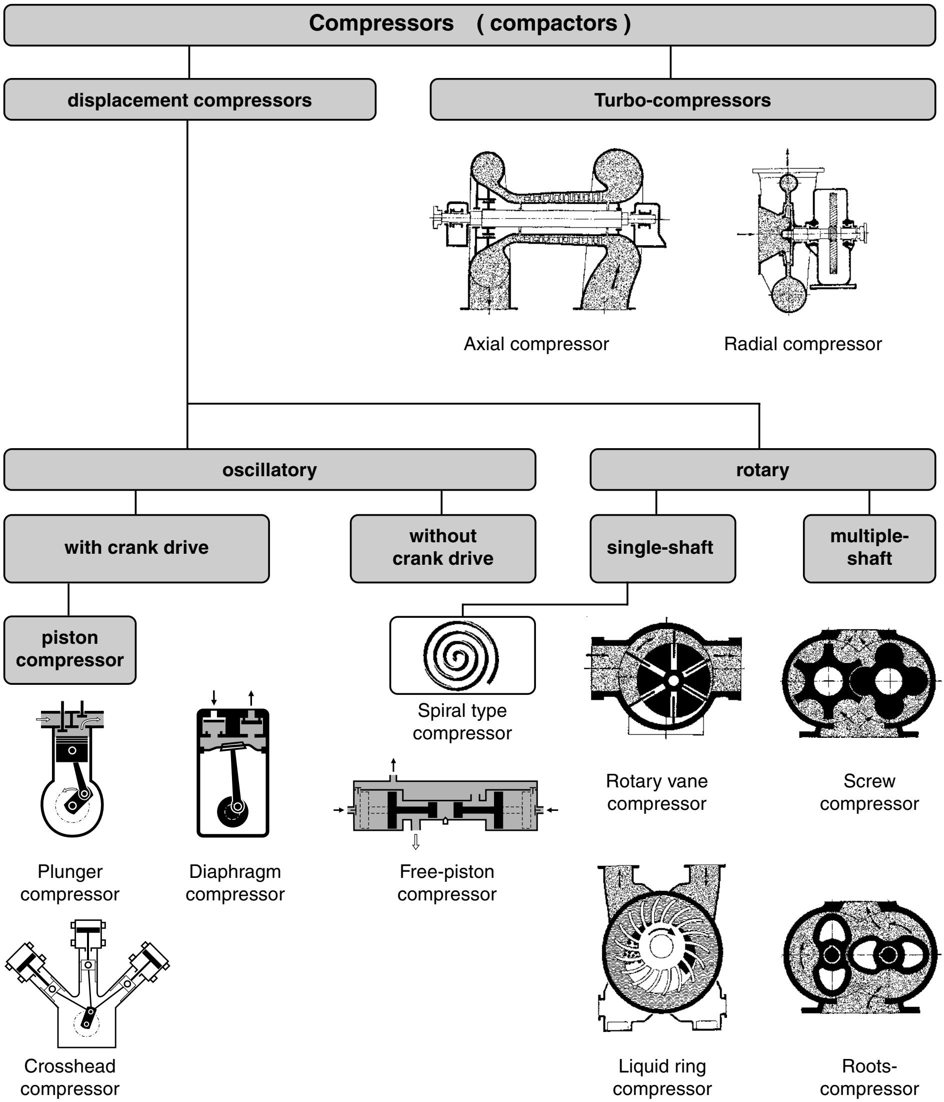
On displacement compressors the compression chamber closes completely after taking in the air. The volume is reduced and the air compressed by force.

Displacement compressors are to be preferred for **small quantities of medium** and **high medium pressures**.

3.2 Types of compressor

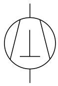
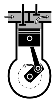
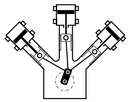
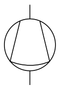
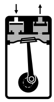
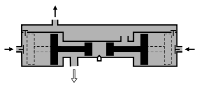
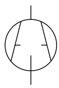
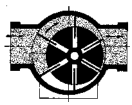
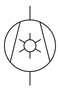
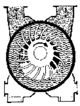
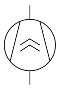
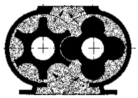
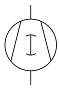
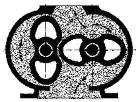
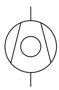
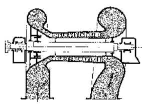
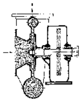
The summary shows the compressors divided according to their operating principle.

With all compressors, a distinction is drawn between non-oil-lubricated and oil-lubricated compressors.



3.2.1 Standard compressors

The table shows the typical areas of work for various standard types of compressor.

Type	Symbol	Op. diagram	Pressure range [bar]	Volume flow [m^3 / h]
Plunger compressor			10 (1-stage) 35 (2-stage)	120 600
Crosshead compressor			10 (1-stage) 35 (2-stage)	120 600
Diaphragm compressor			low	low
Free piston compressor			limited use as gas generator	
Rotary vane compressor			16	4 500
Liquid ring compressor			10	
Screw compressor			22	3 000
Roots compressor			1,6	1 200
Axial-compressor			10	200 000
Radial-compressor			10	200 000

3.2.2 Piston (reciprocating) compressor

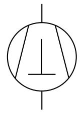


Fig. 3.1:
Symbol for piston compressor

Piston compressors draw in air by way of pistons moving up and down, compress it and then push it out. The processes control induction and pressure valves.

By arranging several compression stages in series it is possible to generate **various pressures,** and **differing quantities of air** can be generated by using several cylinders.

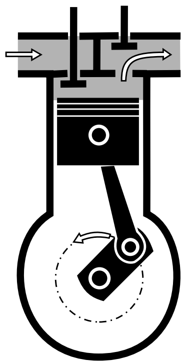


Fig. 3.2:
Op. diagram of plunger compressor

Plunger compressor

On plunger compressors, the piston is connected directly to the crankshaft via the con-rod.

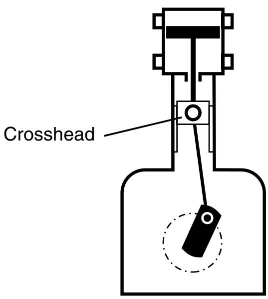


Fig. 3.3:
Op. diagram of crosshead compressor

Crosshead compressor

The piston is powered by a piston rod and that by the crosshead.

Properties of piston compressors:

– Highly efficient.

– High pressures.

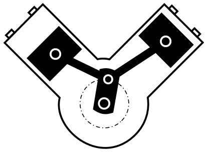


Fig. 3.4:
V-type plunger compressor

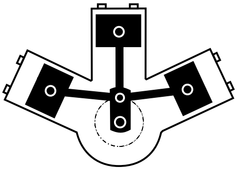


Fig. 3.5:
W-type plunger compressor

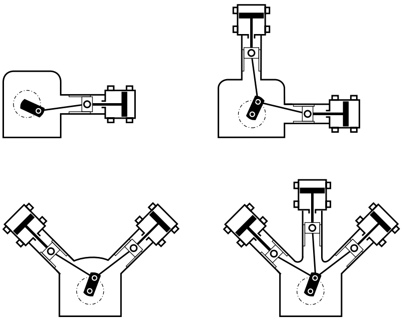


Fig. 3.6:
Crosshead compressors
Horizontal, L-type, V-type, W-type

The piston compressors are differentiated according to the **arrangement of their cylinders**:

– Vertical cylinders.
  No stress on the piston or piston ring through the weight of the piston.
  Small base area.

– Horizontal cylinders.
  Only as multi-cylinder compressor in Boxer construction.
  Low forces of gravity. This benefit is only noticeable when output is greater.

– V-, W- or L-type compressors.
  Good mechanical balance.
  Low space requirement.

3.2.3 Diaphragm compressors

The diaphragm compressor belongs to the family of displacement compressors.

An elastic diaphragm causes the compression. Instead of a piston moving linear between two end positions, the diaphragm is moved in non-linear vibrations. The diaphragm is attached to the side and is moved by a con-rod. The stroke of the con-rod depends on the elasticity of the diaphragm.

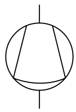


Fig. 3.7:
Symbol for diaphragm compressor

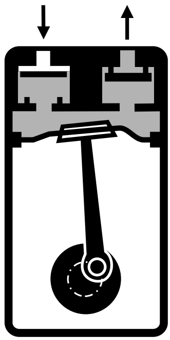


Fig. 3.8:
Op. diagram of diaphragm compressor

Features:

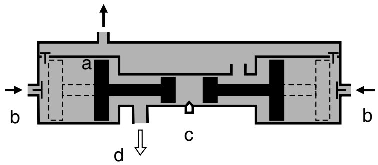
- Large cylinder diameter.
- Small stroke.
- Economical with low output quantities, low pressures, and when generating a vacuum.

3.2.4 Free piston compressor

The free piston compressor belongs to the family of displacement compressors.

It is a compressor with an integrated two-stroke diesel engine.

Compressed air acts on the raised pistons and pushes them back inside, thereby starting the compressor. The combustion air thus compressed in the engine cylinder drives the pistons apart again upon combustion of the injected fuel. The enclosed air is compressed. After letting out the necessary scavenging air the greater part of the compacted air is pushed out through a pressure holding valve. Any remaining air is pushed back in by the piston for the new cycle. The induction valves draw in new air again.



a	=	Pneum. outlet aperture
b	=	Inlet aperture
c	=	Fuel injection nozzle
d	=	Exhaust aperture

Fig. 3.9:
Op. diagram of free piston compressor

Features:

- Highly efficient.
- Smooth-running.
- Simple principle, but seldom used.
  In practice, the piston movements need to be synchronised and extensive control equipment fitted.

3.2.5 Rotary vane compressor

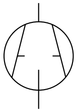


Fig. 3.10:
Symbol for rotary vane compressor

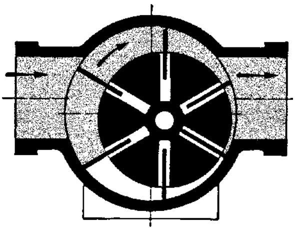


Fig. 3.11:
Op. diagram of rotary vane compressor

The rotary vane compressor (lamellar or rotary multi-vane compressor) is one of the rotary displacement compressors.

The housing and rotary pistons moving inside form the chamber for inducting and compressing the medium.

A cylindrical rotor on eccentric bearings turns inside a closed housing. The rotor (drum) has radial slots along its entire length. Inside the slots, slides move in a radial direction.

When the rotor reaches a certain speed, the working slide is pressed outwards against the inner walls of the housing by centrifugal force. The compression chamber between the rotor and the housing is divided by slides into individual cells (work chambers).

As a result of the eccentric arrangement of the rotor, the volume increases or decreases during a rotation.

The pressure chambers are lubricated by loss lubrication or oil injection.

By injecting larger quantities of oil into the compression chamber one achieves, in addition to lubrication, a cooling effect and a sealing of the slides against the inner wall of the housing. The injected oil can be separated from the compound of oil and air after compression and directed back to the oil circuit.

Features:

- Very quiet running.
- Pulse-free and even output of air.
- Low space requirement and easy to service.
- Low efficiency.
- High maintenance costs due to wear on the slides.

3.2.6 Liquid ring compressor

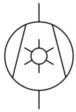
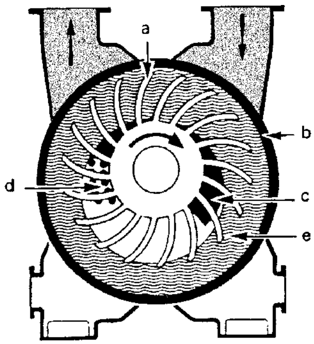


Fig. 3.12:
Symbol for liquid ring compressor



a = Paddle wheel
b = Housing
c = Inlet aperture
d = Outlet aperture
e = Liquid

Fig. 3.13:
Op. diagram of liquid ring compressor

The liquid ring compressor belongs to the category of rotary displacement compressors.

The eccentrically borne shaft in the housing with fixed radial paddle displaces the sealing liquid during rotation. This forms the liquid ring that seals the spaces between the paddles against the housing.

The content of the chamber is changed by the rotation of the shaft, causing air to be inducted, compressed and transported.

The liquid generally used is water.

Features:

– Oil-free air (through oil-free transport medium).

– Low sensitivity to soiling and chemicals.

– Liquid disperser required because auxiliary liquid is forced continually into the pressure chamber.

– Low degree of efficiency.

3.2.7 Screw compressor

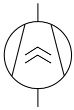


Fig. 3.14:
Symbol of screw compressor

The screw compressor is a rotary displacement compressor.

Two parallel rotors with differing profiles work in opposite directions inside a housing.

The intake air is compressed in chambers, which continuously decrease in size due to the rotation of the rotors until the final pressure is reached, and is then forced out of the discharge outlet. The chambers are formed by the casing walls and the meshing helical gears of the rotors.

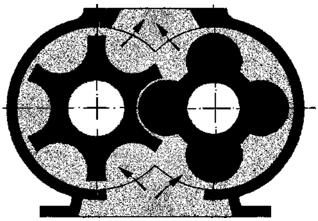


Fig. 3.15:
Op. diagram of screw compressor

Oil-free screw compressors

On screw compressors that seal without oil, and with which the air in the compression chamber does not come into contact with oil, the two rotors are connected by a synchronised transmission so that the surface profiles do not touch.

Screw compressors with oil-injection cooling

On screw compressors with oil-injection cooling only the main rotor is under power. The secondary rotor turns without contact.

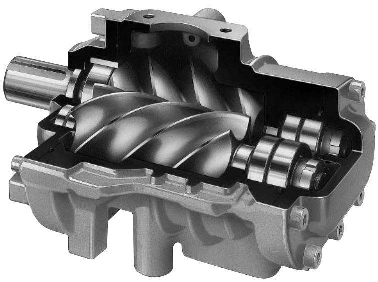


Fig. 3.16:
Section through screw compressor stage

Features:

- Small size.
- Continuous air production.
- Low final compression temperature.
  (with oil-injection cooling)

3.2.8 Roots compressor

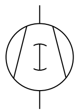


Fig. 3.17:
Symbol of Roots compressor

The Roots compressor belongs to the displacement family of compressors.

Two symmetrically shaped rotary pistons turn in opposite directions inside a cylindrical chamber. They are connected by a synchronised transmission and operate without contact.

The air to be compressed is directed from the intake side into the compressor case. It is enclosed in the chamber between the wing and case. At the moment in which the piston releases the edge to the pressure side the gas flows into the discharge outlet and fills the pressure chamber. When the wing turns further, the content of the transport chamber is pressed out against the full counter pressure. Constant compression takes place. The compressor must always work against the full dynamic pressure.

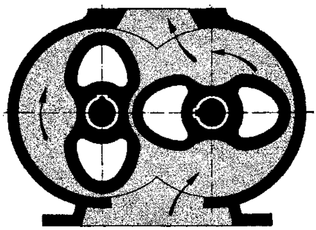


Fig. 3.18:
Op. diagram of Roots compressor

Features:

– No wear on the rotary piston, and therefore no lubrication is required.

– Air contains no oil.

– Sensitive to dust and sand.

3.2.9 Axial compressor

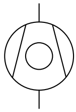


Fig. 3.19:
Symbol of turbo-compressor

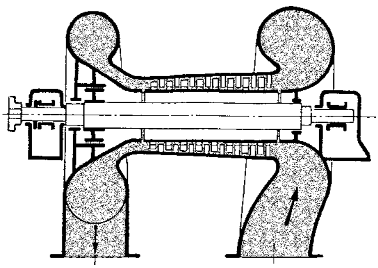


Fig. 3.20:
Op. diagram of axial compressor

Axial compressors are flow devices by which the air flows in alternatingly in an axial direction through a series of rotating and stationary paddles.

The air is first accelerated and then compressed. The paddle ducts form randomly expanded channels in which the kinetic energy generated by circulation of the air delays and is converted into pressure energy.

Features:

– Uniform output.

– No oil content in air.

– Sensitive to changes in load and stress.

– Minimum output quantities required.

3.2.10 Radial compressor

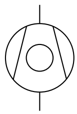


Fig. 3.21:
Symbol of turbo-compressor

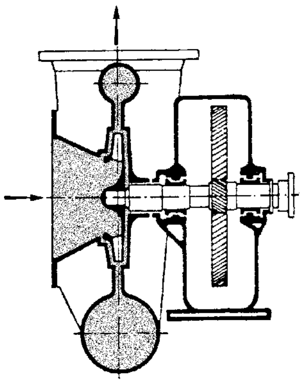


Fig. 3.22:
Op. diagram of radial compressor

Radial compressors are flow devices in which the air is directed to the centre of the rotating running wheel.

The air is moved by centrifugal force against the periphery. The rise in pressure is caused by the accelerated air being directed through a diffusor before it reaches the next running wheel. The kinetic energy (speed energy) converts into static pressure during this process.

Features:

– Uniform output.

– No oil content in air.

– Sensitive to changes in load and stress.

– Minimum output quantities required.

3.3 Piston compressors

3.3.1 General

Fig. 3.23:
BOGE piston compressor

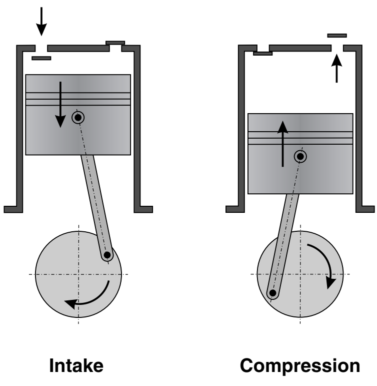


Fig. 3.24:
Principles

Piston compressors operate according to the displacement principle. The piston **intakes** air through the intake valve during the downwards stroke. It closes at the start of the downwards stroke. The air is **compressed** and forced out of the pressure valve. The piston is driven by a crank drive with crankshaft and conrods.

Piston compressors are available with one and several cylinders, and in one and multiple-stage versions.

Multi-cylinder compressors are used for higher outputs, multistage compressors for higher pressures.

Single stage compression
Compression to the final pressure in one piston stroke.

Two stage compression
The air compressed in the cylinder in the first stage (low pressure stage) is cooled in the intermediate cooler and then compressed to the final pressure in the second stage (high pressure cylinder).

Single action compressors
One compression action with one rotation of the crankshaft.

Double action compressors
Two compression actions with one rotation of the crankshaft.

Piston speeds

With compression the compression speed or even the motor speed is of secondary importance. The most important factor in assessing wear is the piston speed. So a compressor with a low speed and large stroke can have a high piston speed. In contrast, compressors with high speeds and a small stroke can have low piston speeds. The piston speed, measured in m/s, is extremely low with BOGE piston compressors. This means minimal wear.

3.3.2 Suction capacity - output

Suction rate
Suction rate

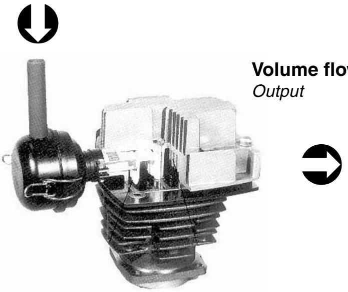


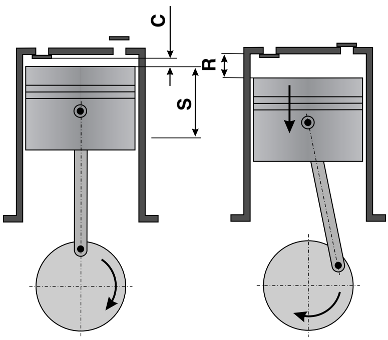
Fig. 3.25:
Suction rate and free air delivered

Suction rate - Output
Stroke volume flow - Volume flow

The suction rate (stroke volume flow) is a calculated size for piston compressors. It is the product of cylinder capacity, compressor speed (number of strokes) and the number of intake cylinders. The stroke volume flow is given in l/min, m^3/min and m^3/h.

The output (free air delivered FAD) is measured according to VDMA Unit Sheet 4362, DIN 1945, ISO 1217 or PN2 CPTC2.

The ratio of output to induction rate is the volumetric efficiency rate.



C = Clearance area
S = Stroke
R = Re-expansion

Fig. 3.26:
Clearance area

Clearance area

The clearance area is a specific dimension located between the top dead centre of the piston and the bottom edge of the valve.

The clearance area includes:

– Design tolerances

– Cavities in the valves and valve seats

– Individual design considerations

During the down stroke of the piston the air in the compression expands to atmospheric pressure. Only at this stage and during the continued downstroke of the piston is air sucked in from outside.

The difference between the suction rate and the output occurs because during suction the pressure of the air already drops in the inlet filter, leakages also occur, the air sucked in heats up and re-expansion occurs in the compression space.

3.3.3 Cooling

Fig. 3.27:
Direction of cooling air on a piston compressor

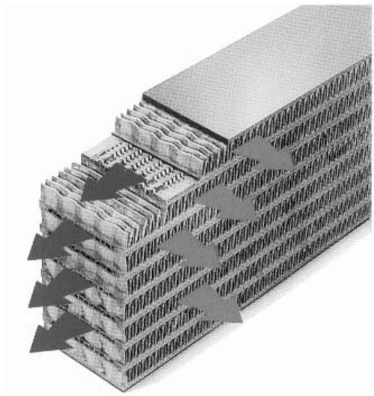


Fig. 3.28:
After-cooler as turbulence lamellar cooler

Heat is generated in all compression processes. The degree of heating depends on the final pressure of the compressor. The higher the final pressure, the higher the compression temperature.

According to safety rules, the final compression temperature on compressors with oil-lubricated pressure chambers and single stage compression, a maximum 20 kW motor rating and maximum 10 bar may be up to 220 °C.

With higher pressures and motor ratings a maximum temperature of 200 °C is allowed. With multiple stage compression and pressures of over 10 bar the maximum final compression temperature is 160 °C.

The greatest part of compression heat must therefore be expelled. High compressed air temperatures can be dangerous as a small amount of lubrication oil is absorbed into the compressed air during compression, this could be flammable. A fire in the line or the compressor would be the least danger, but with higher temperatures the danger of compressed air explosion is potentially greater because the ratio of oxygen contained is far greater than atmospheric air.

Each compressor stage therefore has an intercooler and aftercooler installed in order to cool the compressed air.

The quantity of heat to be removed by cooling depends on the free air delivered and the pressure. Higher pressure compressors have two, three, or more cylinders. The cylinders are located in the best position in the air flow of the cooling ventilator wherever possible. In order to intensify heat extraction, the surfaces of the cylinders and cylinder heads are produced with generous ribbing. However, the intensive cooling and ribbing of the compressor is not enough to obtain a minimum compressed air temperature. The compressed air must also be cooled by an intercooler between the first and second stages and an the aftercooler behind the second stage. If this cooling is not sufficient, multi-stage compression is necessary.

Safety regulation VGB 16 § 9 for oil-lubricated reciprocating compressors stipulates that the cooling air temperature must fall to between 60 °C and 80 °C after the last compression stage. It is also beneficial for the consumer to have a low compressor air outlet temperature, because the cooler compressed air contains less moisture. Apart from this, downstream equipment, such as the compressor receiver and air treatment components can be designed for low compressed air temperatures and thus be purchased at less cost. The air outlet temperature on air-cooled piston compressors is approx. 10 - 15 °C above ambient temperature, depending on the quality of the compressor.

3.3.4 Coolant

Piston compressors are mainly of the **air-cooled variety**. Cold air has the advantage that it is almost everywhere in unlimited quantities.

The cold air is generated by a ventilator. The ventilator forces the cold air over the intercooler and aftercooler and over the compressor.

During compression and cooling stage of the compressed air, condensate forms inside the cooler. Because of the flow speed of the compressed air, the condensate is taken out of the aftercooler by the air, and into the pipe network and compressed air tank.

3.3.5 Control of reciprocating piston compressors

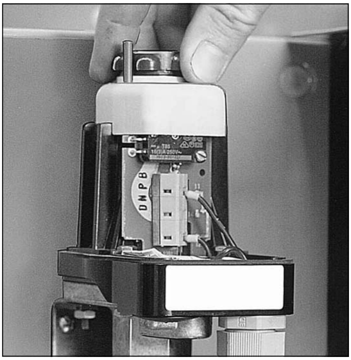


Fig. 3.29: Pressure switch

Piston compressors are normally controlled by pressure switches. The pressure switches must be located in a calm area of the compressed air. This is in the compressed air receiver, for example, and not in the pipeline between the compressor and the receiver.

The pressure switch stops the compressor at maximum pressure and switches it back on at 20 % below maximum pressure. The actuation is therefore 8 :10 bar and 12 :15 bar.

A smaller differential is not recommended because the compressor will then cycle too often and the wear on the compressor and the motor increases. The cut-in pressure can be lowered with the cut-out pressure remaining constant. This has the advantage that the compressor has longer running times but longer stationary times too. The cut-in pressure may not be lower than the minimum pressure of the pneumatic network.

Piston compressors do not continue running (running-on) but switch off immediately after the maximum pressure is reached (intermittent operation).

Piston compressors are particularly suitable as peak load machines. The compressor only switches on when there is an increased demand for compressed air and switches off without run-on time when the maximum pressure is reached, i.e., saving approx. 30 % energy consumption in idling mode.

3.3.6 Advantages of reciprocating piston compressors

– Compression of nearly all technical gases possible
– Economical compression of pressures up to 40 bar
– Can be used as a booster compressor
– Easy control
– Economical start-stop-operation (no idle running time)

3.3.7 Components of a piston compressor

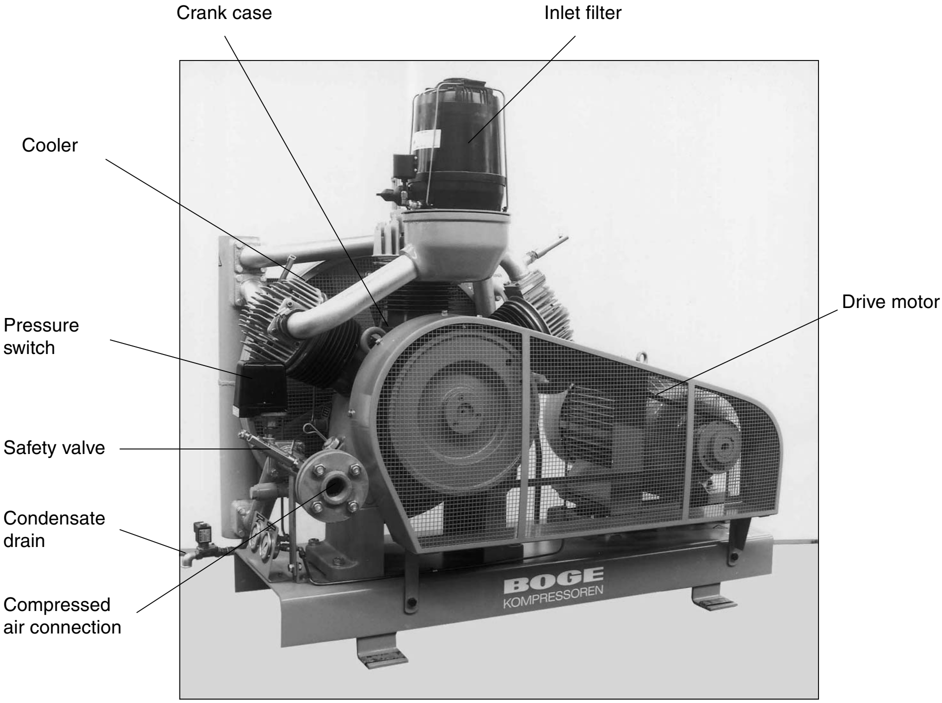


Fig. 3.30:
Layout of a piston compressor

3.4 Screw compressors

3.4.1 General

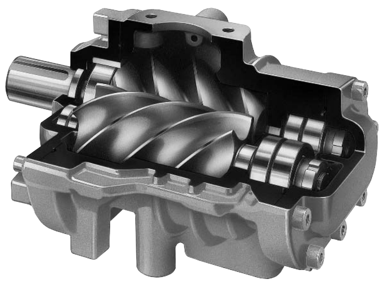


Fig. 3.31:
Section through a screw compressor air end

In contrast to the piston compressor, the screw compressor is a relatively new construction. Although the principle was developed as early as 1878 by Heinrich Krigar in Hannover, the construction was only perfected after the second world war. The Swedish company "Svenska Rotor Maskiner" (SRM) developed the screw compressor technically to series standard.

Screw compressors operate on the displacement principle. Two parallel rotors with different profiles work in opposite directions inside a housing.

3.4.2 Compression process

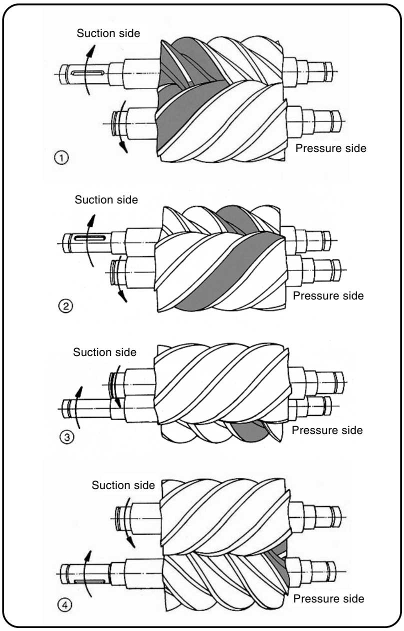


Fig. 3.32:
The compression process in a screw compressor stage

The intake air is compressed to final pressure in chambers which continuously decrease in size through the rotation of the screw rotors. When the final pressure is reached the air is forced out through the discharge outlet. The compression chambers are formed by the casing walls and the meshing helical profiles of the rotors.

Intake (1)

The air enters through the inlet aperture into the open screw profiles of the rotors on the intake side.

Compression (2) + (3)

The air inlet aperture is closed by the continued rotation of the rotors, the volume reduces and the pressure increases.

Oil is injected during this process.

Discharge (4)

The compression process is completed. The final pressure is reached and the discharge begins.

3.4.3 Method of operation

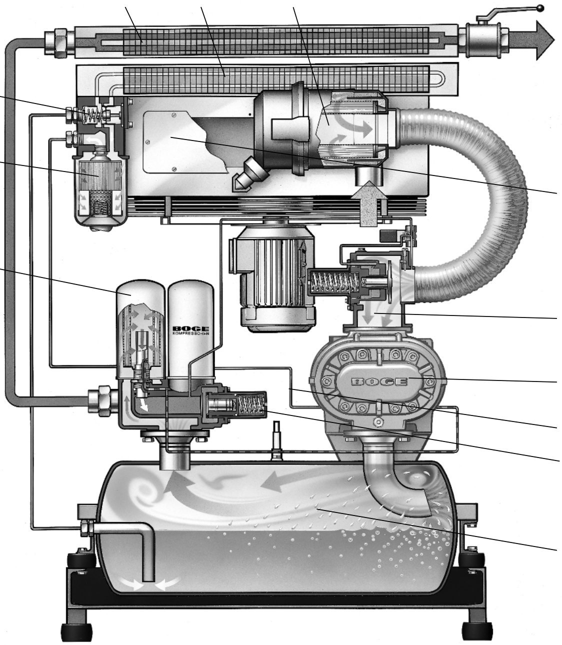


Fig. 3.33:
Sectional diagram of a
BOGE S-series screw compressor

1 = Intake filter with paper microfilter insert

2 = Multifunction suction controller

3 = Oil injection

4 = Compressor air end

5 = Oil separator tank

6 = Spin-on oil separator cartridge

7 = Minimum pressure valve

8 = Oil cooler

9 = Aftercooler parallel to flow of cool air

10 = Oil microfilter

11 = Thermostat valve

12 = Cleaning aperture

BOGE screw compressors draw in atmospheric air through the cyclonic suction filter **1** fitted with a paper microfilter cartridge and with soiled filter facility. After passing through the multi-function suction controller **2** the air enters the compressor stage and is compressed **4**. Continuously cooled BOGE oil is injected **3** into the compressor stage. The oil absorbs and removes the heat generated during the compression process which increases in temperature to approx. 85 °C. According to EC machinery guidelines the final maximum compression temperature may not exceed 110 °C.

A large proportion of the oil is separated from the compressed air in the combined air/oil separation vessel **5**. The residual oil is removed by the spin-on fine oil separator **6,** which removes the residual oil in the compressed air down to only approx. 1-3 mg/m^3.

The compressed air then passes through a minimum pressure valve **7** into the compressed air aftercooler **9** where it is cooled down to a temperature of only 8 °C above ambient and is then directed through the standard **BOGE** stop valve into the compressed air system.

The oil in the oil separator is cooled from 85°C to 55°C in the amply dimensioned oil cooler **8**. It then passes through a replaceable spin-on oil filter **10**. A thermostatic valve **11** in the oil circuit ensures that the oil temperature is ideal in every operating phase.

3.4.4 Oil circuit

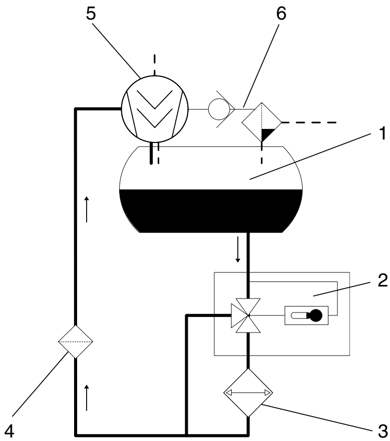


Fig. 3.34:
Components of the oil circuit

The oil injected into the compressor stage performs the following functions:

– Extraction of compression heat (cooling)
– Sealing the gap between the rotors and their housing
– Lubricating the bearings

1 = Compressed air/oil separator vessel
The oil is separated from the compressed air by reducing the air flow velocity in the separator vessel in which the oil collects

System pressure forces this oil out of the separator vessel into the compressor stage.

2 = Thermal bypass valve
The thermal bypass valve directs the oil through the oil cooler or through a bypass (e.g., in the warm-up stage).The oil is thus always at its optimum operating temperature.

3 = Oil cooler (air or water)
The oil cooler reduces the oil temperature to optimum conditions prior to injection into the compressor stage.

4 = Oil filter
The oil filter retains impurities from the oil and prevents problems of contamination in the oil circulation system.

5 = Compressor air end
The oil injected in the compressed air is directed back into the compressed air/oil vessel, where it is separated by gravitational forces.

6 = Scavenging line
The compressor air end draws any residual oil that has collected in the separator back into the oil circuit via the scavenging line.

3.4.5 Pneumatic circuit

The air sucked into the compressor air end is compressed to final pressure by the rotors.

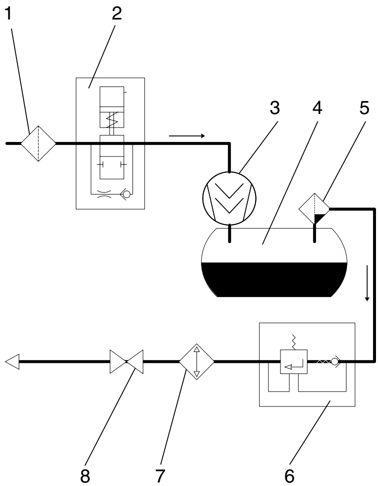


Fig. 3.35:
Components of the pneumatic circuit

1 = Intake filter
The intake filter cleans the air drawn in by the compressor stage.

2 = Suction controller
The suction controller opens (operation mode) or closes (idling mode and stopped) the intake line, depending on the operating status of the compressor.

3 = Compressor air end
The compressor stage compresses the intake air.

4 = Compressed air/oil vessel
Inside the compressed air/oil vessel the compressed air and oil are separated by gravity.

5 = Oil separator
The oil separator removes the residual oil from the compressed air.

6 = Minimum pressure valve MPV
This valve opens only when the system pressure has risen to 3.5 bar, which causes a fast build-up of system pressure and assures lubrication in the start-up and pressure phase of the compressor. When the compressor is switched off the minimum pressure valve prevents compressed air from flowing out of the compressor.

7 = Compressed air aftercooler (air cooled)
The compressed air is cooled in the aftercooler. During this phase, a large proportion of the moisture in the air condenses out.

8 = Stop valve
The screw compressor can be isolated from the system via the stop valve located at the outlet of the compressor.

3.4.6 Heat reclamation

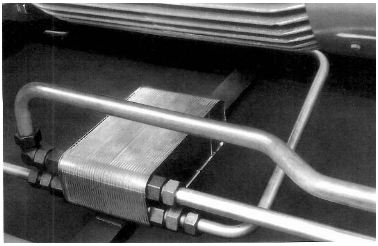


Fig. 3.36:
Heat exchanger BOGE-DUOTHERM

The oil removes approx 85% of compression heat from screw compressors with oil injected cooling. When using a heat exchanger the heat can be extracted from the oil and used for utility or water heating.

The water passing through the heat exchanger is heated to +70°C. The quantity of water heated depends on the temperature difference.

3.4.7 Intake control

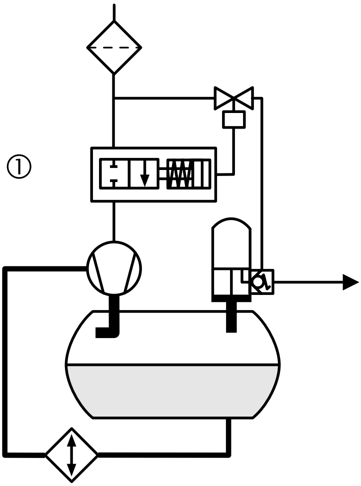


Fig. 3.37:
Intake control with ventilation/control valve

The suction controller controls the intake line of the screw compressor.

- Fully unloaded start-up through closed controllers.
- Seals hermetically on idling, stopped and emergency cut-out.

3.4.8 Advantages of screw compressors

- when compressed air is required on a continuous basis
- ideal as a base load machine
- economical with 100 % operating availability
- proportionale control possible
- ideal for use with frequency controller

3.4.9 Main components of a screw compressor

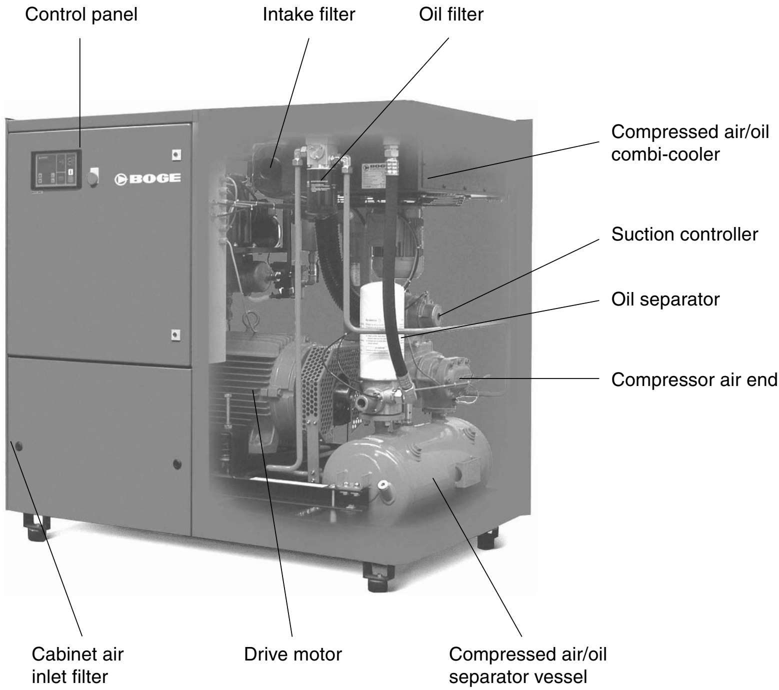


Fig. 3.38:
Layout of a screw compressor

3.5 Components

3.5.1 Drive motor

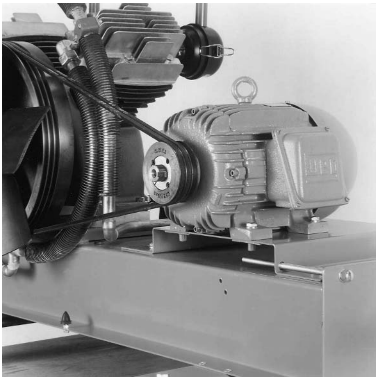


Fig. 3.39:
Drive motor with belt and tensioner

Drive motors are normally AC motors and mainly operate at a a speed of 3.000 min^{-1}. The appropriate compressor speed is obtained by drive belt transmission.

Normal drive motor supply is TEFV (totally enclosed fan vented) IP 55 class F insulation.

3.5.2 Drive belts

The compressor is driven via drive belt transmission.

Using the BOGE patented GM-drive system on screw compressors, drive belts are practically maintenance-free and have a calculated design life of up to 25,000 hours, depending on site conditions.

3.5.3 Belt tensioning

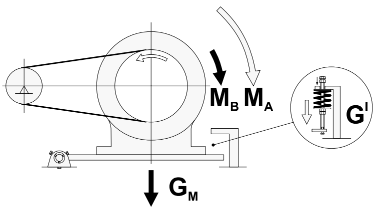


Fig. 3.40:
BOGE-GM-drive system

Motors on piston compressors are normally located on a sliding plate for belt tensioning. The plate is fitted with a threaded central spindle which together with parallel guides ensure accurate alignment of the drive belts across the pulleys.

BOGE screw compressors are equipped with the patented **BOGE-GM-drive system**. This take account of different belt tension forces caused by motor weight, start-up torque and running torque, and ensures that compressors have constant belt tension in every operating stage, without the need for retensioning and alignment on belt change.

3.5.4 Inlet and pressure valves

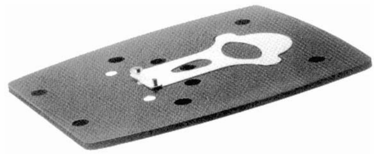


Fig. 3.41:
BOGE-ferax®-Tongue valve

The tongue valve controls the inlet and outlet of air in the cylinder chamber of the **piston compressor**.

BOGE-ferax®-tongue valves have fewer components than conventional valves, with friction-free operation, minimal dead space flow resistance. This means more FAD, higher valve working life expectancy and practically no carbonised oil deposits on the valves, which can be produced by high compression temperatures.

3.5.5 Safety valve

Fig. 3.42:
Safety valve on screw compressor

The safety valve must blow off the full output of the compressor at 1.1 times the nominal pressure of the compressed air tank.

3.5.6 Intake filter

Dust separator Paper filter insert

Automatic dust extraction

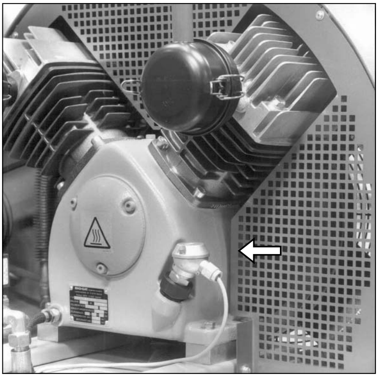
Fig. 3.43:
Intake filter with paper insert

Screw compressors draw in atmospheric air through the air inlet filter inside the compressor cabinet and through the suction filter with paper microfilter cartridge. The inlet filter separates solid impurities such as dust particles from the intake air, minimising wear in the compressor and providing the customer with clean compressed air.

In dusty conditions (e.g., cement works) paper insert filters are used. These have a higher separation rate than standard wet air or foam filtration.

The filter inserts can be cleaned on larger compressors. There is a possibility to monitor the intake filter for pressure differential, allowing soiled filters to be recognised at an early stage.

3.6 Compressor lubricants and coolants



Fir. 3.44:
Oil level check with a dipstick

Compressor oils are standardised to DIN 51506. No HD (high density) oils may be used to lubricate compressors. HD oils tend to emulsify and thus quickly lose their lubricating properties.

Mineral and synthetic oils are allowed. Mineral oils have a useful life of around 2.000 to 3.000 operating hours under normal operating conditions. Synthetic oils can be changed at longer intervals.

The oil level of the compressor must be checked regularly.

The first oil change is made after the running-in period (approx. 300 to 500 operating hours).

Compressors must not be operated with too little oil. Even a short trial run without oil (e.g., to check the direction of rotation) can lead to damage.

The oil filter must be cleaned or replaced each time the oil is changed.

Compressor oils and the condensate from oil-lubricated compressors may not be discharged into the public drains. They must be disposed of in an environmentally acceptable manner. Appropriate oil-water-separators can be used to minimise disposal costs.

Piston compressors

Synthetic-base oils allow compressor running times of up to 8.000 operating hours.

Screw compressors

The lifetime of mineral oils for screw compressors is typically about 3.000 operating hours. Synthetic oils can reach a lifetime of up to 9.000 operating hours.

Synthetic oils have a much higher oxidation and ageing stability than mineral oils. An increased lifetime is the positive result. Also unwished deposits in the oil circuit are prevented. Other advantages of synthetic oils are the lower volatility of synthetic oils, resulting in lower oil carry over and residual oil content, especially at high temperatures and a better viscosity-temperature-behaviour, ensuring good and stable lubrication in a wider temperature range.

When compressed air is used in the food or pharmacy industry and accidental contact with the product is possible, USDA-H1 oils have to be used that fulfil the strict requirements of the food and drugs industry.

4. Control of compressors

The aim of control is to minimise energy consumption and wear and maximise availability.

There are various types of control, depending on the construction type, size and area of application:

- the final pressure (network pressure).
- the inlet pressure.
- the generated volume flow.
- the absorbed power of the compressor motor.
- the climatic conditions of compressor humidity after the compressor stage.

Controlling the final pressure is the most important of all control tasks.

4.1 Pressure definitions

Network pressure p_N [bar $_{op}$]

The network pressure $\mathbf{p_N}$ is the pressure at the compressor outlet behind the outlet valve. This is the pressure in the pipeline network.

The network target pressure p_{Ns} [bar $_{op}$]

The network target pressure $\mathbf{p_{Ns}}$ is the minimum pressure that must be available in the network.

System pressure p_S [bar $_{op}$]

The system pressure $\mathbf{p_S}$ is the pressure inside a screw compressor up to the minimum pressure non-return valve.

Cut-in pressure p_{min} [bar $_{op}$]

The cut-in pressure $\mathbf{p_{min}}$ is the pressure below which the compressor will cut-in.
The cut-in pressure $\mathbf{p_{min}}$ should be at least 0.5 bar above the network target pressure $\mathbf{p_{Ns}}$.

Cut-out pressure p_{max} [bar $_{op}$]

The cut-out pressure $\mathbf{p_{max}}$ is the pressure above which the compressor switches off.
The cutout pressure $\mathbf{p_{max}}$ for piston compressors should be approx. 20% more than the cut-in pressure (e.g., cut-in pressure 8 bar, cut-out pressure 10 bar).
On screw compressors the cut-out pressure $\mathbf{p_{max}}$ should be 0.5 to 1 bar over the cut-in pressure (e.g., cut-in pressure 9 bar, cut-out pressure 10 bar).

4.2 Operating status

The operating status is the current operating mode of a compressor. The operating status is the basis for compressor control.

4.2.1 Stopped (L_0)

The compressor is stopped but ready for operation. If compressed air is needed it switches on automatically.

4.2.2 Idle (L_1)

The compressor is running off load and no air is being compressed (Energy used for compression is saved). If compressed air is needed it switches to operating mode without delay.

Idle operating mode reduces the motor cycles, thus reducing wear.

Various techniques are used to control the idle mode:

Circulation switching

The intake line is connected directly to the pressure line. High pressure losses occur and it is essential that a non-return valve be installed.

Flowback switching

The intake valves of the compressor are open during the compression process. The air does not compress, it flows back to the intake side.

The flowback method is suitable for start-up relief, because the first working stroke is already completely relieved.

Intake line closure

A valve closes the intake line of the compressor. The intake volume is reduced to zero and there is no air available for compression. The pressure losses are low.

Pressure line closure

A valve closes the pressure line of the compressor. The compressed air can not be emitted. No volume flow can occur.

4.2.3 Part-load

The output of the compressor is adjusted to the relevant compressed air requirement. The energy consumption falls slightly if the output is lower. The network pressure $\mathbf{p_N}$ is constant.

There are several methods of varying volume flow. These can also be combined if necessary:

Speed control

Changing the motor speed also varies the output of the compressor. This occurs mainly with engine-driven compressors. With electrically-powered compressors speed control is usually accomplished with the aid of a frequency converter.

The output is economically continuously controlled from 25 - 100 %.

Emergency chamber control (piston compressors only)

By increasing the dead space there is a stronger reverse expansion of the compressed air. If several emergency chambers are opened one after the other the output can be reduced in steps. There are also variations by which an emergency chamber can be continuously expanded.

Flowback control (piston compressors only)

The output of the compressors is reduced by opening the intake valves during the compression stroke. The opening time of the intake valves determines the amount by which compressed volume flow is reduced.

A part-load control of approx. 25 - 100 % of output is possible. When the intake valve is open for the full compression stroke the output drops back to zero.

Proportionale regulation

An adjustable throttle in the intake line reduces the intake volume. When the system pressure drops, the valve opens accordingly, the compressor takes in more air, and the output rises. As soon as system pressure becomes constant the throttle valve closes and the compressor operates in idling mode.

The output varies between 0 - 100 %. The electrical power requirement does not fall below 70 % during this time.

4.2.4 Operating load (L_2)

The compressor delivers its maximum output and consumes the maximum energy.

4.3 Controlling individual compressors

Compressor control has two objectives:
Energy-saving and minimisation of wear.

To meet these objectives, the 4 operating modes of compressors are combined in various control methods. The method used depends on marginal conditions.

4.3.1 Intermittent control

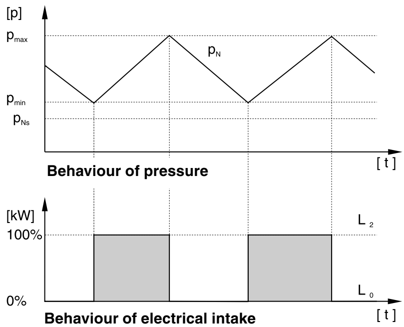


Fig. 4.1 :
Op. diagram of cutout control

With intermittent control a pressure switch or pressure transducer actuates the compressor, depending on network pressure.

The compressor has two operating modes, **Operating mode (L_2)** and **Stopped (L_0)**.

This arrangement has the best energy consumption of all types of control. It is recommended when there is a large compressed air receiver. A large storage volume also reduces the number of motor cycles.

- The network pressure p_N rises to the cut-out pressure p_{max}. The compressor switches to **Stopped (L_0)**.
- The network pressure p_N drops to cut-in pressure p_{min}. The compressor switches **Operating mode (L_2)**.

The intermittend control is the typical control of piston compressors.

4.3.2 Idle mode control

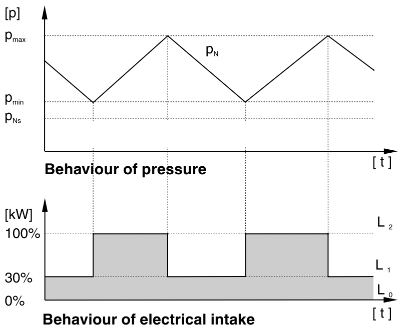


Fig. 4.2 :
Op. diagram of idle mode control

A pressure switch or pressure transducer switches the compressor to operating load or idle mode depending on network pressure.

In **Idle mode (L_1)** the drive motor continues to run, but the compressor does not produce any compressed air. The electrical power demand falls to approx. 30 % of the operating mode requirement.

Continuous operation of the drive minimises the number of motor cycles, which especially with large motors causes increased wear.

Idle operating mode is used in pneumatic systems with relatively small storage volumes, in order not to exceed the maximum switch cycles of the drive motor.

- The system pressure p_N rises to cut-out pressure p_{max}. The compressor switches to **idle mode (L_1)**.
- The network pressure p_N drops to cut-in pressure p_{min}. The compressor switches to **operating mode (L_2)**.

4.3.3 Delayed intermittent control

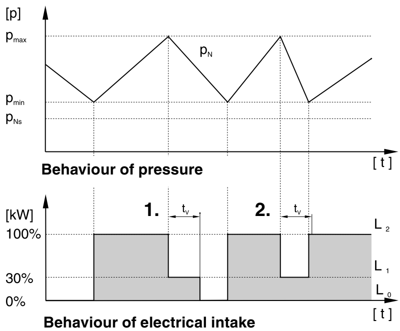


Fig. 4.3
Op. diagram of delayed intermittent control

A pressure switch or pressure transducer works in conjunction with a timer and controls the compressor independently of system pressure.

The compressor goes through the modes of **Operating mode (L_2)**, **Idle mode (L_1)** and **Stopped (L_0)**. The modes are linked with each other via the timer $\mathbf{t_V}$.
The delayed intermittent control combines the benefits of intermittent control and idling control. It is a middle path with lower energy consumption than the idling control method.

The delayed intermittent control operates with two switching variants:

1st Variant

– The system pressure $\mathbf{p_N}$ rises to cut-out pressure $\mathbf{p_{max}}$. The compressor switches to **idle mode (L_1)**.
– The system pressure $\mathbf{p_N}$ has not reached cut-in pressure $\mathbf{p_{min}}$ after expiry of the time $\mathbf{t_V}$.
The compressor switches to **stopped (L_0)**.
– System pressure $\mathbf{p_N}$ drops below cut-in pressure $\mathbf{p_{min}}$. The compressor switches to **operating mode (L_2)**.

2nd Variant

– The system pressure $\mathbf{p_N}$ rises to cut-out pressure pmax. The compressor switches to **idle mode (L_1)**.
– System pressure $\mathbf{p_N}$ reaches cut-in pressure $\mathbf{p_{min}}$ before expiry of the time $\mathbf{t_V}$.

The compressor switches to **operating mode (L_2)**.

There are 2 possibilities to activate the timer $\mathbf{t_V}$:

1. Switching on the compressors **(p_{min})** starts the timer $\mathbf{t_V}$. This provides shorter idling times and therefore lower energy costs as with 2.
2. On reaching the cut-out pressure **(p_{max})** the timer $\mathbf{t_V}$ starts.

4.3.4 Part-load control

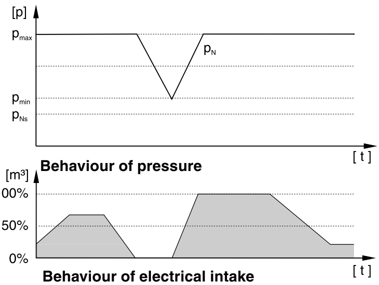


Fig. 4.4
Op. diagram of part-load control

The volume control of the compressor is adjusted to the respective requirement for compressed air.

The network pressure $\mathbf{p_N}$ is largely constant due to the variable output control. The fluctuations of $\mathbf{p_N}$ vary depending on the method of part-load control used.

The part-load control method is used with systems with small storage capacities and/or heavy consumption fluctuations. The number of cycles drops.

4.3.4.1 Proportional regulation

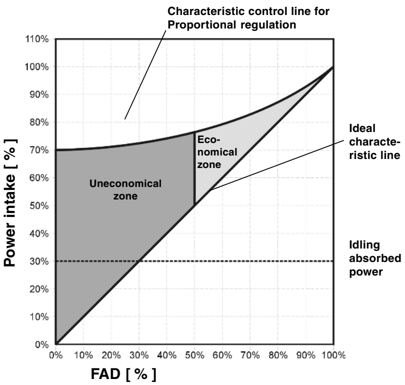


Fig. 4.5a:
Correlation between FAD and absorbed power when using proportional regulation

In addition to the ARS control unit, BOGE offers an optional **proportional regulation** for screw compressors with oil injection cooling. This control intervenes in the processes of the suction control and operates according to the suction throttle principle. It adapts the FAD to the actual compressed air demand.

The proportional regulation from BOGE is set at the factory to a production rate of between 50 and 100% of FAD. If FAD drops to below 50%, the compressor is working uneconomically. Depending on the switching cycle the compressor either switches off or continues on idle mode. With this method on-off-cycles and idling times are minimised and energy is saved.

By adapting the FAD to the actual compressed air demand a constant net pressure is reached as the delivered volume corresponds to the air volume taken from the net. The net pressure can thus be reduced to a lower level and additional energy can be saved by preventing energy consuming over pressure. This method of regulating the FAD is cheaper than the more common frequency control and can be ideally used in a peak load compressor with low changing air demands to use the cost saving potential.

4.3.4.2 Frequency control

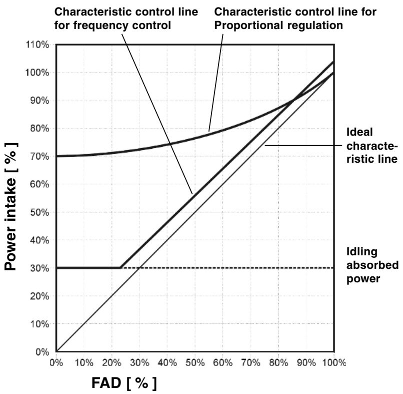


Fig. 4.5b:
Correlation between FAD and absorbed power when using proportional regulation and frequency control

The frequency control allows for a very large FAD regulation from 25% up to 100%. Adaptation to differing compressed air requirements is assured by infinite speed adjustment of the drive motor which is actuated by a frequency converter with simultaneous speed adjustment of the compressor air end.

The frequency converter is designed for soft starts and stops of the drive motor. Even during the switch-on phase, the starting current does not exceed its rated current which, in case of extreme drive performance, can be of some advantage in the supply of energy.

If the output falls below 25%, the compressor is working uneconomically. Depending on the switching cycle the compressor either switches off or continues on idle mode. Owing to the variable adaptation of air delivery, switching operations and idle mode are widely avoided thus enabling the screw compressor to actually work in continuous operation in the most efficient manner possible.

Due to frequency converter occasioned losses, the absorbed power of a frequency controlled compressor is, at full load, approximately between 3% and 5% higher than that of a non-controlled compressor. On the other hand, when adjusting the delivered air quantity by changing the speed of the motor and, consequently, the speed of the compressor air end, the frequency control serves to almost proportionally reduce the absorbed power at the same time. The frequency control proves to be specially advantageous when it comes to smaller flow rates where the proportional regulation by means of suction control appears to work in a rather inefficient manner. Taking into account the two identification lines, it is obvious that at average flow rates between 100% up to approx. 85% the use of the proportional regulation is far more economical, whereas for lower values the frequency control has considerable advantages. Hence follows that the frequency control is ideally suited for use even at low flow rates with considerably varying compressed air demand.

Due to the extensive range of frequency adjustments, it is possible, even in case of little air consumption, to adapt the air delivery to the actual demand in compressed air and to almost completely avoid switching and idle operation.

Under ideal circumstances the flow rate control can be used to constantly maintain the net pressure at a 0.1 bar tolerance. Any excess compression, as is normal with non-controlled compressors due to the difference between switch-on and switch-off pressures, can thus be avoided resulting in energy savings from 6% up to 10% for every 1 bar of high compression.

Owing to its continuous and wide ranging volume flow regulation the frequency control is perfectly suited for strongly varying demands in compressed air consumption with view to both single compressors and peak load compressors in a compressor system.

Depending on prevailing circumstances, all energy savings resulting from the use of a frequency control are immense. Saving potentials can be achieved by elimination of high compression, minimization of idling times and losses due to compressor switch cycles as well as adaptation of the absorbed power to the actually needed and delivered quantity of compressed air.
Also for the earlier mentioned case of proportional regulation by using the suction regulator similar saving potentials can be achieved in cases of lower air flow fluctuations.

4.4. The ARS control concept

BOGE screw-type compressors and supersilenced piston compressors are equipped with the modern **ARS**-control concept (**A**utotronic, **R**atiotronic, **S**upertronic).
The **ARS**-control differs in features and control functions.

ARS is an integrated control and monitoring concept with two objectives:

– Energy-saving and thus a reduction of running costs.
– Extending the lifetime of the compressor by allowing only as little wear as possible.

The ARS-control on screw compressors uses a microcontroller to obtain the cheapest intermittent operation while taking into account the max. permissible motor cycles. Piston compressors only use economical intermittent operation.

All programmed data are stored in an EEPROM storage module that can be electronically written to and erased. The stored information is thus still available in the event of a power failure.

Modular design

The ARS-control comprises standard components that are individually obtainable. Components can also easily be added at a later date. The controls can therefore be ideally configured for the individual requirements of the customer. The controls can be rapidly replaced in the event of failure, thus increasing the availability of the compressors. There is therefore no need for time-consuming and costly examination by specialists.

4.4.1 Autotronic

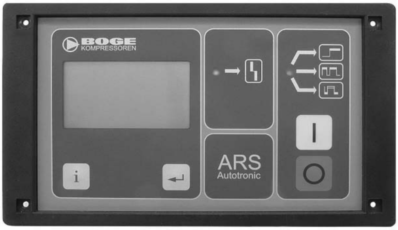


Fig. 4.6 :
The BOGE Autotronic

The **Autotronic** is an intelligent control and monitoring unit for screw and piston compressors. It offers:

- Convenient and well arranged operating panel with 7-segment display with international symbols, all values are displayed directly and precise
- Automatic selection of the best operating mode
- Operating hours counter
- Programmable control
- Protection of important program-parameter by code request
- Accurate pressure sensors instead of pressure switches
- Permanent display of actual compression temperature and pressure
- Digital pressure and temperature display
- Automatic freeze protection for temperatures to –10°C
- Idle mode control for extreme short time rating
- Display of fault and maintenance notifications,
- On-site software update possible
- Optional RS 485 interface

4.4.2 Ratiotronic

Fig. 4.7 :
The BOGE Ratiotronic

The **Ratiotronic** is an extension of the Autotronic for screw or piston compressors. It offers the following additional features:

- Connection to diverse bus systems
- Local or remote operation
- Monitoring of a compressed air preparation component
- Additional system pressure sensor
- Ring buffer for the last 30 faults
- Potential free contacts for fault and maintenance messages and operating mode.

4.4.3 Supertronic

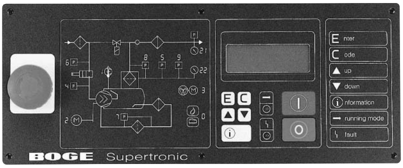


Fig. 4.8 :
The BOGE Supertronic for screw compressors

The **Supertronic** is a complex operating and monitoring unit for screw compressors. In comparison to the other control units it has comprehensive additional functions:

– Well-arranged LCD-display with 4 x 20 characters (digits) and clear text.
– Adjustment of network pressure by keyboard.
– Comprehensive display and monitoring of major operating data.
– Comprehensive compressor monitoring
  Malfunction and warning messages shown on the LCD - display.
– Integrated electronic real-time clock for switching on and off. Operated via keyboard.
– All operating parameters can be adjusted via the keyboard.
– Access to all functions with a few additional keystrokes.

4.5 Control of several compressors

For users of compressed air with high, much fluctuating consumption a single, large compressor is not the best solution. In these cases, a combined compressor system consisting of several compressors is much the better alternative. Greater operating reliability and economy are the aguments in favour of this.

Organisations that are very dependent on compressed air can guarantee their supply at all times by a combined compressor system. If one compressor fails or servicing work is necessary, the other compressors continue the supply.

Several small compressors can be adjusted more easily to compressed air consumption than one large compressor. The idling costs of a large compressor are moreover higher than those of small, stand-by compressors. These facts provide the greater economy.

A combined compressor is operated economically and low on wear by a master control system.

4.5.1 MCS 1 and MCS 2

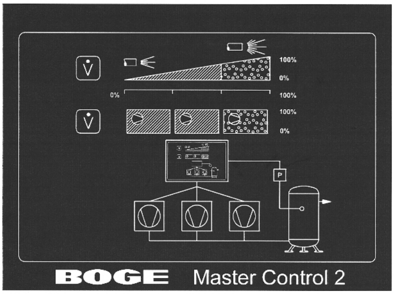


Fig. 4.9:
The BOGE Master Control System 2

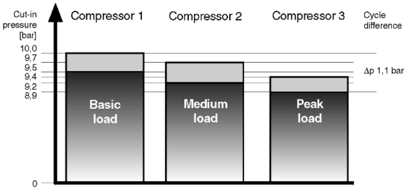


Fig. 4.10:
The circuit diagram of the BOGE MCS 2

MCS 1 controls 2 compressors of the same size as basic load and peak load. The compressors are cyclically changed and switched on and off via their own pressure switches. The control unit offers:

– Cyclic change via a timer.
– Time lag cycling of the compressors by the control unit through pressure graduation.
– Even use of compressors.
– Constant pressure in the pressure range.
– Minimal cycle difference Δp = 0,8 bar

MCS 2 controls up to 3 compressors of the same size as basic load, medium load and peak load. The compressors are cyclically changed and switched on and off via their own pressure switches. The upgrade to 3 compressors and the greater cycle difference is the only difference to the MCS 1. The features are otherwise the same.

– Minimal cycle difference Δp = 1,1 bar

4.5.2 MCS 3

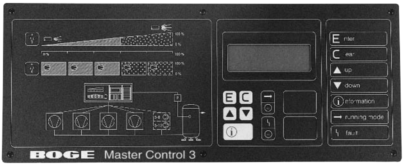


Fig. 4.11:
The BOGE Master Control System 3

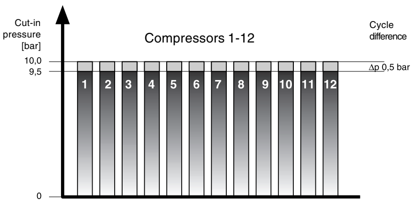


Fig. 4.12:
Circuit diagram of the BOGE MCS 3

MCS 3 controls a maximum of 4, 8, or 12 compressors of the same and/or different size and type in a system. All compressors are controlled by a common pressure sensor on the compressed air receiver.
The **MCS 3** has at 0.5 bar a very small cut-in difference. The individual compressors are not given fixed cut-out and cut-in pressures. All compressors work in the same pressure range (**Δp** = 0.5 bar). The compressors cut-in dynamically according to requirement via set intermediate pressure values. The speed of pressure rise and fall is measured. The compressors switch on and off dynamically.

The control offers:

– Dynamic pressure control by microcontroller in connection with electronic pressure controllers for a minimum cut-in difference of 0.5 bar.
(no over-compression → energy saving)
– Time dependent allotment of compressors in rank stages for shift operation with differing compressed air requirement.
– Individual assignment of individual compressors to load range groups, uniform usage of compressors.
– Adjustable basic load changeover cycle.
– Independent rotation of compressors into the load range groups.
– Time offset allocation of compressors if demanded by the control unit.
– Well arranged LCD-display with 4 x 20 characters and clear text.
– Possibility of checking all inlets and outlets via a test-menu.
– Automatic reverting to pressure switches of individual compressors in the event of voltage loss.
– The individual compressors work independently without the **MCS 3**. They are then controlled from their own pressure switches.

4.5.3 MCS 4

Fig. 4.13:
The BOGE Master Control System 4

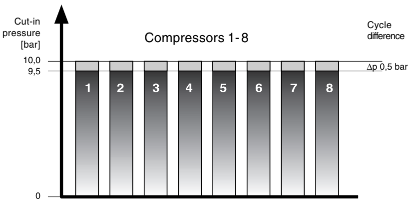


Fig. 4.14:
Op. diagram of the BOGE MCS 4

MCS 4 controls a maximum of 4 or 8 compressors of the same and/or different sizes and types in a system. All compressors are controlled by a common pressure sensor at the compressed air receiver.

The basic load with this control unit is normally covered by the largest compressor or combination of compressors. The smallest compressor takes the peak load. Compressors of the same size change over in providing the basic load.

The **MCS 4** computes compressed air consumption continually from programmed compressor performance data and information from the pressure sensor. It selects the compressor that most closely matches the requirement.

The control offers:

– need-oriented use of the various compressors and compressor combinations.
– ideal use of the benefits of screw and piston compressors.
– minimal cut-in difference of 0.5 bar. (no over-compression → energy-saving)
– three different pressure profiles per day by a timer programme to adapt the control to differing compressed air requirement.
– Time adjusted use of compressors on demand by the control unit.
– Well-organised LCD-display with 2 x 20 characters and clear text output.
– Possibility to check all inlets and outlets via a test menu.
– Automatic switchover to pressure switches in the event of voltage loss.
– The individual compressors operate independently without the **MCS 4**. They are then controlled by their own pressure switches.
– Two potential-free timer contacts for control of additional components.

4.5.4 MCS 5

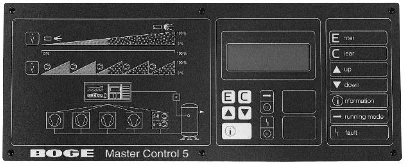


Fig. 4.15:
The BOGE Master Control System 5

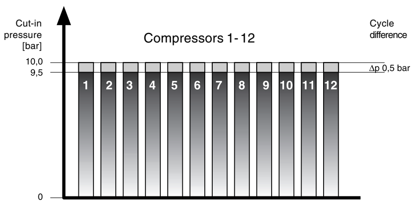


Fig. 4.16:
Op. diagram for the BOGE MCS 5

MCS 5 controls a maximum of 4, 8, or 12 compressors with infinite output control of the same and/or different size and construction in a system. All compressors are controlled by a common pressure sensor on the compressed air receiver. The peak load compressor controls accordingly the requirement of compressed air via its infinite output control.

If the compressed air demand drops this compressor switches off and the medium load compressor takes over via its infinite output control,dependng on the level of priority.

Up to their use of infinite control the **MCS 3** and **MCS 5** are similar.

The control unit offers:

- Adaptation of FAD to the compressed air demand by infinite output control by the peak load compressor.
- Minimal pressure fluctuations in the pneumatic network.
- Dynamic pressure control by microcontroller in conjunction with the electronic pressure control for a minimum cut-in difference of 0.5 bar.
  (no over-compression → energy savings)
- Time-independent allocation of compressors in level of priority for shift operation with differing compressed air demand.
- Individual allocation of compressors in the load range groups with even usage of compressors.
- Adjustable basic load change cycle.
- Independent rotation of compressors in the load range groups.
- Time adjusted use of compressors on demand by the control unit.
- Well arranged LCD-display with 4 x 20 characters and clear text output.
- Possibility to check all inlets and outlets via a test menu.
- Automatic change to pressure switches of individual compressors in the event of voltage loss.
- The individual compressors operate independently without the **MCS 5**. They are then controlled by their own pressure switches.

4.5.5 MCS 6

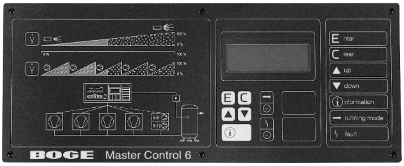


Fig. 4.17:
The BOGE Master Control System 6

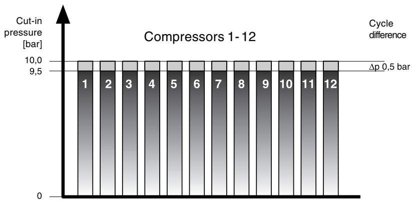


Fig. 4.18:
Op. diagram for the BOGE MCS 6

MCS 6 Controls a maximum of 4, 8, or 12 compressors with speed frequency control of the same, and/or different size and design/type in a system. All compressors in the system are controlled through a common pressure sensor on the compressed air receiver. The peak load compressor controls the compressed air demand via its speed frequency control.

When the compressed air demand falls this compressor switches off and the medium load compressor takes over the control through its speed frequency control.

Apart from the speed frequency control the **MCS 3** and the **MCS 6** systems are similar.

The control system offers:

– Adaptation of the FAD to the compressed air demand through speed frequency control of the peak load compressor.

– Minimum pressure fluctuations in the pneumatic system.

– Dynamic pressure control by microcontroller in conjunction with the electronic pressure controller for a minimum cut-in difference of 0.5 bar.
(no over-compression → energy saving)

– Time-dependent allocation of compressors in priorities for shift operation with differing compressed air demand.

– Individual allocation of individual compressors in the load range groups with even work rates among the compressors.

– Adjustable basic load change cycle.

– Independent rotation of compressors in the load range groups.

– Time adjusted use of compressors on demand by the control unit.

– Well-arranged LCD-display with 4 x 20 characters and clear text output.

– Possibility to check all inlets and outlets via a test menu.

– Automatic switchover to the pressure switches of individual compressors in the event of power failure.

– The individual compressors work independently without the **MCS 6**. They are then controlled by their own pressure switches.

4.5.6 MCS 7

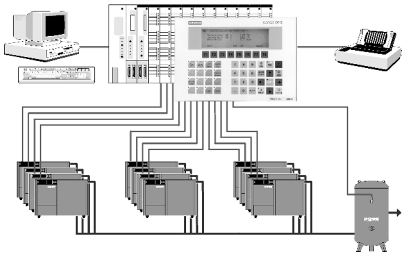


Fig. 4.19:
The BOGE Master Control System 7

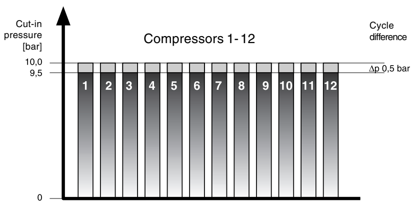


Fig. 4.20:
Op. diagram for the BOGE MCS 7

MCS 7 controls, regulates and monitors a complete pneumatic station with the Siemens-control **S 5 (S7)** and the operator terminal **OP 15**.

The basic features include:

– 8 Compressors.
– 2 Refrigeration compressed air dryers.
– 2 Adsorption dryers.
– 10 Bekomats.
– 2 Potentional-free switch channels for control of additional devices.

The **MCS 7** is available in three versions:

Version 1

Version 1 offers an extended software program of **MCS 3**. It uses pressure-dependent control of up to 8 or 12 compressors of the same and/or different size by priorities and timer programmes.

Version 2

Version 2 offers an extended software program of **MCS 5**. It uses pressure-dependent control of up to 8 or 12 compressors of the same and/or different size with infinite output control.

Version 3

Version 3 offers an extended software program of **MCS 6**. It uses pressure-dependent control of up to 8 or 12 compressors of the same and/or different size with speed frequency control.

In addition to the basic individual software functions the control offers:

– Recording of the operating status of the compressors and additional components of the compressor station.
– Storage of operating, warning and malfunction messages. This makes servicing and repair of the compressor system much simpler.
– Control and monitoring of the compressed air treatment components and the pneumatic system.
– BUS-coupling with Profibus (optional)
This allows connection to a central control facility.
– System visualisation in master control equipment (optional) Comprehensive information can be obtained about the entire compressed supply.

5. Compressed air treatment

5.1 Why treatment ?

Modern production equipment needs compressed air. The many conditions in which it is used range from untreated blowing air to absolutely dry, oil-free and sterile compressed air.

The impurities in our atmosphere are usually invisible to the naked eye. But they can seriously impede the reliable operation of a pneumatic system and consumer devices, and have an adverse effect on the quality of products.

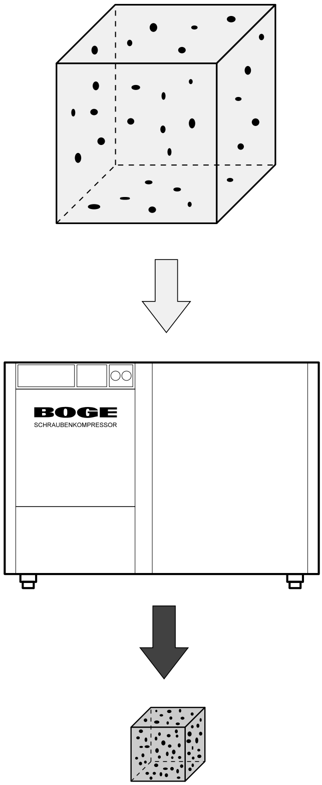


Fig. 5.1 :
Concentration of impurities in the air during compression

1 m^3 of atmospheric air contains many impurities such as

- Up to 180 million particles of dirt.
  These are between 0.01 and 100 µm in size.
- 5 - 40 g/m^3 Water in the form of atmospheric humidity.
- 0.01 to 0.03 mg/m^3 Oil in the form of mineral oil aerosols and unburnt hydrocarbons
- Traces of heavy metals such as lead, cadmium, mercury, iron.

Compressors draw in atmospheric air and the impurities they contain and concentrate them many times. At compression of 10 bar-op (10 bar over-pressure = 11 bar absolute) the concentration of impurities rises by 11 times. In 1 m^3 compressed air there will then be up to 2 billion particles of dirt. Lubrication oil and scuff also passes from the compressor in the compressed air.

Correct treatment of compressed air brings benefits :

- Increased working life of consumer devices.
- Improved and consistent product quality.
- Pneumatic lines free of condensate and rust.
- Fewer malfunctions.
- Pipelines without condensate collectors.
- Lower servicing outlay.
- Lower pressure loss from leakage and flow resistance.
- Lower energy consumption due to lower pressure loss.

5.1.2 Planing information

BOGE recommends the air treatment described on this page for the various applications of compressed air.

Area of application of compressed air	Quality classes DIN ISO 8573-1** Oil	Particle	Water	Compressor	Dust separator *)	Prefilter	Refrigeration dryer	Microfilter	Membrane dryer	Adsorption dryer	prefilter	Active carbon filter	Active carbon absorber	Sterile filter
General air	—	—	—	BOGE screw and piston compressors										
Blowing air	—	—	—											
Sand blasting	—	3	—											
Simple varnishing work	—	3	—											
General works air	5	3	4											
Conveyance air	5	3	4											
Simple spray painting	5	3	4											
Sandblasting with higher quality requirements	5	3	4											
Pneumatic tools	1	1	4											
Control air	1	1	4											
Process control eqpt.	1	1	4											
Spray painting	1	1	4											
Conditioning	1	1	4											
Fluid elements	1	1	4											
Dental laboratories	1	1	4											
Photo laboratories	1	1	4											
Breathing air	1	1	1-3											
Instrument. air	1	1	1-3											
Pneumatics	1	1	1-3											
Spray painting with higher quality requirements	1	1	1-3											
Surface treatment	1	1	1-3											
Medical equipment	1	1	3-4											
Conveyance air with higher quality requirements	1	1	3-4											
Food and luxury food industry	1	1	3-4											
Breweries	1	1	1-3											
Dairies	1	1	1-3											
Pharmaceuticals industry	1	1	1-3											

*) *The dust separator is not required under certain circumstances.*
**) *DIN ISO 8573-1; 1991*

5.1.3 Consequences of poor treatment

If the impurities and water from atmospheric air remain in the compressed air the consequences can be unpleasant. This applies to the pipeline and the consumer devices, and products can also suffer if the quality of compressed air is poor. In some applications the use of compressed air without adequate treatment is dangerous and a health hazard.

Solid matter particles in compressed air

- Wear on pneumatic systems.
  Dust and other particles cause scuff. This effect is increased if the particles combine with lubricating oil or grease to form a grinding paste.
- Particles that are hazardous to health.
- Chemically aggressive particles.

Oil in the compressed air

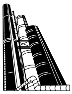
- Old and different oil in the pneumatic system.
  Resinified oil can reduce pipe diameters and cause blockages. This increases flow resistance.
- Oil-free compressed air.
  With pneumatic conveyance, oil can stick to the product being conveyed and thus cause blockages.
  In the food and pharmaceutical industries compressed air **must** be free of oil for health reasons.

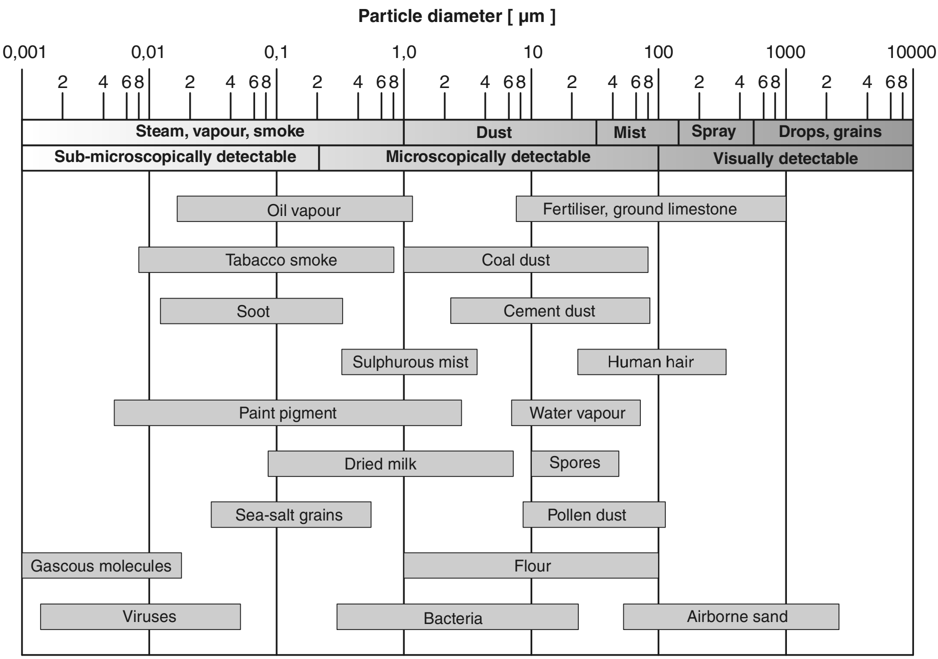
Water in the compressed air

- Corrosion in the pneumatic system.
  Rust forms in the pipelines and operating elements and causes leaks.
- Gaps in lubricant films.
  Gaps in lubricant films lead to mechanical defects.
- Formation of electrical elements.
  Electrical elements can form when some metals come in contact with water.
- Formation of ice in the pneumatic network.
  In low temperatures water in the network can freeze and cause frost damage, reduce pipe diameter and block pipes.

5.1.3 Impurities in the air

In our atmosphere there are particles of dirt that are not visible to the naked eye. This chapter contains a general summary of the type, size and concentration of these particles.

Concentration of particles in atmospheric air		Limits [mg/ m³]	Average values [mg/ m³]
	In the country	5 - 50	15
	In the town	10 - 100	30
	In an industrial area	20 - 500	100
	In large factory plants	50 - 900	200



5.2 Water in the compressed air

5.2.1 Atmospheric humidity

There is always a certain amount of moisture in the atmosphere. This is known as atmospheric humidity and its content varies depending on the time and place. At any temperature a certain volume of air can only contain a maximum quantity of moisture. However, atmospheric air usually contains less than this maximum amount.

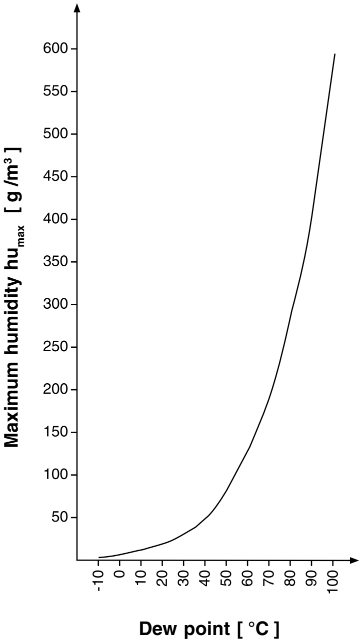


Fig. 5.2 :
Maximum humidity depending on dew point

Maximum humidity hu_{max} [g/m³]

Maximum humidity **hu_{max}** (saturation quantity) means the maximum quantity of moisture that 1 m³ air can hold at a certain temperature. The maximum humidity does not depend on pressure.

Absolute humidity hu [g/m³]

Absolute humidity **hu** means the actual quantity of moisture held by 1 m³ air.

Relative humidity φ [%]

Relative humidity **φ** means the ratio of absolute to maximum humidity.

$$\varphi = \frac{hu}{hu_{max}} \times 100\ \%$$

φ	= relative humidity	[%]
hu	= absolute humidity	[g/m³]
hu_{max}	= maximale humidity	[g/m³]

Since maximum humidity **hu_{max}** depends on the temperature the relative humidity changes with the temperature, even if the absolute humidity remains constant. On cooling to dew point, the relative humidity rises to 100 %.

5.2.2 Dew points

Atmospheric dew point [°C]

Atmospheric dew point means the temperature to which **atmospheric** air (1 bar abs) can be cooled without precipitation.

The atmospheric dew point is of minor importance for pneumatic systems.

Pressure dew point [°C]

The pressure dew point means the temperature to which **compressed** air can be cooled without precipitation of condensate. The pressure dew point depends on the final compression pressure. If pressure drops, the pressure dew point drops with it.

5.2.3 Air moisture content

The following table shows the maximum air humidities at certain dew points:

dew point [°C]	max. humidity [g/ m^3]	dew point [°C]	max. humidity [g/m^3]	dew point [°C]	max. humidity [g/m^3]	dew point [°C]	max. humidity [g/m^3]	dew point [°C]	max. humidity [g/m^3]	dew point [°C]	max. humidity [g/m^3]	dew point [°C]	max. humidity [g/m^3]
+100°	588,208	+76°	248,840	+52°	90,247	+28°	26,970	+4°	6,359	-19°	0,960	-43°	0,083
+99°	569,071	+75°	239,351	+51°	86,173	+27°	25,524	+3°	5,953	-20°	0,880	-44°	0,075
+98°	550,375	+74°	230,142	+50°	82,257	+26°	24,143	+2°	5,570	-21°	0,800	-45°	0,067
+97°	532,125	+73°	221,212	+49°	78,491	+25°	22,830	+1°	5,209	-22°	0,730	-46°	0,060
+96°	514,401	+72°	212,648	+48°	74,871	+24°	21,578	0°	4,868	-23°	0,660	-47°	0,054
+95°	497,209	+71°	204,286	+47°	71,395	+23°	20,386			-24°	0,600	-48°	0,048
+94°	480,394	+70°	196,213	+46°	68,056	+22°	19,252	-1°	4,487	-25°	0,550	-49°	0,043
+93°	464,119	+69°	188,429	+45°	64,848	+21°	18,191	-2°	4,135	-26°	0,510	-50°	0,038
+92°	448,308	+68°	180,855	+44°	61,772	+20°	17,148	-3°	3,889	-27°	0,460	-51°	0,034
+91°	432,885	+67°	173,575	+43°	58,820	+19°	16,172	-4°	3,513	-28°	0,410	-52°	0,030
+90°	417,935	+66°	166,507	+42°	55,989	+18°	15,246	-5°	3,238	-29°	0,370	-53°	0,027
+89°	403,380	+65°	159,654	+41°	53,274	+17°	14,367	-6°	2,984	-30°	0,330	-54°	0,024
+88°	389,225	+64°	153,103	+40°	50,672	+16°	13,531	-7°	2,751	-31°	0,301	-55°	0,021
+87°	375,471	+63°	146,771	+39°	48,181	+15°	12,739	-8°	2,537	-32°	0,271	-56°	0,019
+86°	362,124	+62°	140,659	+38°	45,593	+14°	11,987	-9°	2,339	-33°	0,244	-57°	0,017
+85°	340,186	+61°	134,684	+37°	43,508	+13°	11,276	-10°	2,156	-34°	0,220	-58°	0,015
+84°	336,660	+60°	129,020	+36°	41,322	+12°	10,600	-11°	1,960	-35°	0,198	-59°	0,013
+83°	324,469	+59°	123,495	+35°	39,286	+11°	9,961	-12°	1,800	-36°	0,178	-60°	0,010
+82°	311,616	+58°	118,199	+34°	37,229	+10°	9,356	-13°	1,650	-37°	0,160	-65°	0,00640
+81°	301,186	+57°	113,130	+33°	35,317	+9°	8,784	-14°	1,510	-38°	0,144	-70°	0,00330
+80°	290,017	+56°	108,200	+32°	33,490	+8°	8,234	-15°	1,380	-39°	0,130	-75°	0,00130
+79°	279,278	+55°	103,453	+31°	31,744	+7°	7,732	-16°	1,270	-40°	0,117	-80°	0,00060
+78°	268,806	+54°	98,883	+30°	30,078	+6°	7,246	-17°	1,150	-41°	0,104	-85°	0,00025
+77°	258,827	+53°	94,483	+29°	28,488	+5°	6,790	-18°	1,050	-42°	0,093	-90°	0,00010

5.2.4 Quantity of condensate during compression

Air contains water in the form of moisture. Since air can be compressed and water can not, when air is compressed the water precipitates in the form of condensate. The maximum humidity of the air depends on temperature and volume. It does not depend on quantity.

Atmospheric air can be imagined as a moist sponge. It can take in a certain amount of water when it is relaxed. But if it is squeezed, part of the water runs out. Some of the water will always stay in the sponge regardless of how hard it is squeezed. Compressed air is very similar.

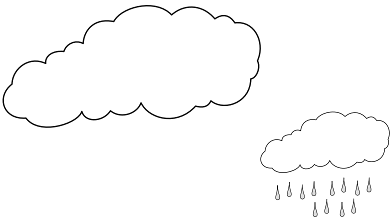
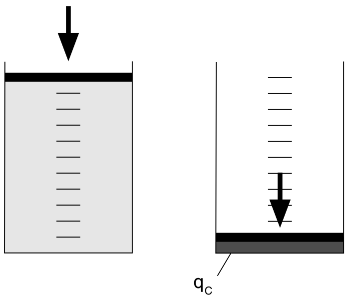


Fig. 5.3 :
A wet sponge being squeezed

The following examples illustrate the quantity of condensate to be expected $\mathbf{q_c}$ when air is compressed. The example assumes a humid Summer day with 35° C and 80 % atmospheric humidity.



V_1 = 6,5 m^3	V_2 = 0,59 m^3
p_1 = 0 bar-op = 1 bar abs	p_2 = 10 bar-op = 11 bar abs
T = 35° C	T = 35° C
φ_1 = 80 %	φ_2 = 100 %
hu_{max} = 39,286 g/m^3	

Fig. 5.4 :
Precipitation of condensate during compression

$$q_c = \frac{V_1 \times hu_{max\,1} \times \varphi_1}{100} - \frac{V_2 \times hu_{max\,1} \times \varphi_2}{100}$$

$$q_c = \frac{6{,}5 \times 39{,}286 \times 80}{100} - \frac{0{,}59 \times 39{,}286 \times 100}{100}$$

$$q_c = \frac{m^3 \times g/m^3 \times \%}{\%} - \frac{m^3 \times g/m^3 \times \%}{\%}$$

$$q_c = 181{,}108\ g$$

q_c	= precipitated condensate	[g]
V_1	= Volume at 0 bar-op	[m^3]
V_2	= Volume at 10 bar-op	[m^3]
$hu_{max\,1}$	= max. humidity at 35° C	[g/m^3]
φ_1	= relative humidity of V_1	[%]
φ_2	= relative humidity of V_2	[%]

Because the water that comes out of the compressed air is the part the air can not store, the humidity φ of the compressed air rises to 100 %.

When compressing 6,5 m^3 air to 10 bar pressure, at a constant temperature 181,108 g water will precipitate in the form of condensate.

5.2.5 Example for calculating quantities of condensate

An example shows the amount of condensate $\mathbf{q_c}$ that actually occurs when air is compressed. It is to be noted that the condensate occurs at several points of the compressor station and at different times.

The task here is to calculate the occurrence of condensate on a screw compressor with an output of $\dot{V}$ = 2720 m³/h and a final compression pressure of p_{op} = 10,5 bar. Connected in series to the compressor are a compressed air tank and a refrigeration compressed air dryer.

Ambient air

p_1 = 1 bar abs
T_1 = 33° C
φ_1 = 80 %
$hu_{max\,1}$ = 35,317 g/m³

Compressor

p_2 = 11,5 bar abs
T_2 = 40° C
φ_2 = 100 %
$hu_{max\,2}$ = 50,672 g/m³

Refrigeration compressed air dryer

p_3 = 11,5 bar abs
T_3 = 3° C
φ_3 = 100 %
$hu_{max\,3}$ = 5,953 g/m³

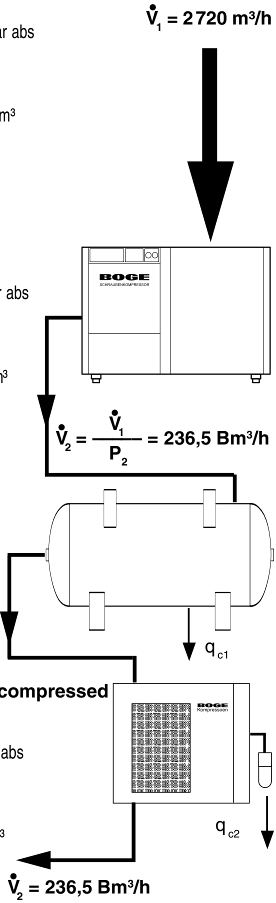


Fig. 5.5 :
Condensate precipitation when compressing with a dryer

The atmospheric air contains a certain amount of water under these conditions:

$$\mathbf{q_w = \dot{V}_1 \times hu_{max\,1} \times \varphi_1 / 100}$$
$$\mathbf{g/h = m^3/h \times g/m^3 \times \%/\%}$$
$$\mathbf{q_w = 2720 \times 35{,}317 \times 80/100}$$
$$\mathbf{q_w = 76849{,}79\ g/h \mathrel{\hat{=}} 76{,}85\ l/h}$$

During the **compression** process, the temperature rises above the pressure dew point of the compressed air, and therefore no moisture will precipitate. In the aftercooler of the compressor the compressed air is cooled down to T_2 = 40°C. The first condensate occurs and is taken with the air into the compressed air receiver. The volume flow calms down and the droplets of water precipitate. A considerable amount of condensate collects there:

$$\mathbf{q_{c1} = q_W - (\dot{V}_2 \times hu_{max\,2} \times \varphi_2 / 100)}$$
$$\mathbf{q_{c1} = 76849{,}79 - (236{,}5 \times 50{,}672 \times 100/100)}$$
$$\mathbf{q_{c1} = 64865{,}86\ g/h \mathrel{\hat{=}} 64{,}87\ l/h}$$

After this the compressed air is cooled down in the **refrigeration compressed air dryer** to a temperature corresponding to a pressure dew point of 3°C. The condensate precipitates in the dryer and is drained off.

$$\mathbf{q_{c2} = (\dot{V}_2 \times hu_{max\,2}) - (\dot{V}_2 \times hu_{max\,3})}$$
$$\mathbf{q_{c2} = (236{,}5 \times 50{,}672) - (236{,}5 \times 5{,}953)}$$
$$\mathbf{q_{c2} = 10576{,}04\ g/h \mathrel{\hat{=}} 10{,}58\ l/h}$$

In addition to the individual flows of condensate, there is also the quantity of condensate that needs to be dealt with by the condensate treatment equipment.

Condensate quantity q_c = q_{c1} + q_{c2}

Condensate quantity q_c = 75441,9 g/h
= 75,4 l/h

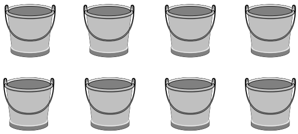


Fig. 5.6 :
Approx. 8 10 l buckets of condensate precipitate in 24 hours

With 3-shift operation working at 100 % efficiency the compressor is running 24 hrs. per day. This means, with the basic assumptions unchanged:

Condensate quantity q_{cD} = 1810605,6 g/D
= 1810,6 l/D

The following quantity of condensate will then occur in one year:

Condensate quantity q_{cY} = 659 060 438 g/Y
= 659 060 l/Y

5.2.6 Quantity of condensate on a humid Summer day

The quality of compressed air must always remain the same **if the surrounding conditions are unchanged**. i.e., the pressure dew point of the compressed air must be 3°C even on a humid Summer day with an air temperature of 40°C and 90 % atmospheric humidity.

FAD $\dot{V}_1$	=	2 720 m³/h
Inlet pressure p1	=	1 bar abs
Inlet temperature T1	=	40° C
Relative humidity φ_1	=	90 %
Pressure dew point T_3	=	2° C

Under these conditions the quality of compressed air remains constant but the quantity of condensate is much higher.

Condensate quantity q_c = 122,6 l/h

With 3-shift operation working at 100 % efficiency the compressor is running 24 hrs. per day. This means, with the basic assumptions unchanged:

Condensate quantity q_{cD} = 2 943,3 l/D

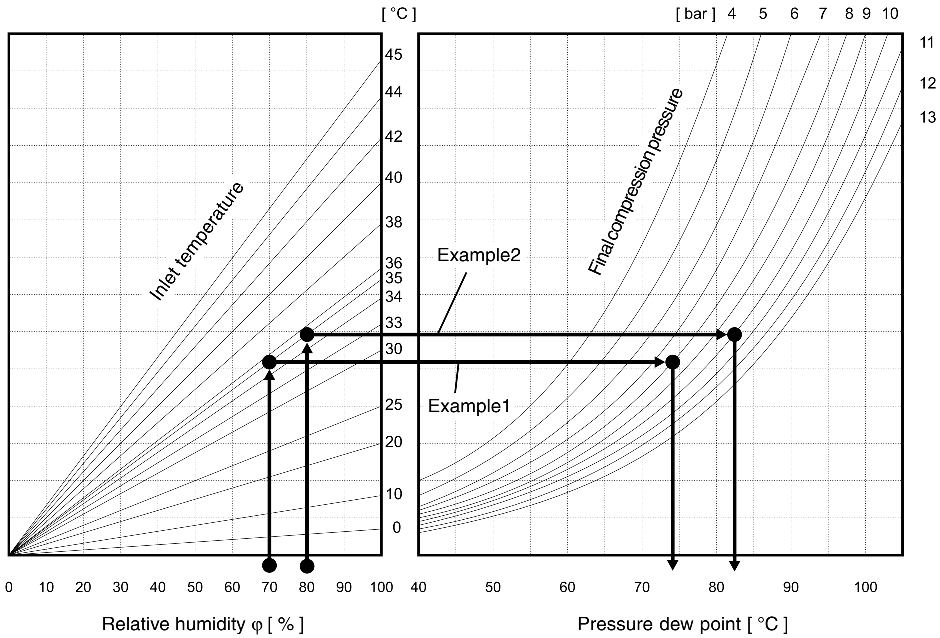
The following quantity of condensate will then occur in one year:

Condensate quantity q_{cY} = 1 071 358 l/Y

5.2.7 Determining the pressure dew point

The pressure dew point means the temperature to which the **compressed** air can be cooled without condensate precipitating. The pressure dew point depends on the final compression pressure. If the pressure drops, the pressure dew point drops with it.

The following diagrams are used to determine the pressure dew point of the compressed air after compression:



Example 1

Intake air

- relative atmospheric humidity φ = 70 %
- inlet temperature T = 35°C

Compressed air

- Final compression pressure p_{abs} = 8 bar

⇒ **The pressure dew point is approx. 73°C**

Example 2

Intake air

- relative atmospheric humidity φ = 80 %
- inlet temperature T = 35°C

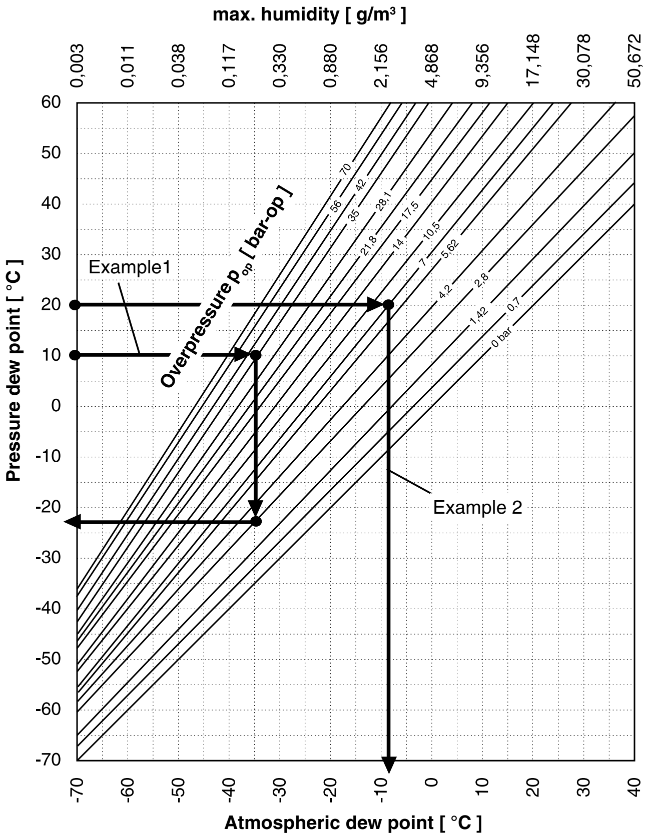
Compressed air

- Final compression pressure p_{abs} = 10 bar

⇒ **The pressure dew point is approx. 82°C**

5.2.8 Pressure dew point after removal of pressure

When compressed air relaxes (pressure released) the pressure dew point drops. The following table is used to determine the new pressure dew point and atmospheric dew point after relaxation:



Example 1

Compressed air

– p_{abs} = 35 bar air pressure

– Pressure dew point 10°C

relaxed compressed air

– p_{abs} = 4 bar air pressure

⇒ **The new pressure dew point is approx. –22°C**

Example 2

Compressed air

– p_{abs} = 7 bar air pressure

– Pressure dew point 20°C

relaxed compressed air

– atmospheric air pressure p_{abs} = 1 bar

⇒ **The atmospheric dew point is approx. –8°C**

5.3 Compressed air quality

5.3.1 Quality classes defined in DIN ISO 8573-1

The quality classes for compressed air defined in DIN ISO 8573-1 make it easier for the user to set his requirements and choose the equipment he needs to treat the air. The norm is based on maker's specifications giving defined limits for their equipment and machinery pertaining to purity of compressed air.

The DIN ISO 8573-1 norm defines quality classes for compressed air according to:

Oil content
Definition of the residual quantity of aerosols and hydrocarbons contained in the compressed air.

Particle size and density
Definition of the size and concentration of solid matter particles that may remain in the compressed air.

Pressure dew point
Definition of the temperature to which the **compressed** air can be cooled without condensation of the moisture it contains. The pressure dew point changes with the air pressure.

Impurities and quality classes according to DIN ISO 8573-1

Class	Solid impurities				Humidity	Max. oil content mg/m³
	Max. particle size per m³				Max. pressure dewpoint	
	Max. particle size in µm					
	< = 0,1	0,1 < d < = 0,5	0,5 < d < = 1,0	1,0 < d < = 5,0		
0	As specified by user					
1	A/R	100	1	0	< = –70°C	< = 0,01 mg/m³
2	A/R	100.000	1.000	10	< = –40°C	< = 0,1 mg/m³
3	A/R	A/R	10.000	500	< = –20°C	< = 1 mg/m³
4	A/R	A/R	A/R	1.000	< = + 3°C	< = 5 mg/m³
5	A/R	A/R	A/R	20.000	< = + 7°C	–
6	–	–	–	–	< = +10°C	–
	Classes 6 and 7 are defined according to the maximum particle size and maximum density. Class 6: d < = 5 µm and density < = 5 mg/m³ Class 7: d < = 40 µm and density < = 10 mg/m³				Classes 7 to 9 are defined according to their liquid water content. Class 7: C_w < = 5 mg/m³ Class 8: 0.5 g/m³ < C_w < = 5 mg/m³ Class 9: 5 g/m³ < C_w < = 10 mg/m³	

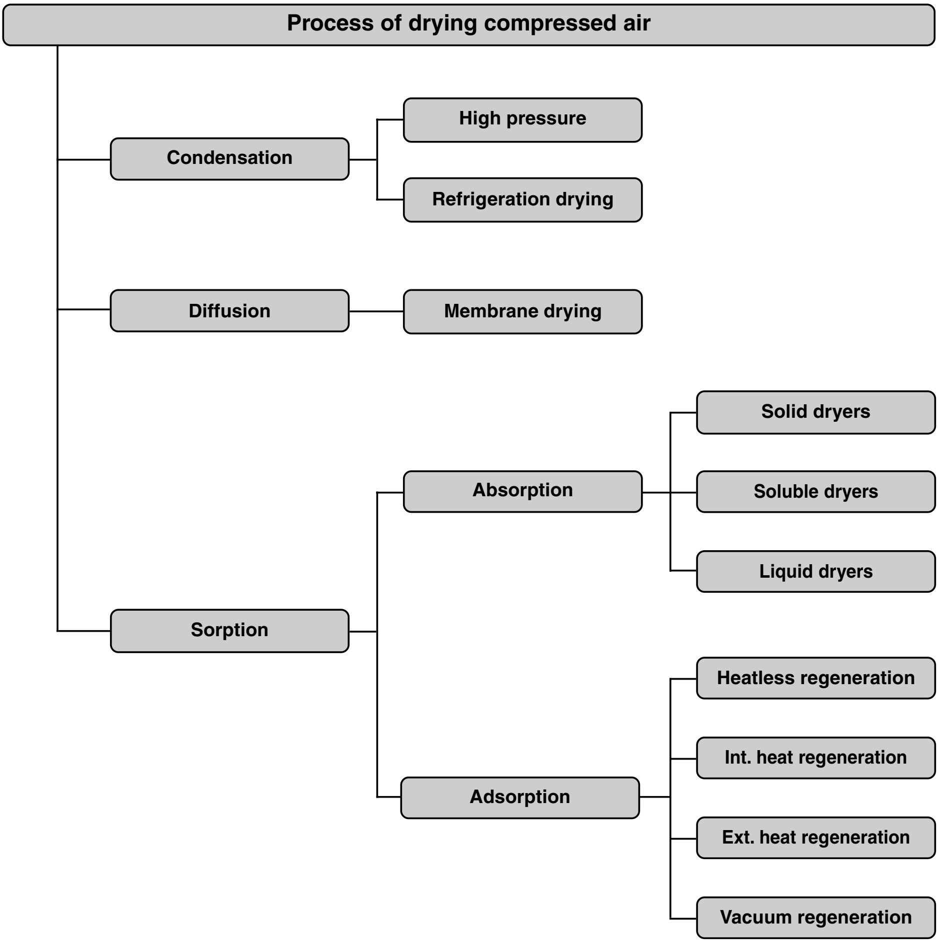
5.4 Methods of drying

The summary presents the methods of drying compressed air according to their principle of operation. A distinction is always made between condensation, sorption and diffusion.

Condensation is the separation of water by going below the dew point.

Sorption is drying by removal of moisture.

Diffusion is drying by molecular transfer.



5.4.1 Operating conditions

The through-flow rate of a dryer refers to the intake rate of air during compression by a compressor according to PN2 CPTC2, ISO 1217 (DIN 1945 Part 1).

– Intake pressure $p = 0\ bar_{op} \triangleq 1\ bar_{abs}$
– Intake temperature $T_0 = 293\ K \triangleq 20°\ C$

Drying equipment is designed according to DIN ISO 7183 for certain operating conditions. The performance data given for the equipment is only correct under these conditions:

– Operating pressure $p = 7\ bar_{op} \triangleq 8\ bar_{abs}$
– Ambient temperature $t_A = 298\ K \triangleq 25°\ C$
– Entry temperature $t_{En} = 308\ K \triangleq 35°\ C$

If a dryer is used under different operating conditions, appropriate conversion factors must be taken into account. These factors differ in the various drying processes.

Example for the layout of a refrigeration compressed air dryer

Conversion factors for operating conditions and ambient temperature:

Op. pressure p [bar_{op}]	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**14**	**16**
Factor f	**0,62**	**0,72**	**0,81**	**0,89**	**0,94**	**1**	**1,04**	**1,06**	**1,09**	**1,1**	**1,12**	**1,15**	**1,17**

Ambient temperature t_A [°C]	**25**	**30**	**35**	**40**	**43**
Factor t	**1,00**	**0,92**	**0,85**	**0,79**	**0,75**

A BOGE refrigeration compressed air dryer, model D8, has a through-flow rate **R** of 45 m³/h. It is operated at an average ambient temperature of $t_A = 40°\ C$ and an operating pressure of $p = 10\ bar_{op}$.

$R = 45\ m^3/h$

$p = 10\ bar_{op} \Rightarrow f = 1,09$

$t_A = 40°\ C \Rightarrow t = 0,79$

$$\mathbf{R_{Ad} = R \times f \times t}$$
$$\mathbf{R_{Ad} = 45\ m^3/h \times 1,09 \times 0,79}$$
$$\mathbf{R_{Ad} = 38,75\ m^3/h}$$

R_{Ad} = Adjusted through-flow rate [m³/h]

R = Through-flow rate [m³/h]

f = Conversion factor for $p = 10\ bar_{op}$

t = Conversion factor for $t_A = 40°\ C$

With changed operating conditions the dryer has through-flow rate of 38,75 m³/h.

5.4.2 Condensation by high pressure

Pressure dew point [°C]	Operating pressure [bar_{op}]	Volume flow [m^3/h]	Entry temperature [°C]
approx. – 70°C	Depends on compressor	Depends on compressor	–

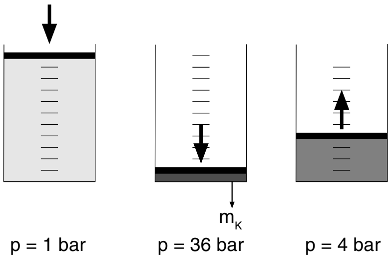


Fig. 5.7 :
Over-compression with subsequent relaxation

The air is compressed far beyond the necessary pressure, and afterwards cooled and reduced to operating pressure.

Operating principle

With rising pressure and thus reduced volume the air is able to hold less water. During pre-compression at high pressure a large amount of condensate precipitates. The absolute humidity of the air goes down. If the compressed air is now relaxed, the relative humidity drops and with it the dew point.

Example:

Compressed air is pre-compressed to 36 bar. The dew point is 10°C. The condensate precipitates. After reducing to 4 bar the compressed air has a new pressure dew point of approx. – 18°C.

Features

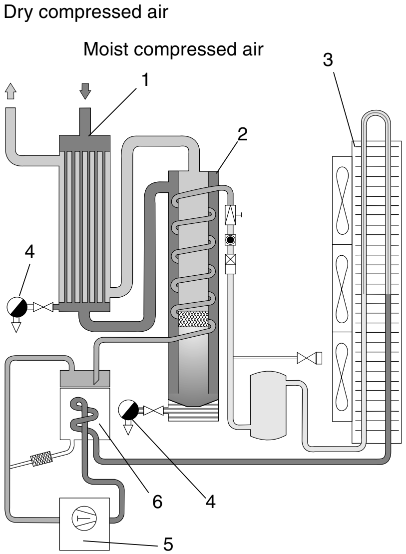
– Simple process with continuous volume flow.
– No expensive refrigeration and drying equipment.
– Only economical for small output quantities.
– Very high energy consumption.

5.4.3 Condensation by refrigeration drying

Pressure dew point [°C]	Operating pressure [bar_{op}]	Through-flow rate [m^3/h]	Entry temperature [°C]
to −2°C	to 210	11-35 000	to +50°C

When the temperature falls, air loses its ability to hold water. To reduce the moisture content, compressed air can be cooled down in a refrigeration dryer.

Refrigeration drying is a process by which compressed air is cooled down by a dryer in a heat exchanger. The moisture contained in the air precipitates in the form of condensate. The quantity of condensate that precipitates rises with the difference between the entry and exit temperature of the compressed air.



1 = Air/Air heat exchanger
2 = Air/refrigerant heat exchanger
3 = Refrigerant/air heat exchanger
4 = Condensate drain
5 = Refrigerant compressor
6 = Vapour outlet

Fig. 5.8 :
Op. diagram of a refrigeration compressed air dryer

Operating principle

Refrigeration drying runs in two phases. This is done to improve effectiveness and to obtain maximum use of the refrigerant.

1st Phase

Inside an air/air heat exchanger the compressed air already cooled by the refrigeration dryer cools new air flowing in. 70 % of the moisture contained in the air precipitates here in the form of condensate.

2nd Phase

The compressed air flows through a refrigerant/air heat exchanger and cools down almost to freezing point. The precipitated condensate is directed off before re-heating in the first cooling phase.

Features:

- Highly economical.
  Refrigeration drying is the most economical process in approx. 90 % of all applications.
- Separation of impurities.
  Almost 100 % of all solid particles and water droplets larger than 3 μm are separated.
- Lower pressure loss in the dryer.
  The pressure loss **Δp** from the dryer is approx. 0,2 bar.

5.4.4 Diffusion by membrane drying

Pressure dew point [°C]	Operating pressure [bar_{op}]	Through-flow rate [m^3/h]	Entry temperature [°C]
0 to – 20°C	5 -12,5	11 - 130	2° to 60°C

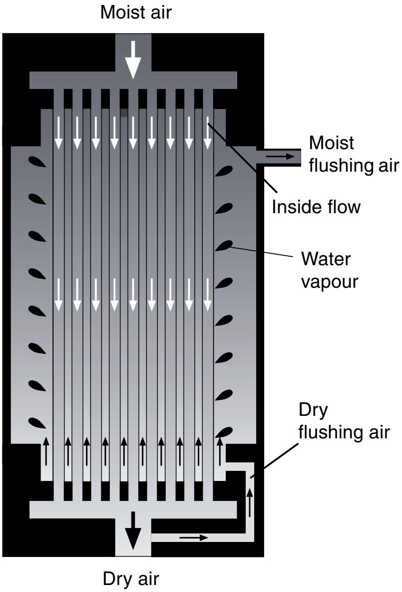


Fig. 5.9 :
The principle of a membrane dryer

The principle of the membrane dryer is based on the fact that water penetrates a specially coated hollow fibre 20 000 times faster than air.

The membrane dryer consists of a bundle of thousands of coated hollow fibre membranes. These hollow fibres are made of a solid, temperature and pressure-resistant plastic. Their inside surface is coated with an ultra-thin (less than the length of a light wave) coating of a second plastic. The hollow fibres (membranes) are installed in a pipe where the inner channel of the fibres is open at the end.

Operating principle

The moist compressed air flows through the inside of the hollow fibres (internal flow). The moisture contained in the air penetrates through the layer of coating on the hollow fibres towards the outside. To do this a concentration gradient of moisture is required between the inside and outside of the hollow fibres.

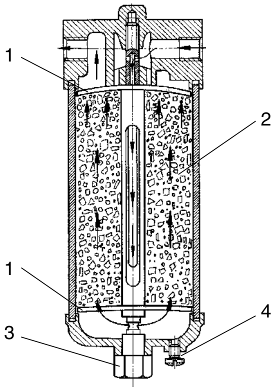
A quantity of air for flushing is taken from the main volume flow of the compressors and relaxed (decompressed). Since the maximum air humidity depends on volume, the relative air humidity drops. The flushing air becomes very dry. The flushing air flows around the hollow fibres and provides the necessary concentration gradient of moisture. The flushing air can escape unfiltered into the open and is lost for the system. No liquid water is created with this method. The water is removed in vapour form.

Features

- Low level of particles in the air.
  A filter must always be connected upstream of the membrane dryer, in order to filter out particles up to a size of 0.01 µm. If installed directly downstream of the compressor, the filter should be connected to dust separator.
- Low pressure loss in the dryer.
  The pressure loss **Δp** from the dryer is max. 0.2 bar.
- Compact construction.
  The dryer can be installed as a component of the pipeline.
- No servicing.
  There are no moving parts in the dryer.
- No precipitation of condensate during drying
- No additional energy costs.
- Silent.
- No fluorocarbons.
- No moving parts.
- No motor.

5.4.5 Sorption by Absorption

Pressure dew point [°C]	Operating pressure [bar_{op}]	Through-flow rate [m^3/h]	Entry temperature [°C]
Depends on entry temperature	–	–	to 30°C



1 = Screen
2 = Solid drying agent
3 = Cover
4 = Condensate drain

Fig. 5.10 :
Absorption dryer with solid drying agent

With absorption drying the moisture is separated by a chemical reaction with a hygroscopic drying agent. Since the absorption properties of the drying agent diminish over time, periodic renewal is necessary.

There are 3 different types of drying agent. The soluble agents liquify with increased absorption. The solid and liquid agents react with the moisture without changing their aggregate status.

Drying agent		
Solid	**Soluble**	**Liquid**
Dehydrated chalk	Lithium chloride	Sulphuric acid
oversour magnesium salt	Calcium chloride	Phosphoric acid
		Glycerine
		Triethylene glycol

Operating principle

During absorption the compressed air flows upwards through a drying middle bed. During this it gives up some of its moisture to the drying agent. A drain directs the condensate to a floor tank. The pressure dew point is lowered by 8 – 12 %.

Example

Compressed air enters a dryer operating with calcium chloride at a temperature of + 30°C. The pressure dew point achieved here is between 18 and 22°C.

Features

– Low entry temperature.
  High temperatures soften the drying agent and bake it together.
– Very corrosive drying agents.
  The dried compressed air can take drying agent with it into the pneumatic system. This can cause considerable damage.
– No input of outside energy.

Due to its properties, absorption drying has only become established in fringe applications of pneumatic engineering. One example of this is its use for compressed treatment air in laboratories.

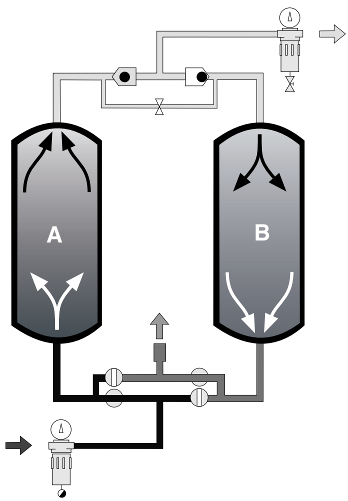
5.4.6 Sorption by Adsorption

Drying compressed air by adsorption is a purely physical process. The moisture is bound to the drying agent by force of adhesion (unbalanced molecular attraction).The moisture stays on the inner and outer surfaces of the adsorption material without a chemical reaction taking place.

The adsorption material has an open porous structure and a large inner surface. The most common adsorption materials are aluminium oxide, silicagel, active carbon and molecular screens. Different adsorption materials are used for the various regeneration processes.

Adsorption material	Properties of Adsorption material *)			
	Obtainable press. dew point [°C]	Entry temperature [°C]	Regeneration temperature [°C]	Surface [m2/g]
Silicagel (SiO_2), raw	– 50	+ 50	120 - 180	500 - 800
Silicagel (SiO_2), spherical	– 50	+ 50	120 - 180	200 - 300
Activated Aluminium oxide (Al_2O_3)	– 60	+ 40	175 - 315	230 - 380
Molecular screens (Na, AlO_2, SiO_2)	– 90	+ 140	200 - 350	750 - 800

*) *The properties of the adsorption material change with the pressure and temperature of the gas to be dried*



Operating principle

During the drying process the moist compressed air flows through an adsorption tank. The moisture is bound, which dries the compressed air. This process generates heat. The adsorption material must be regenerated when the adhesive forces are balanced by water deposits. This means that the water must be removed from the adsorption material. For this reason there must be two parallel drying tanks with continual operation. The active tank **A** dries the compressed air, while the inactive tank **B** regenerates without pressure.

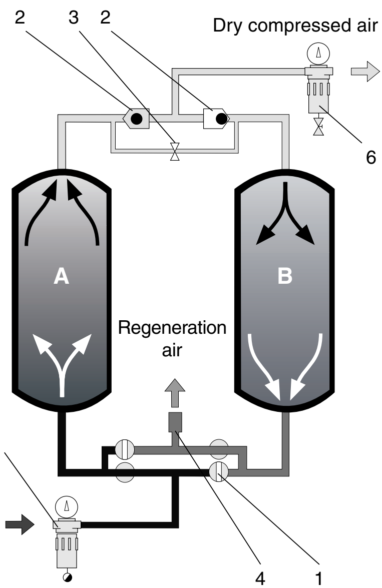
The following processes are mainly used to regenerate the adsorption material:

– heatless regeneration
– internal hot regeneration
– external hot regeneration
– vacuum regeneration

5.4.6.1 Heatless regeneration

Pressure dew point [°C]	Operating pressure [bar_{op}]	Through-flow rate [m³/h]	Entry temperature [°C]
to – 70°C	4 - 16	4 - 5600	to + 60°C

Fig. 5.11 :
Adsorption material after 5 min. drying time



1 = Valve block
2 = Non-return valve
3 = Perforated cover
4 = Outlet valve
5 = Pre-filter
6 = After-filter

Fig. 5.12 :
Op. diagram of an adsorption dryer, cold regeneration

With heatless regeneration the drying and regeneration time is around 5 min. For this reason the moisture only deposits on the outer surface of the drying agent.

Cold regeneration adsorption dryers operate according to the pressure alternation process. With this method the desorption (regeneration) takes place without additional input of heat. A part of the dried volume flow is branched off. This part-flow relaxes to a pressure of just over 1 bar and is thus extremely dry. This dry air then flows through the regeneration drying tank **B**. In this process it takes on the moisture stored in the drying agent and directs it out into the open through an outlet valve.

Features

- **Economical on smaller systems with low volume flows.**
- Simple dryer construction.
- Can be used at high ambient temperatures.
- Low volume of drying agent.
  Drying and regeneration times approx. 5 min.
- High operating costs.
  The regeneration air is taken from the pneumatic system and can not be used further.
- Regeneration without outside energy.
- The percentage ratio of regeneration air to the output of the compressor falls with a higher final compression pressure.

Final comp. pressure [bar_{abs}]	Ratio of regeneration air [%] Press. dew point –25° to –40°C	Press. dew point –40° to –100°C
5	25,83	27,14
7	17,22	18,1
10	11,49	12,07
15	7,39	7,77
20	5,46	5,47

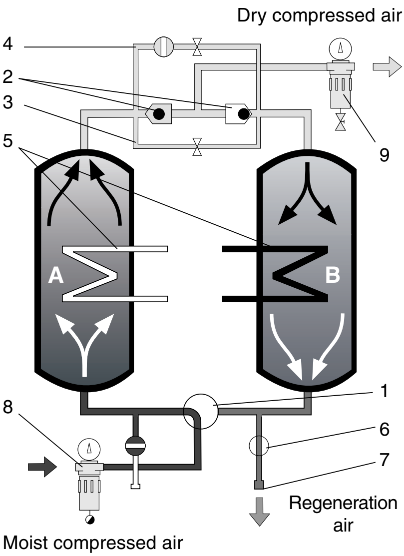
These values are physically fixed and it is not possible to go below them. They are taken from the correlation between air moisture and compressed air pressure relief. Specific calculations for the regeneration air demand have to be done for every application.

- Prefiltration of intake air.
  A prefilter removes most of the oil, water droplets and particles of dirt.
- Postfiltration of dried compressed air.
  Drying material taken with the compressed air from the drying tank must be filtered out.

5.4.6.2 Internal heat regeneration

Pressure dew point [°C]	Operating pressure [bar_{op}]	Through-flow rate [m^3/h]	Entry temperature [°C]
to – 40°C	2 - 16	200 - 5600	to + 50°C

Fig. 5.13 :
Adsorption material after 6 - 8 hrs drying time



1 = Valve block
2 = Non-return valve
3 = Diversion line with perf. cover **1st Phase**
4 = Diversion line with perf. cover **2nd Phase**
5 = Heating
6 = Stop valve
7 = Outlet valve
8 = Prefilter
9 = After-filter

Fig. 5.14 :
Op. diagram of an adsorption dryer, internal hot regeneration

With heat regeneration the drying and regeneration times are around 6 - 8 hrs. During the long drying time the moisture deposits on the inner and outer surfaces of the adsorption material. To reverse this process heat must be brought from outside. If the regeneration temperature of the drying material is exceeded by heat from outside, the surface energies that occur outweigh the adhesive forces in the drying material and the water evaporates. A small flow of regeneration air drains off the moisture.

The regeneration temperature depends on the pressure dew point of the regeneration air. The lower it is, the lower the regeneration temperature of the dryer.

With **internal regeneration** the heat is transmitted directly from a heater in the drying tank to the adsorption material. This happens in two phases:

1st Phase

Drying tank **B** is slowly heated by the internal heating to the necessary regeneration temperature. If the regeneration temperature is exceeded, the moisture releases itself from the adsorption material. Approx. 2 - 3 % of the dried flow of compressed air from the compressor relaxes and at slight pressure is directed through a diversion line through drying tank **B**. This flow of regeneration air absorbs the moisture and directs it out into the open through an outlet valve.

2nd Phase

In a cooling phase the operating pressure drops back to the temperature of the drying bed. A second diversion line opens for this purpose. Approx. 5 % of the compressor FAD is directed through drying tank **B**. The internal heating is no longer operating at this point.

Features

- **Economical with high volume flows.**
- Simple dryer construction.
- Little dried compressed air is required to regenerate the dryer.
- Prefiltration of intake air.
  A pre-filter removes most of the oil, water droplets and dirt particles from the compressed air.
- Postfiltration of dried compressed air.
  Drying materials taken with the compressed air from the drying tank must be filtered out of the compressed air.

5.4.6.3 External heat regeneration

Pressure dew point [°C]	Operating pressure [bar_{op}]	Though flow rate [m^3/h]	Entry temperature [°C]
to – 40°C	2 - 16	500 - 15 000	to + 50°C

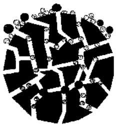
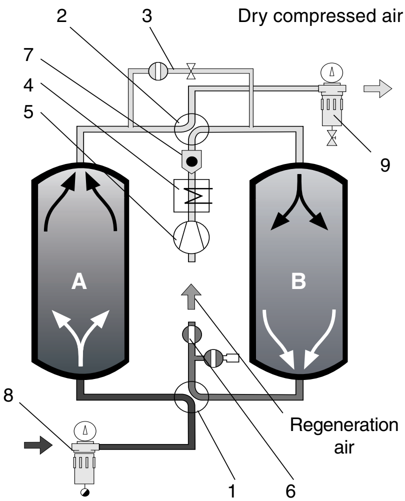


Fig. 5.15 :
Adsorption material after 6 - 8 hrs drying time



1 = Bottom valve block
2 = Top valve block
3 = Diversion line with perf. cover **3rd Phase**
4 = Heating register
5 = Fan
6 = Stop valve
7 = Non-return valve
8 = Prefilter
9 = After-filter

Fig. 5.16 :
Op. diagram of an adsorption dryer, external hot regeneration

With heat regeneration the drying and regeneration times are around 6 - 8 hrs. During the long drying time the moisture deposits on the inner and outer surfaces of the adsorption material. To reverse this process heat must be brought from outside. If the regeneration temperature of the drying material is exceeded by heat from outside, the surface energies that occur outweigh the adhesive forces in the drying material and the water evaporates. A small flow of regeneration air drains off the moisture.

The regeneration temperature depends on the pressure dew point of the regeneration air. The lower it is, the lower the regeneration temperature of the dryer.

With **external regeneration** air is drawn in from the atmosphere by a fan and heated in a heating register. This happens in three phases:

1st Phase

The drying tank **B** is slowly heated to the necessary regeneration temperature by the flow of hot air. Once the regeneration temperature is reached, the water releases itself from the Adsorption material. The fan continues to supply hot regeneration air through drying tank **B**. This flow of regeneration air takes on the moisture and transports it into the open through an outlet valve.

2nd Phase

In a cooling phase the operating temperature drops back to the temperature of drying tank **B**. For this purpose the heating register of the fan is switched off and cold air from the atmosphere is directed through the drying tank.

3rd Phase

At the end of cooling, dry, relaxed compressed air flows from the compressor and through the drying tank, in order that the atmospheric does not bring moisture back into the dryer.

Features

- **Economical with high volume flows**
- Higher regeneration temperatures allow a lower pressure dew point.
- Low additional consumption of compressed air.
  Only a small part of the regeneration air is taken from the pneumatic system.
- Prefiltration of inlet air.
  A pre-filter removes most of the oil, water droplets and dirt particles from the compressed air.
- Postfiltration of dried compressed air.
  Drying materials taken with the compressed air from the drying tank must be filtered out of the compressed air.

5.4.6.4 Vacuum regeneration

Pressure dew point [°C]	Operating pressure [bar_{op}]	Through-flow rate [m^3/h]	Entry temperature [°C]
to – 80°C	4 - 16 bar	400 - 7400	to + 40°C

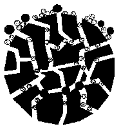
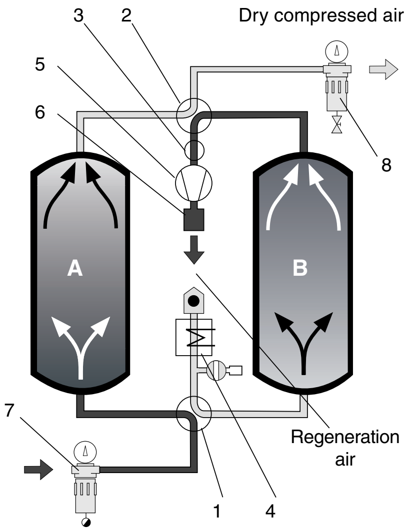


Fig. 5.17 :
Adsorption material after 6 - 8 hrs drying time



1 = Bottom valve block
2 = Top valve block
3 = Non-return valve
4 = Heating register
5 = Fan
6 = Silencer
7 = Prefilter
8 = After-filter

Fig. 5.18 :
Op. diagram of an adsorption dryer,
Vacuum regeneration

Vacuum regeneration is a variation of external hot regeneration. As with hot regeneration the drying and regeneration times are around 6 - 8 hrs. During the long drying time the moisture deposits on the inner and outer surfaces of the adsorption material. To reverse this process heat must be brought from outside. If the regeneration temperature of the drying material is exceeded by heat from outside, the surface energies that occur outweigh the adhesive forces in the drying material and the water evaporates. A small flow of regeneration air drains off the moisture.

The regeneration temperature depends on the pressure dew point of the regeneration air. The lower it is, the lower the regeneration temperature of the dryer.

With **vacuum regeneration** atmospheric air is drawn with a partial vacuum into the drying tank. This flow of air heats externally. Vacuum regeneration occurs in two phases.

1st Phase

A vacuum pump draws in air from the outside. This flow of air is heated by a heating register and drawn through the drying tank. Once the regeneration temperature is reached, the water releases itself from the Adsorption material. The flow of regeneration air takes on the moisture and transports it into the open through an outlet valve.

2nd Phase

In a cooling phase the operating temperature drops back to the temperature of the drying tank. For this purpose the heating register is switched off and cold air from the atmosphere is directed through the drying tank.

Features

- **Economical with high volume flows**
- No additional compressed air consumption.
  No compressed air is taken from the system for regeneration.
- Long utility time of drying agent.
  Thermal stress on the drying agent is low.
- Energy savings through lower regeneration temperature.
- Prefiltration of inlet air.
  A pre-filter removes most of the oil, water droplets and dirt particles from the compressed air.
- Postfiltration of dried compressed air.
  Drying materials taken with the compressed air from the drying tank must be filtered out of the compressed air.

5.4.7 Arrangement of the refrigeration compressed air dryer

There are two basic possibilities for arranging a refrigeration compressed air dryer in a compressor station. It can either be installed **before** or **after** the compressed air receiver. No general decision on this matter is possible because there are advantages and disadvantages with both constellations.

5.4.7.1 Dryer before the compressed air receiver

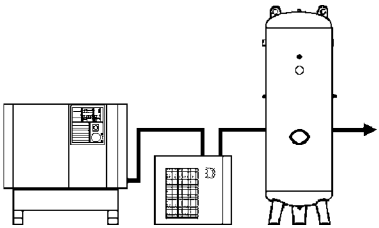


Fig. 5.19 :
Dryer before the compressed air receiver

Advantages:

– Dried air in the compressed air receiver.
  No precipitation of condensate in the compressed air receiver.
– Consistent compressed air quality.
  Even with abrupt, heavy withdrawal of compressed air the pressure dew point of the compressed air remains unchanged.

Disadvantages:

– Large size dryer.
  The dryer must be designed for the entire effective output of installed compressor. The dryer is often over-dimensioned if consumption is low.
– Drying of pulsating compressed air.
  As a result of their construction, piston compressors in particular deliver a pulsating flow of air. This puts stress on the dryer.
– High entry temperature of compressed air.
  The compressed air comes directly from the after-cooler of the compressor.
– Drying of a partial air flow is not possible.
– Large quantity of condensate.
  The entire quantity of condensate precipitates in the dryer.
– With systems containing several compressors, each compressor must have a dryer connected.

Conclusion

Installing a dryer before the compressed air receiver can seldom be recommended. However, an arrangement of this type makes good sense when sudden peaks of requirement are anticipated and the quality of the compressed air must not deteriorate.

5.4.7.2 Dryer behind the compressed air receiver

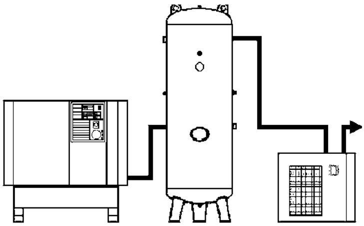


Fig. 5.20 :
Dryer behind the compressed air receiver

Advantages:

– Favourable dryer size.
  The dryer can be sized according to the actual consumption of compressed air, or for a partial flow of compressed air that needs to be dried.
– Drying of a non-turbulent volume flow.
– Low compressed air entry temperature.
  The compressed air has the opportunity to cool down further in the compressed air receiver.
– Low quantities of condensate.
  The droplets of condensate collect in the compressed air receiver and do not burden the rest of the system.

Disadvantages:

– Condensate in the compressed air receiver.
  Moisture in the compressed air receiver leads to corrosion.
– Overload of the Dryer.
  The dryer is overloaded if there is any abrupt, heavy withdrawal of compressed air. The pressure dew point of the compressed air rises.

Conclusion

In most cases, BOGE recommends installing the dryer behind the compressed air receiver. The argument of economy is in favour of it. A smaller dryer can normally be chosen. Its efficiency rate is better.

5.5 Compressed air filters

5.5.1 Basic terminology of filters

To assess and operate filters it is first necessary to define and explain certain sizes and factors.

5.5.1.1 Filter separation rate η [%]

The filter separation rate **η** gives the difference in concentration of impurities **before** and **after** the filter. It is also called the efficiency rate. The filter separation rate **η** is a measure of the efficiency of the filter. The minimum grain size [µm] that the filter can separate must always be specified.

$$\eta = 100 - \left[\frac{C_2}{C_1} \times 100 \right]$$

C_1 = Concentration of impurities **before** the filter.

C_2 = Concentration of impurities **after** the filter.

η = Filter separation rate [%]

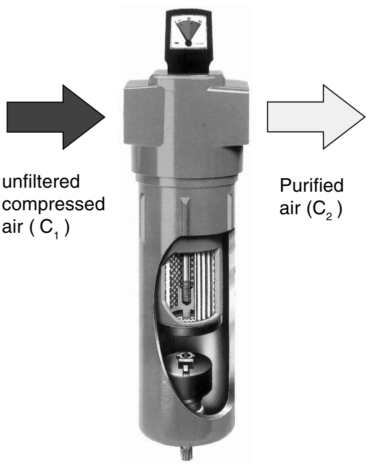


Fig. 5.21 :
BOGE Pre-filter, series V
η = 99,99 % relevant to 3 µm

The concentration is usually measured in proportion of weight per unit of volume [g/m³] of compressed air. With weaker concentrations, the concentration is usually defined by counting the particles per unit of volume [Z/cm³]. The particles per unit of volume method is nearly always used to measure the efficiency of high-performance filters. Measuring the weight proportion per unit of volume with sufficient accuracy would involve a disproportionate amount of effort.

Example

Compressed air contains an impurity particle concentration of C_1 = 30 mg/m³ prior to filtering. The purified air after the filter still has an impurity particle concentration of C_2 = 0,003 mg/m³ with particle sizes over 3 µm.

$$\eta = 100 - \left[\frac{0{,}003}{30} \times 100 \right]$$

$$\eta = 99{,}99\ \%$$

The filter has a separation rate in per cent of 99,99 % relative to 3 µm.

5.5.1.2 Pressure drop Δp

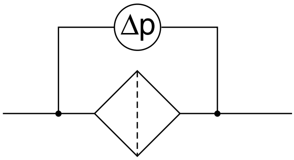


Fig. 5.22 :
General filter with Δ p measuring device

The pressure drop **Δp** is the difference in pressure **before** and **after** the filter caused by flow. The pressure drop **Δp** in the filter grows with time as particles of dust and dirt are collected in the filter element.

- **Δp_0** is the pressure drop for new filter elements.
  It is between 0,02 and 0,2 bar, depending on the type of filter.
- The economically acceptable limit for pressure drop **Δp** is around 0,6 bar.

Devices that measure the pressure difference are installed in most filters.

If the pressure drop **Δp** exceeds the limit, either the filter must be cleaned or the element replaced.

5.5.1.3 Operating pressure

The maximum volume flow of a filter always refers to the norm pressure p_{op} = 7 bar. When pressure changes the maximum through-flow rate of the filter also changes. The change to the through-flow rate can be easily calculated with the aid of appropriate conversion factors **f**.

Pressure [bar_{op}]	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**
Factor f	**0,25**	**0,38**	**0,5**	**0,65**	**0,75**	**0,88**	**1**	**1,13**	**1,25**	**1,38**	**1,5**	**1,63**	**1,75**	**1,88**	**2**

Example

A BOGE pre-filter V50 with a nominal performance of 300 m^3/h at a norm pressure of p_{op} = 7 bar is to be operated at p_{op} = 10 bar.

R_7 = 300 m^3/h

p_{op} = 10 bar $\Rightarrow$ f = 1,38

$$\mathbf{R_{10} = R_7 \times f}$$
$$\mathbf{R_{10} = 300\ m^3/h \times 1{,}38}$$
$$\mathbf{R_{10} = 414\ m^3/h}$$

R_{10} = effective rate at p_{op} = 10 bar [m^3/h]

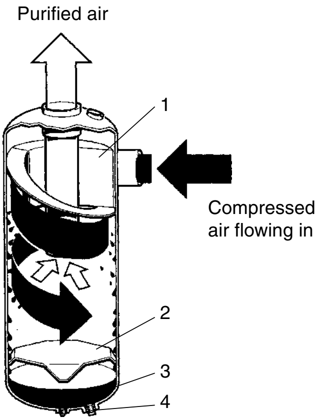
R_7 = effective rate at p_{op} = 7 bar [m^3/h]

f = Conversion factor for p_{op} = 10 bar

At a pressure of p_{op} = 10 bar the filter has an effective nominal performance of 414 m^3/h.

5.5.2 Dust separators

Pressure difference Δp [bar]	Separation rate [%]	Particle size [µm]	Residual oil content [mg/m³]
> 0,05 bar	95 %	> 50 µm	not influenced



1 = Vortex insert
2 = Baffle plate
3 = Collection chamber
4 = Condensate drain

Fig. 5.23 :
Cyclone separator

After coming out of the compressor, the compressed air contains water in the form of steam and also droplets of condensate. These droplets are formed during the compression process because the air is no longer able to accommodate it when its volume is reduced.

This water normally deposits in the storage tank as the compressed air become more inert. From there the condensate is drained off.

Operating principle

The dust separator operates according to the principle of mass inertia. It consists of a vortex cartridge and a catch pan. The vortex cartridge is designed to put the compressed air into rotary movement. Solid and liquid components in the air are forced against the inside walls of the pan by their own mass inertia. This causes heavy particles of dirt and water to separate. These separated impurities flow past a baffle plate into the collection chamber. The baffle plate also prevents the flow of air from taking the separated liquid with it.

The condensate can be drained off automatically or by hand from the collection chamber and properly disposed of or treated.

Features

- Almost complete separation of water droplets.
- Heavy particles of dust and dirt filtered out.
- The filtering capacity of the dust separator depends on the flow speed of the air. The higher the flow speed, the more efficient the filter is. Of course, when the flow speed increases the pressure loss in the separator rises also.

Areas of application

- No compressed air receiver in the pipeline system.
- Large distances between the compressor and the receiver. If the receiver is a long way from the compressor, then it makes sense to install a cyclone separator directly downstream of the compressor. It prevents unnecessary „transport of water" in the pipeline.
- Rising lines between the compressed air receiver and the compressor. The line between the compressor and the receiver goes vertically upwards. When the compressor is idle the condensate flows back into the compressor. In this case it makes sense to install a cyclone separator directly downstream of the compressor.

5.5.3 Pre-filters

Pressure difference Δp [bar]	Separation rate [%]	Particle size [µm]	Residual oil content [mg/m³]
> 0,03 bar	99,99 %	> 3 µm	not influ- enced

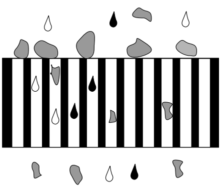


Fig. 5.24:
Filtrations mechanism of surface filters

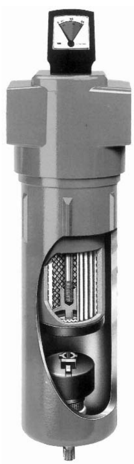


Fig. 5.25:
BOGE Pre-filter, Series V

Pre-filters filter solid impurities to a particle size of approx. 3 µm out of the compressed air but filter out very little oil and moisture. Pre-filters take the load off high performance filters and dryers when the air is very dusty. Finer filters can be dispensed with if the demands on the quality of the compressed air are low.

Operating principle

Pre-filters operate according to the principle of superficial filtration. They have a purely sifting effect. The pore size determines the size of particle that can be filtered out. The impurities remain only on the outer surface of the filter elements. Standard materials for filter elements are:

– Sintered bronze.
– Highly molecular polyethylene.
– Sintered ceramics.
– Bronze or brass wire (coarse filtration).
– Pleated cellulose paper inserts.

Air flows through the filter from the **outside** towards the **inside**. An opposite direction of flow would allow the separated particles to build up inside the filter element. The growing collection of solid matter would block the effective area of the filter.

Features

– Re-usability.
  Because the separated particles are only collected on the surface of the pre-filter element it is possible to clean the element.

5.5.4 Microfilters

Pressure difference Δp [bar]	Separation rate [%]	Particle size [µm]	Residual oil content [mg/m³]
> 0,1 bar	99,9999 %	> 0,01 µm	> 0,01

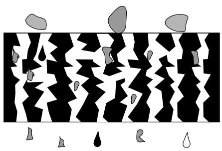


Fig. 5.26 :
Filtrations mechanism of deep-bed filters

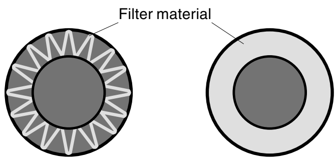


Fig. 5.27 :
Pleated and wound filter material

Fig. 5.28 :
BOGE-Microfilter, Series F

Microfilters are used when high quality compressed air is required. They deliver technically oil-free compressed air. Microfilters reduce the residual oil content of compressed air to 0,01 mg/m³. They filter out dirt particles with a separation rate of 99,9999 % relative to 0,01 µm.

Operating principle

Microfilters, also called high-performance filters, are deep-bed filters. The filter the water and oil condensate phase from the compressed air in the form of fine and ultra-fine droplets.

The deep-bed filter is a fibrous web consisting of a tangle of very fine individual fibres. The fibres are randomly intertwined and thus form a porous structure. Between the fibres there is a labyrinth-like system of passages and openings. This system has flow channels that are sometimes much larger than the particles than the particles to be filtered. Filtration occurs along the entire path travelled by the compressed air on its way through the filter element.

Microfilters work with pleated filter material. This enlarges the effective filter surface by approx. 1/3 in comparison to wound filters. The pressure drop **Δp** is also considerably reduced. There are several advantages in this:

– Increased through-flow rate.
– Lower energy loss.
– Longer service life.

Air passes through deep-bed filters from the **inside** towards the **outside**. The liquid phase from oil and water deposits on the fibrous web when passing through the filter. The flow of air then drives the condensate and growing droplets further on through the filter towards the outside. A part of the condensate leaves the filter element again as a result of this effect. Following the laws of gravity, the condensate collects in the collection chamber of the filter.

The working lives of the filters are longer because the condensate filtered out is no longer a burden to the element with this direction of flow.

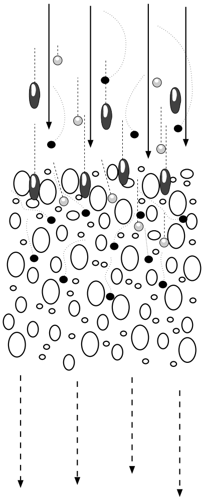


Fig. 5.29
Mechanisms of deep-bed filtration

Filter mechanisms

Three different mechanisms operate together to separate fine particles from the air.

– Direct contact.
  Larger particles and droplets hit the fibres of the filter materials directly and are bound.
– Impact.
  Particles and droplets hit the randomly arranged fibres of the filter material. There they bounce off, are directed out of the path of flow and are absorbed by the next fibre.
– Diffusion.
  Small and ultra-fine particles coalesce in the field of flow and following Brown's law of molecular movement come together to form ever-growing particles. These particles are then filtered out.

Borosilicate fibre in the form of fibreglass layers is the most widespread material in high performance filters. It is used as a material for deep-bed filters. The following are also used:

– Metallic fibres.
– Synthetic fibres.

Features

– Separation of oil in the liquid phase.
  Hydrocarbons are found in two aggregate conditions in compressed air:
  – in gaseous form as oil gas.
  – liquid in the form of droplets.
  A high-performance filter removes almost 100% of the oil droplets. The oil gas can not be filtered out.
– Low operating temperatures.
  The efficiency of the filter drops when the operating temperature rises. Some of the oil droplets vaporise and go through the filter. With a rise in temperature from +20° to +30°C, 5 times as much oil passes through the filter.
– Recyclable.
  The materials used are chosen with ecological aspects in mind.

5.5.5 Active carbon filters

Pressure difference Δp [bar]	Separation rate [%]	Particle size [µm]	Residual oil content [mg/m³]
> 0,02 bar	99,9999	0,01	> 0,005

Fig. 5.30 :
BOGE-Filter combination, Series AF
An active carbon filter with microfilter connected in series

After passing through high-performance filters and dryers, the technically oil-free compressed air still contains hydrocarbons and diverse odorous and taste substances.

There are many applications of pneumatics where these residues would lead to disruptions of production, adverse quality and unpleasant smells.

An active carbon filter removes the hydrocarbon vapours from the compressed air. The residual oil-content can be reduced to 0.005 mg/m³. The quality of the compressed air is better than that demanded for breathing air. The condensated droplets of oil are already removed by the series-connected filter (BOGE-Microfilter Series F).

Operating principle

The filtration of compressed air by absorption is a purely physical process. The hydrocarbons are bound to the active carbons by powers of adhesion (uneven molecular attraction). Chemical compounding does not take place in this process.

The dried and pre-filtered compressed air is directed through a pleated active carbon filter element. The appearance of this filter element is similar to that of the microfilter. As with the microfilter, the compressed air is directed through the filter element from the inside towards the outside.

Features

– Pre-filtration.
  An active carbon filter must always be connected upstream from a high-performance filter and a dryer. Unfiltered compressed air destroys the adsorbant and reduces the filtration effect.
– No Regeneration.
  The active carbon filling can not be regenerated. It must be replaced, depending on the degree of saturation.
– Working life.
  The filter element of an active carbon filter must be replaced after approx. 300 - 400 hours of operation.

Areas of application

– Food and luxury food industry.
– Pharmaceuticals industry.
– Chemicals industry.
– Surface treatment.
– Medical equipment.

5.5.6 Active carbon adsorbers

Pressure difference Δp [bar]	Separation rate [%]	Particle size [µm]	Residual oil content [mg/m³]
> 0,1 bar	–	–	> 0,003

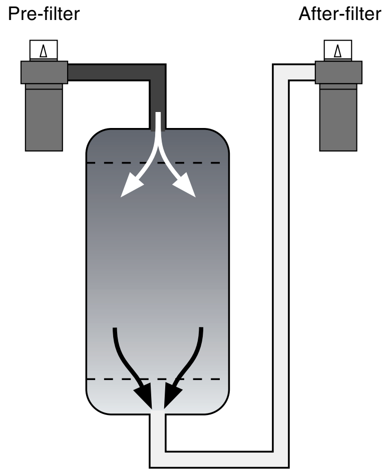


Fig. 5.31 :
Op. plan of a BOGE active carbon adsorber Type DC

After passing through high-performance filters and dryers, the technically oil-free compressed air still contains hydrocarbons and diverse odorous and taste substances. There are many applications of pneumatics where these residues would lead to disruptions of production, adverse quality and unpleasant smells.

An active carbon adsorber removes the hydrocarbon vapours from the compressed air. The residual oil-content can be reduced to 0.003 mg/m³. The quality of the compressed air is better than that required for breathing air. The condensated droplets of oil are already removed by the series-connected filter (BOGE-Microfilter Series F).

Operating principle

The filtration of compressed air by adsorption is a purely physical process. The hydrocarbons are bound to the active carbons by powers of adhesion (uneven molecular attraction). Chemical compounding does not take place in this process.

The dried and filtered compressed air is directed through a diffusor into the loosely piled active carbon bed. The diffusor distributes the compressed air evenly over the entire bed. This allows long contact times and ideal use of the adsorption material. After the adsorber bed the compressed air passes through emission collector and leaves the active carbon adsorber.

Features

– Pre-filtration.
  An active carbon filter must always be connected upstream from a high-performance filter and a dryer. Unfiltered compressed air destroys the adsorbant and reduces the filtration effect.

– After-filtration.
  For safety reasons a high performance filter should be connected downstream from the adsorber. The compressed air take very fine particles of carbon dust (smaller than 1 µm) from the active carbon bed with it.

– No Regeneration.
  The active carbon filling can not be regenerated. It must be replaced, depending on the degree of saturation.

– Long working life.
  The active carbon filling must only be replaced after 8000 - 10000 hours of operation.

Areas of application

– As for active carbon filters.

5.5.7 Sterile filters

Pressure difference Δp [bar]	Separation rate [%]	Particle size [µm]	Residual oil content [mg/m^3]
> 0,09 bar	99,9999	0,01	–

Fig. 5.31 :
BOGE Sterile filter, Series ST

Living organisms such as bacteria, bacteriophages and viruses are a big health problem in many areas. Sterile filters create 100 % sterile and germ-free compressed air.

Operating principle

The pre-purified flow of air is directed from outside towards the inside through the filter element. The filter element is composed of two filter stages. The pre-filter retains microorganisms up to a size of 1 µm. The second filter stage consists of a chemically and biologically neutral, three-dimensional microfibre web made of borosilicate. The remaining organisms are filtered out here. The filter elements are fixed in place by a stainless steel cage.

The filters can be cleaned and sterilised up to 100 times. They are steamed for this purpose. In this process, hot steam of up to +200°C flows through the filter. The steam can be sent through the filter from both sides. Sterilisation by other media is also possible.

– Hot water
– Hot air
– Gas (ethylene oxide, formaldehyde)
– H_2O_2

Features

– Stainless steel material.
All metal parts of the filter are made of high-alloy stainless steel. Stainless steel offers microorganisms no nutritive substratum and can neither corrode nor rot.
– Resistent.
The filter medium is inactive and resistent to chemicals and high temperatures. Bacteria can not grow on or through it.
– Short sterile contact distances.
A sterile filter should be installed directly on the end consumer device.

Areas of application

– Food and luxury food industry.
– Pharmaceuticals industry.
– Chemicals industry.
– Packing industry.
– Medical equipment.

6. Disposal of condensate

6.1 Condensate

Condensate consists primarily of the water contained in the air drawn into the compressor and which forms during compression. The **Condensate** also contains many **impurities**.

– Mineral oil aerosols and unburnt hydrocarbons from the air.
– Particles of dust and dirt of the most varied kinds from the air.
– Cooling and lubricating oil from the compressor.
– Rust, scuff, pieces of sealing material and weld from the pipeline.

Condensate is highly contaminated because of its high content of harmful substances, and for this reason it must be disposed of responsibly. The mineral oils in the condensate are hard to biodegrade and are detrimental to oxygen enrichment and material disintegration in sewage works. This reduces the efficiency of the entire water treatment effort. The consequences are a hazard to nature and human health.

Distinctions must be made between condensate from different pneumatic systems. The condensate has different properties, depending on environmental conditions and the compressor. For example :

– Oil lubricated compressor systems.
  On compressors of this type the oil washes a part of the aggressive and solid matter out of the air in the compression chamber. The result of this is that oil-lubricated systems normally produce condensate that has a pH-value in the neutral range.
– Oil-free compressor systems.
  Most of the harmful substances in oil-free systems are discharged with the condensate. This is why the condensate has an acidic pH-value. pH-values between 4 and 5 are not uncommon.

The consistency of the condensates also changes with marginal conditions. Most condensates are as fluid as water. But pasteous condensates can occur in exceptional cases.

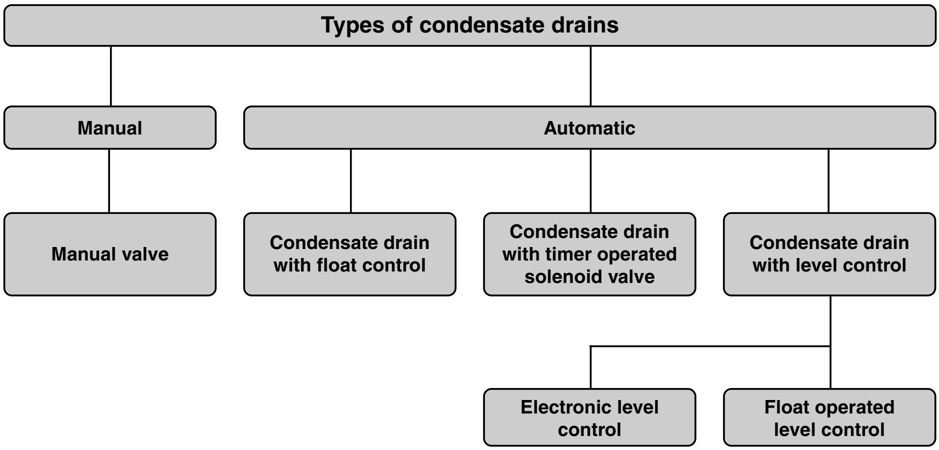
6.2 Condensate drains

Everywhere condensate occurs in a pneumatic system it also has to be drained. If it is not, the flow of air takes it with it, and it enters the pipeline.

The fact that condensate collection tanks are under pressure makes condensate drains costly. The condensate must be drained off under control to unnecessary pressure loss.

It should also be taken into account that condensate does not occur on a continuous basis. The quantity of condensate changes with the temperature and moisture of air drawn in by the compressor.

The summary shows the various construction types according to their method of operation.



When selecting condensate drains, regardless of the construction type, the condensate itself and the marginal conditions must always be taken into consideration. Special applications require special forms of condensate drain :

– very aggressive condensates.
– pasteous condensates.
– explosion danger areas.
– low pressure and partial vacuum networks.
– high and very high pressure networks.

Condensate drains can not be used without heating in sub-zero temperatures. The water component of the condensate will freeze.

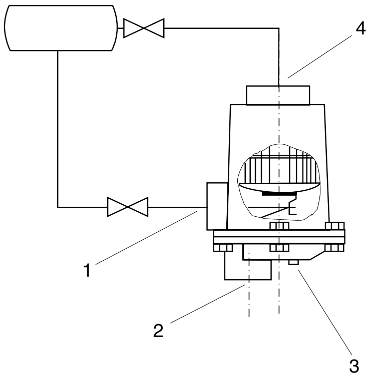
6.2.1 Condensate drains with manual valves

The condensate collects in an appropriate tank (vessel). The servicing or operating staff must check the level of the collection tank at regular intervals. If necessary, the condensate must be drained off with the aid of a valve fitted to the bottom of the tank.

Features

- Simple, inexpensive construction.
- No electricity connection required.
- No alarm function.
- Regular checks necessary.
  The condensate must be drained at regular intervals.

6.2.2 Condensate drains with float control



1 = Inlet line
2 = Outlet line
3 = Drain plug
4 = Vent

Fig. 6.1 :
Condensate drain with float control

Inside the condensate tank there is a float which controls an outlet valve at the bottom of the tank by means of a lever. If the level in the tank rises above a certain level, the outlet valve is opened. Excess pressure in the system forces the condensate out. If the level in the tank falls below the minimum, the valve closes automatically before compressed air can escape.

The condensate is now separated from the compressed air and can now be sent by pipe to the treatment equipment.

Features

- Simple, inexpensive design.
- No electricity connection required.
  Ideal for use in explosion danger areas.
- No blowing off of compressed air.
- Susceptible to malfunctions.
  The moving parts of the system can solidify, stick or corrode through direct contact with condensate.
- Regular servicing required.
  As a result of its susceptibility to malfunctions, it does require regular servicing.
- No external alarm signal.
- Inflexible.
  Float valves must be specially adapted for the condition of the condensate.

6.2.3 Condensate drains with timer operated solenoid valves

The condensate is collected in an appropriate tank. At fixed, regular intervals (1.5 to 30 min.) a solenoid valve with timer opens the drain at the bottom of the tank. After an opening time of 0.4 to 10 s the valve closes again. The condensate is forced out of the drain by system pressure.

The drain valve is connected condensate disposal facility by pipes.

Fig. 6.2 :
Solenoid operated drain valve

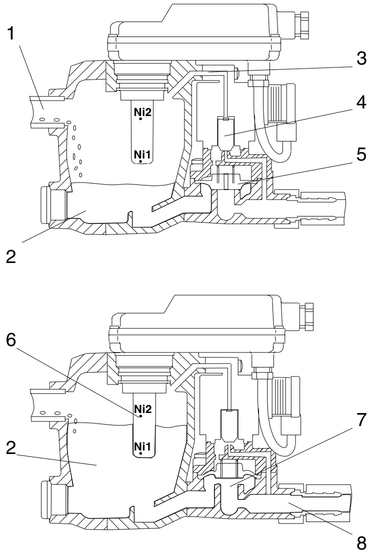
Note

If you wish to avoid having condensate in the pipe system the entire volume of condensate must be drained off. Individually adjustable opening times for the solenoid valve guarantee perfect drainage of the condensate.
The quantity of condensate in Summer is far greater than in Winter because atmspheric humidity is higher. If the opening times set in Summer for the high humidity are not changed later for Winter, low temperatures will cause high pressure loss because the magnetic valves will be open for too long. Not only the condensate but also large quantities of compressed air will be blown off too.
To minimise compressed air loss the cycle times of the valves must always be adjusted to suit local conditions.
Because the weather is not always consistent, it is not possible to set time intervals and valve opening times and not lose compressed air at all. Either a part of the condensate remains in the system or some compressed air is lost.

Features

- Very reliable operation.
  The system operates reliably, even with problematic condensates.
- Electricity connection required.
- No external malfunction signal.
- No alarm function.
- The solenoid valve operates when the pneumatic station is switched on, even if no compressed air is required (e.g., at weekends).

6.2.4 Condensate drains with electronic level control



1 = Inlet line
2 = Collection tank
3 = Pre-control line
4 = Magnetic valve
5 = Valve diaphragm
6 = Dipstick
7 = Valve seat
8 = Outlet line

Fig. 6.3 :
Condensate drain with electronic volume measurement

Operation

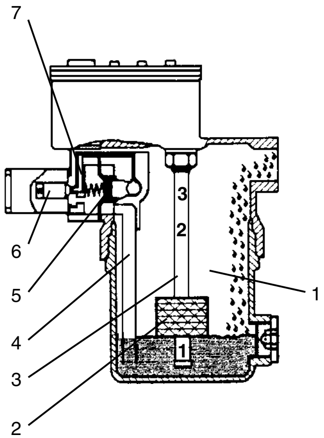
The condensate is collected in an appropriate tank. As soon as the capacity level sensor **Ni2** reports that maximum level, a magnetic valve opens a pre-control line. The pressure on the valve diaphragm is released and the outlet line is opened. The excess pressure in the housing forces the condensate out through the line to the reprocessing facility.

As soon as the level reaches capacity sensor **Ni1**, the magnetic valve is electronically closed. The valve diaphragm closes before compressed air can escape.

Features

- Very reliable operation.
  The system operates reliably, even with problematic condensates.
- Large cross-section.
  Even large impurities and coagulated matter can be discharged without difficulty.
- No pressure loss.
- Electricity connection required.
- Flexible application.
  The system adapts itself automatically to changing operating conditions(e.g., varied condensate viscosity and pressure fluctuations).
- Alarm function.
  If there is a malfunction in draining the condensate the alarm mode is switched after 60 s. The magnetic valve then opens the valve diaphragm at certain intervals.
- External malfunction signal.
  A red LED blinks and a potential-free signal is ready.
- Wide performance range.

6.2.5 Condensate drains with float operated level control



1 = Collection tank
2 = Level float
3 = Guide
4 = Rising pipe
5 = Valve diaphragm
6 = Magnetic valve
7 = Control line

Fig. 6.4 :
Condensate drain with level float for measuring the level

The collected condensate is directed into the collection chamber of the condensate drain. A float moves on a guide together with the level of the condensate in the chamber. The guide has three contacts that electronically register the level in the chamber. As soon as the float reaches **Contact 2**, the electronic control opens the magnetic valve. The pressure on the valve diaphragm is released via a pre-control line and the outlet line is opened. The system pressure forces the condensate out of the condensate drain through a rising pipe.

The level of condensate in the pipe drops and after a set time **t** the control closes the drain before compressed air can escape. If the condensate level does not reach **Contact 1** inside the time **t**, the drain is opened at fixed time intervals and reclosed after a set period. This guarantees that the condensate collection chamber is completely emptied.

If the condensate level reaches **Contact 3,** the control actuates the main alarm. The switching intervals and opening times remain unchanged.

Features

– Cleaning cycle times.
  Even with longer idle times there is no dried condensate.
– No pressure loss.
– electricity connection required.

6.3 Condensate treatment

Condensate from oil-lubricated compressors has an oil content of between 200 and 1000 mg/l, depending on the season. This means that the condensate is around 99 % water and only 1 % oil. Even so, the law requires that this condensate be treated as waste water containing oil. As such it may not be discharged into the public sewers. The stipulations for water purity are set forth in § 7a of the [German] Water Purity Act (WHG). This states that the level of harmful substances in waste water is to be kept as low as the „generally recognised practices of engineering“ allow. These practices have been defined by the German government in general administrative rules.

According to ATV (Waste Water Association, a non-profit-making German organisation) worksheet A 115 the limit for residual oil content in waste water is 20 mg/l. However, the local authorities have the final word. In some areas the limits for residual oil content are well below 20 mg/l.

This means that condensate from oil lubricated compressors must either be disposed of properly or treated. One other solution is the use of biodegradable compressor oils.

Disposal

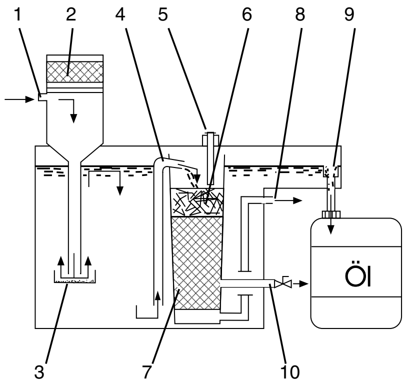
Disposal by a specialised company is a safe but involved and very expensive procedure. Disposal costs currently run at around 250 € per m^3 of condensate. The costs for approved collection tanks and pipelines must also be taken into account.

Local treatment

Because of the high water content, it is always worth treating oily condensate on site. Properly treated water can be discharged into the public drainage lines. The oil separated from it must be disposed of in an environmentally safe way.

The legal limits can not be reached by using normal light liquid separators as per DIN 1999 and simple gravity separators. Standard oil-water separators provide excellent law-compliant treatment.

6.3.1 Oil-water separators



1 = Condensate inlet
2 = Pressure relief chamber
3 = Impurity tank
4 = Overflow pipe
5 = Level reporter
6 = Pre-filter
7 = Adsorption filter
8 = Water outlet
9 = Oil overflow, height adjustable
10 = Specimen removal valve

Fig. 6.5:
Op. diagram of an oil-water separator

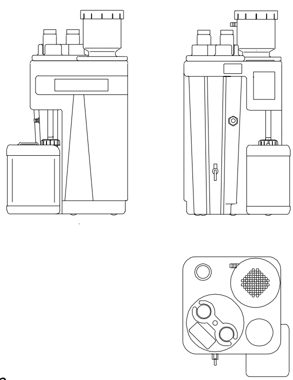


Fig. 6.6 :
Oil-water separator

The oil-water separator is suitable for treating condensates that occur during the operation of screw compressors with oil injection cooling and 1- and 2-stage piston compressors.

The oil-water separator parts condensate from piston and screw compressors without difficulty as long as oils that do not emulsify are used.

Operation

The oily condensate is directed into the pressure relief chamber of the oil-water separator. There the pressure reduces without causing whirling movement in the vessel. The impurities carried with the condensate gather in the removable collector.

Inside the separation vessel, the oil deposits on the surface as a result of its lower specific density. Via a height-adjustable overflow the oil is directed into the oil catch pan and is available for disposal.

The pre-purified condensate flows through a pre-filter that filters out the remaining droplets of oil. After that an adsorption filter binds the last oil parts of the oil.

Note

All oil-water separation systems are water treatment plants and must be licensed by law. The oil-water separator should have a specification inspection symbol. The expensive licensing process is then no longer required. All that needs to be done is to register at the local water authority.

Features

– Weekly filter test.
  A specimen of the condensate is compared with a reference liquid. A change of filter is necessary after the admissible cloudiness is reached.
– No separation of oil-water emulsions.
  Special reprocessing systems with emulsion-splitting apparatus is required for these stable emulsions.

7. Compressed air requirement

The first step in designing a compressor station and the respective pneumatic network is to determine the requirement for compressed air and the resulting FAD of the compressor.

The first value to be found when determining the capacity of a compressor station is the expected total consumption. The consumption of the individual consumer devices is added together and adjusted to operating conditions with the aid of calculations. The compressor can then be selected according to the resulting FAD figure.

The procedure is similar for determining the size of pipelines. Definition of the type and number of consumer devices on a certain section of line comes first. The consumption of the individual devices is added together and adjusted with the appropriate factors. The diameter of the respective section of piping can then be deduced according to the result.

Loss through leakage must also be taken into account when determining the expected consumption of compressed air.

7.1 Consumption of compressed air by pneumatic devices

Determining the total consumption of compressed air is often difficult due to lack of information about individual components. This chapter provides guideline values for the requirements of individual components.

The information given here concerning the consumption of individual devices are average values. Please contact the makers of the devices for exact figures.

7.1.1 Consumption of nozzles

The consumption of compressed air by nozzles of different shapes can vary greatly and depends on various factors :

- Diameter of the nozzle.
  The larger the nozzle, the greater the consumption.
- Operating pressure of nozzle.
  The higher the operating pressure, the greater the consumption.
- Shape of nozzle.
  A simple, cylindrical through-hole consumes much less compressed air than a conical or Laval-nozzle (expansion nozzle).
- Surface quality of aperture.
  If the quality of the aperture is very good (surface very smooth, no grooves and unevenness), more compressed air can flow through.
- Spraying or blowing.
  The consumption of compressed air rises if the air is being used as a medium for paint, sand, or the like.

7.1.1.1 Compressed air consumption of cylindrical nozzles

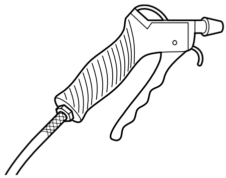


Fig. 7.1 :
Blow-out gun

Nozzles with a simple, cylindrical bore (e.g., blow-out guns) generate strong whirling and turbulence in the compressed air that flows out. This reduces the speed of with which it flows. Consumption is comparatively low.

The following table gives reference values for the compressed air consumption of cylindrical nozzles depending on operating pressure and nozzle diameter :

Nozzle ∅	Operating pressure [bar_{op}]						
[mm]	2	3	4	5	6	7	8
0,5	8	10	12	15	18	22	28
1,0	25	35	45	55	65	75	85
1,5	60	75	95	110	130	150	170
2,0	105	145	180	220	250	290	330
2,5	175	225	280	325	380	430	480
3,0	230	370	400	465	540	710	790

*Air consumption values in the table are given in **l/min**.*

7.1.1.2 Compressed air consumption of paint spray guns

Fig. 7.2 :
Paint spray gun with paint tank

Paint applied by a spray gun must be even and not drip. The nozzles of spray guns are therefore designed for an expanding, non-turbulent volume flow with a high exit speed. The consequence is high consumption of compressed air, well above that of cylindrical nozzles.

The consistency and desired quantity of paint to be applied determines the operating pressure and the nozzle diameter of the spray gun. These two values considerably influence the compressed air requirement.

With paint spray guns, a distinction is made between flat, broad and round spray nozzles. The type of spray influences the application of paint. There is also a difference in the compressed air requirement. On many spray guns it is possible to switch the types of spray.

The following table gives reference values for the compressed air consumption of spray paint nozzles depending on operating pressure, nozzle diameter and type of spray:

Nozzle Ø	Operating pressure [bar_{op}] Flat and broad spray						
[mm]	2	3	4	5	6	7	8
0,5	100	115	135	160	185	–	–
0,8	110	130	155	180	225	–	–
1,0	125	150	175	200	240	–	–
1,2	140	165	185	210	250	–	–
1,5	160	180	200	225	260	–	–
1,8	175	200	220	250	280	–	–
2,0	185	210	235	265	295	–	–
2,5	210	230	260	300	340	–	–
3,0	230	250	290	330	375	–	–

*Air consumption values in the table are given in **l/min**.*

Nozzle Ø	Operating pressure [bar_{op}] Round spray						
[mm]	2	3	4	5	6	7	8
0,5	75	90	105	–	–	–	–
0,8	85	100	120	–	–	–	–
1,0	95	115	135	–	–	–	–
1,2	110	125	150	–	–	–	–
1,5	120	140	155	–	–	–	–

*Air consumption values in the table are given in **l/min**.*

7.1.1.3 Compressed air consumption of jet nozzles

When spraying, the medium must hit the workpiece with great kinetic energy i.e., with high speed. This is the only method that will achieve the desired result.

For this reason jet nozzles are designed for an extremely high exit speed of the compressed air. This leads to comparatively high consumption of compressed air.

The following table gives reference values for the compressed air consumption of jet nozzles depending on operating pressure and nozzle diameter:

Nozzle ∅	**Operating pressure [bar_{op}]**						
[mm]	**2**	**3**	**4**	**5**	**6**	**7**	**8**
3,0	300	380	470	570	700	–	–
4,0	450	570	700	840	1000	–	–
5,0	640	840	1050	1270	1500	–	–
6,0	920	1250	1600	1950	2200	–	–
8,0	1800	2250	2800	3350	4000	–	–
10,0	2500	3200	4000	4800	6000	–	–

*Air consumption values in the table are given in **l/min**.*

7.1.2 Compressed air requirement of cylinders

Compressed air cylinders are especially used in the area of automation. A distinction is made between two types of cylinder when determining the consumption of compressed air:

– The single-action cylinders use compressed air to generate the movement of the working stroke only. The return stroke is performed by spring power or from the outside.

– Double-action cylinders use compressed air to generate movement in both stroke directions. Force is used for both strokes. Accordingly, he consumption of compressed air is twice as high.

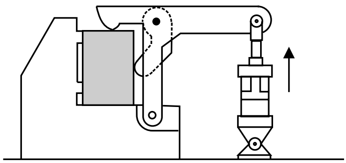


Fig. 7.3:
Clamping device with pneumatic cylinder

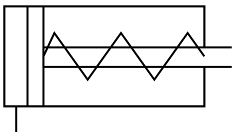
The compressed air consumption **q** of pressure cylinders is determined by using the following formula:

$$q = \frac{d^2 \times \pi}{4} \times S \times p \times a \times b$$

q	=	Air consumption (1 bar_{abs} and 20° C)	[l/min]
d	=	Piston diameter	[dm]
S	=	Length of piston path (Stroke)	[dm]
p	=	Operating pressure	[bar_{abs}]
a	=	Work cycles per minute	[1/min]
b	=	**1** with single-action cylinders **2** with double-action cylinders	

Example

A singe-action cylinder with a piston diameter of 100 mm is required to work at an operating pressure of 7 bar_{abs}. Its working stroke is 130 mm at 47 work cycles per minute.



d = 100 mm ≙ 1 dm

S = 130 mm ≙ 1,3 dm

p = 7 bar_{abs}

a = 47

b = 1

$$q = \frac{1^2 \times \pi}{4} \times 1,3 \times 7 \times 47 \times 1$$

$$q = \text{approx. } 336 \text{ l/min}$$

This cylinder consumes **approx. 336 Litres** of compressed air per minute.

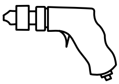
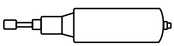
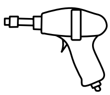
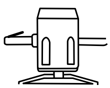
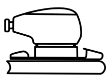
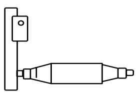
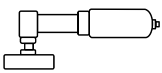
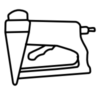
7.1.3 Compressed air consumption of tools

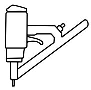
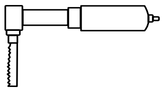
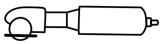
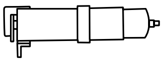
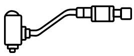
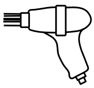
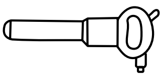
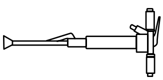
Fig. 7.4 :
Drive screw powered by compressed air

Pneumatic tools are among the most frequent consumers of compressed air in industry and the crafts. They are large numbers of them in almost every environment.

They generally require a working pressure of 6 bar however, there are versions that use other working pressures, depending on the application and the performance required. In these cases the consumption of compressed air will differ from the levels shown in the table.

The following table gives reference values for consumption of compressed air by a number of pneumatic tools. These values may vary for individual tools and should be regarded as averages.

Tool **Working pressure 6 bar$_{op}$**			**Air consumption** **[l/min]**
	Drill	Drill up to 4 mm ∅ 4 – 10 mm ∅ 10 – 32 mm ∅	200 200 – 450 450 – 1750
	Screw machine	M3 M4 – M5 M6 – M8	180 250 420
	Drive screw	M10 - M24	200 – 1000
	Angle grinder		300 – 700
	Vibration grinder	1/4 Sheet 1/3 Sheet 1/2 Sheet	250 300 400
	Belt grinder		300 – 400
	Hand grinder	Collet chucks 6 - 8 mm ∅ 8 - 20 mm ∅	300 – 1000 1500 – 3000
	Tacker, tack chuck		10 – 60

Device **Working pressure 6 bar$_{op}$**		**Air consumption** **[l/min]**
	Nailer	50 – 300
	Sash saw (wood)	300
	Plastic and textile shears	250 – 350
	Metal shears Chamfer mortiser (wood and plastic) Chamfer plane (phases for welding seams)	400 – 900 250 – 400 2500 – 3000
	Rust remover (descaler)	250 – 350
	Needle rust remover (needle descaler)	100 – 250
	Light universal hammer Rivet, chisel and mortise hammer Light pick and shaft hammer Heavy pick and shaft hammer Pneumatic spade Drill hammer	150 – 380 200 – 700 650 – 1500 – 3000 900 – 1500 500 – 3000
	Stamp hammer(foundries) Stamp hammer (concrete and earth) Vibrator (inside - outside)	400 – 1200 750 – 1100 500 – 2500

7.2 Determining compressed air requirement

When determining the compressed air requirement of a pneumatic network, it is not simply a case of adding the consumption values of the individual devices. Other factors that influence consumption must also be taken into account.

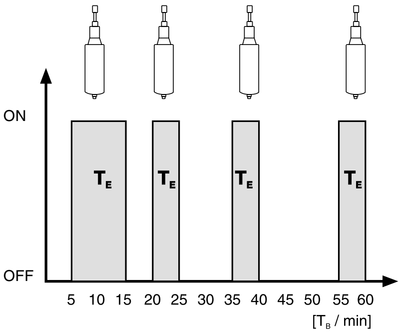
7.2.1 Average operation time

Most pneumatic devices, such as tools, spray paint guns and blow-out guns are not in continuous use. They are switched on and off when needed. It is therefore necessary to find out the average usage rate **UR** in order to obtain an accurate figure for the compressed air requirement.

The following formula is used to determine the average usage rate **UR**:

$$\mathbf{UR} = \frac{\mathbf{T_U}}{\mathbf{T_R}} \times \mathbf{100\,\%}$$

UR	=	average usage rate	[%]
T_U	=	usage time	[min]
T_R	=	reference time	[min]



T_U = 25 min

T_R = 60 min

Fig. 7.5 :
Average operation time

Example

A semiautomatic screwdriver is in use for 25 min in the course of one hour.

$$\mathbf{UR} = \frac{\mathbf{25}}{\mathbf{60}} \times \mathbf{100\,\%}$$

$$\mathbf{UR} = \mathbf{41{,}6\,\%}$$

The usage rate **UR** of the tool is 41,6 %.

The average usage rates **UR** of some widely used pneumatic devices is given in the following table. The figures are based on general experience and may deviate sharply in special cases.

Tool	Average usage rate
Drill	30 %
Grinding machine	40 %
Mortise hammer	30 %
Stamp hammer	15 %
Forming machine	20 %
Blow-out gun	10 %
Tooling machine	75 %

7.2.2 Simultanity factor

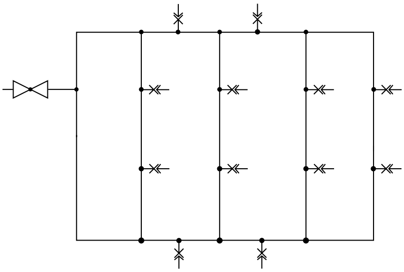


Fig. 7.6 :
Supplying several consumer devices on a pneumatic network

The simultanity factor **f** is an empirical value. It is based on experience of pneumatic devices that are not in use at the same time. The simultanity factor **f** is a multiplier that adjusts the theoretical total consumption of a number of devices to realistic conditions.

The following table gives the generally recognised values for the simultanity factor **f**:

Qty. consumer devices	Simultanity factor f
1	1,00
2	0,94
3	0,89
4	0,86
5	0,83
6	0,80
7	0,77
8	0,75
9	0,73
10	0,71
11	0,69
12	0,68
13	0,67
14	0,66
15	0,64
16	0,63

The simultanity factor **f** is used with the following pneumatic devices:

– Non-automatic nozzles as described in **chapter 7.1.2.**

– Non-automatic pneumatic tools as described in **chapter 7.1.3.**

– Machine tools, production machinery and the like, if no other requirement is specified.

7.2.3 Defining compressed air requirement

When defining the total compressed air requirement for a pneumatic network the consumer devices are divided into two groups:

– Automatic consumer devices.
– General consumer devices.

7.2.3.1 Automatic consumer devices

The consumer group includes automatic pneumatic cylinders, machinery in continuous operation and longer work cycles that require compressed air. These must be calculated at total individual consumption **q** when working out the requirement.

Automatic consumer devices	**Working pressure [bar_{op}]**	**Quantity Q [Units]**	**Individual consump. q [l/min]**	**Q × q [l/min]**
Automatic compressed air cylinders	6	2	336	672
Working machinery	5	1	310	310
Total T_Q compressed air consumption of all automatic devices			**[l/min]**	**Σ 982 l/min**

7.2.3.2 General consumer devices

Most work cycles only run some of the time. An average usage rate **UR** can be calculated for these processes. Also, the consumer devices are not usually all in use at the same time.

The average usage rate **UR** and the simultanity factor **f** are used for general consumer devices as requirement-reducing multipliers when making the calculation.

General compr. air consumers	Working pressure [bar_{op}]	Usage rate UR [%]	Quantity Q [Units]	Individual consump. q [l/min]	$Q \times q \times UR / 100$ [l/min]
Spray paint guns ∅ 1,5 mm	3	40	1	180	72
Blow-out guns ∅ 1,0 mm	6	10	3	65	19,5
Drive screw M10	6	20	3	200	120
Drills up to ∅ 20 mm	6	30	1	700	210
Angle grinders	6	40	2	500	400
Total T air consumption of all general consumer devices [l/min]				**Σ**	**821,5**
Simultanity factor f					**0,71**
Air consumption T_f of general consumer devices		**$T_f = f \times T$**		**[l/min]**	**583,3**

7.2.3.3 Total compressed air consumption

The theoretical total compressed air consumption $\dot{T}$ is the sum of the consumption of automatic and general devices.

$$\dot{T} = T_Q + T_f$$

$$\dot{T} = 982 + 583{,}3$$

$$\dot{T} = 1565{,}3 \text{ l/min} = 1{,}57 \text{ m}^3\text{/min}$$

However, the total compressed air consumption is not yet a suitable figure for determining the capacity of the compressor and the size of the pipes. Several allowances still have to be made.

7.2.4 Allowances for losses and reserves

Allowances	[%]
Losses	5 - 25
Reserves	10 - 100
Error	5 - 15

Several allowances must be taken into account to bring the total consumption figure for individual devices to the actual output requirement of a compressor:

Losses v [%]

Losses **v** through leakage and friction occur in all parts of a pneumatic system. New systems require an allowance of approx. 5 % of total FAD to be added for losses. Since leakages and friction losses generally increase when the equipment gets older, losses of up to 25 % should be assumed for older systems.

Reserves r [%]

A pneumatic system is sized according to a current estimate of compressed air consumption. Experience shows that consumption usually rises later. It is advisable to take short and medium-term extensions of the network into account when planning the size of the compressor and main pipelines. If this is not done, later extension of the system can be unnecessarily expensive. An allowance for reserves **r** of up to 100 % can be taken, depending on the outlook.

Margin for error f [%]

Despite the care taken in calculation, the figures for expected compressed air consumption are still usually wrong. An exact figure can seldom be arrived at because of marginal conditions that are mostly unclear. If a pneumatic system is designed too small and needs to be extended later it will cause additional costs (equipment out of action), and so an allowance **f** of 5 - 15 % is advisable to provide a margin for error.

7.2.5 FAD Required L_B

When calculating the FAD required $\mathbf{L_B}$, allowances of 5 % for losses, 10 % for reserves and a 15 % margin for error are added to the calculated total consumption value $\dot{\mathbf{T}}$

$\dot{T}$ = 1565,3 l/min

v = 5 %

r = 10 %

e = 15 %

$$L_B = \frac{\dot{T} \times (100 + v + r + e)}{100}$$

$$L_B = \frac{1565{,}3 \times (100 + 5 + 10 + 15)}{100}$$

$$L_B = 2035 \text{ l/min} = 2{,}04 \text{ m}^3\text{/min}$$

The FAD quantity $\mathbf{L_B}$, required to give consumer devices an adequate supply of compressed air is approx. 2035 l/min. This value is the basis for determining the size of the compressor and the main pipeline.

7.3 Compressed air loss

Compressed air loss is consumption of air (leakage) in the pipelines without work being performed. These losses can amount to 25 % of the entire FAD of the compressor in unfavourable circumstances.

The causes are numerous:

- Leaking valves.
- Leaking screw and flange joints.
- Leaking weld seams or soldered points.
- Damaged hoses and hose connections.
- Defective solenoid valves.
- Jammed float drains.
- Incorrectly installed dryers, filters and service facilities.
- Corroded lines.

7.3.1 Costs of compressed air loss

Leaks in a pipeline act like nozzles from which compressed air escapes at high speed. These leaks consume compressed air 24 hours per day. The energy needed to compensate for this loss can be considerable. This does not cause physical injury, but the resulting expense can seriously diminish the cost-effectiveness of the pneumatic system.

One example demonstrates the magnitude of the additional cost :

With a network pressure of 8 bar approx. 75 l/min = 4,5 m³/h escape from a leak of 1 mm diameter. A motor output of 0,6 kW is required for this volume flow. At a price of 0,10 €/kWh and 8 000 hours of operation, the additional annual cost amounts to approx. € 480,– depending on the efficiency of the motor.

Leak hole - Ø		Air escaping at 8 bar_{op}	Losses	
			Energy	Cash
[mm]	Size	[l/min]	[kW]	[€/Y]
1	∘	75	0,6	480
1,5	∘	150	1,3	1040
2	○	260	2,0	1600
3	○	600	4,4	3520
4	○	1100	8,8	7040
5	○	1700	13,2	10580

7.3.2 Quantifying leakage

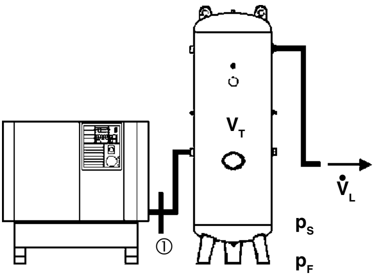
The first step in minimising compressed air loss is to quantify the leakage $\dot{V}_L$. There are two ways of doing this:

7.3.2.1 Quantifying leakage by emptying the receiver

The simplest way of quantifying leakage $\dot{V}_L$ is by emptying the compressed air receiver.

The supply line to the receiver is plugged ①. All consumer devices in the system must be switched off. The receiver pressure p_S drops as a result of the leak to pressure p_F. The time **t** is measured.

The following formula is used to roughly quantify the volume of leakage $\dot{V}_L$:



V_T = 1000 l

p_S = 8 bar

p_F = 7 bar

t = 2 min

$$\dot{V}_L = \frac{V_T \times (p_S - p_F)}{t}$$

$\dot{V}_L$	=	Volume of leakage	[l/min]
V_T	=	Volume of receiver	[l]
p_S	=	Receiver pressure at start	[bar_{op}]
p_F	=	Receiver pressure at finish	[bar_{op}]
t	=	Time measured	[min]

Example

A compressed air receiver with a large pipeline system has a volume of 1000 l. Within 2 min. the receiver pressure drops from 8 to 7 bar_{op}.

$$\dot{V}_L = \frac{1000 \times (8 - 7)}{2}$$

$$\dot{V}_L = 500 \text{ l/min}$$

The leakage volume of this pneumatic system is **approx. 500 l/min**.

Note

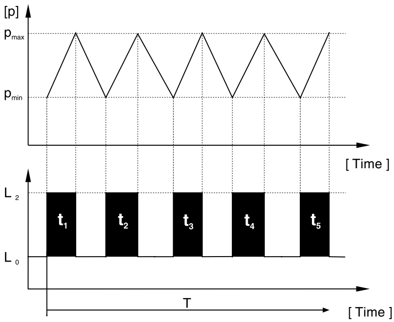
This method of measuring is only suitable for systems where the pipeline system is less than 10 % of the volume of the receiver. Otherwise the results are too inaccurate.

7.3.2.2 Quantifying leakage by measuring working time

The second method of quantifying the volume of leakage $\dot{V}_L$ is by measuring the operating time of the compressor. This method can only be used with compressors having intermittent and idling operation modes.

The consumer devices in the network are switched off. The leaks in the system consume compressed air and the network pressure drops. The compressor must replace this volume.

The total running time **Σ t** of the compressor is measured over a period of time **T**. To obtain a realistic result, the measuring time **T** should last for at least 5 cycle intervals of the compressor.



$\dot{V}$	=	1,65	m³/min
Σt	=	30	s
T	=	180	s

The following formula is used to roughly quantify the volume of leakage : $\dot{V}_L$

$$\dot{V}_L = \frac{\dot{V} \times \Sigma t \times 1000}{T}$$

$$\text{l/min} = \frac{\text{m}^3\text{/min} \times \text{s} \times 1000\ \text{l}}{\text{s} \times \text{m}^3}$$

$\dot{V}_L$	=	Volume of leakage	[l/min]
$\dot{V}$	=	Compressor FAD	[m³/min]
Σt	=	Total running time of compressor $\Sigma t = t_1 + t_2 + t_3 + t_4 + t_5$	[s]
T	=	Measuring time	[s]

Example

A compressor with an effective FAD $\dot{V}$ of 1,65 m³/min has five actuations during a measuring time of **T** = 180 s. Its total running time **Σ t** during the measuring time **T** is 30 s.

$$\dot{V}_L = \frac{1{,}65 \times 30 \times 1000}{180}$$

$$\dot{V}_L = 275\ \text{l/min}$$

The leakage volume of this pneumatic system is **approx. 275 l/min**.

7.3.3 Limits for leakage

Unfortunately, compressed air loss through leakage is inevitable in normal pneumatic systems. The additional costs caused by leakage reduce the cost-effectiveness of the system considerably. Measures can be taken to reduce the loss, but this causes costs money as well. At some point, these costs will outweigh the savings made by cutting the loss of compressed air. The objective must therefore be to minimise the loss of compressed air at acceptable expense.

There are therefore some levels of leakage that should be tolerated for reasons of economy:

- max. 5 % on smaller networks.
- max. 7 % on medium-sized networks.
- max. 10 % on larger networks.
- max. 13 - 15 % on very large networks.
  e.g., foundries, steel mills, shipyards etc.

7.3.4 Measures for minimising compressed air loss

Staff should be instructed to report leaks and damage to the network to the persons in charge. Damage should be rectified immediately. If a system is looked after on a permanent basis, there will normally be no need for expensive reconstruction of the network. Compressed air loss will be kept at an acceptable level.

Leaks

It is usually quite easy to find leaks. Large leaks can be heard.

However, small and very small leaks are harder to find and can not usually be heard. In these cases, the joints, branches, valves etc. are covered with seal checker or soapy water. Bubbles form immediately where there are leaks. Another easy method is the use of on ultrasonic leakage detector. This device detects the ultrasonic noises caused by a leakage.

7.3.5 Reconstructing a pneumatic network

If the leakage volume lies clearly above the levels specified in **chapter 7.3.3**, reconstruction of the system should be considered.

When reconstructing a pneumatic system the following measures should be taken to reduce compressed air loss:

- Tighten leaking joints or reseal them.
- Replace leaking joints and slides.
- Replace leaking hoses and hose connectors.
- Weld leaks on pipelines.
- Upgrade condensate drains.
  Replace mechanical float drains and time-controlled solenoid valves with level-controlled condensate drains.
- Upgrade compressed air preprocessor.
  Remove harmful impurities such as water, oil and dust from the compressed air.
- Check solenoid valves.
  If possible, install normally closed valves.
- Flush or replace old pipelines.
  The inside diameter of old pipes is often reduced by deposits. This causes a drop in pressure.
- Check couplings and pipe connections.
  Reductions in the size of cross-sections causes a drop in pressure.
- Reduce the size of the system for limited periods.
  Cut off parts of large systems with stop valves when not needed.

8. Determining the size of the compressor station

8.1 The type of compressor

The primary decision when installing a compressor station is choosing the type of compressor. Screw or piston compressors are the right choice for nearly all applications.

8.1.1 Screw compressors

Fig. 8.1:
BOGE screw compressor, Series S

Screw compressors are particularly suitable for certain applications.

- Long usage rate **UR**.
  Screw compressors are particularly suitable in situations where consumption of compressed air is continuous and without large peak loads (UR = 100 %). They are excellent as base load machines in composite compressor systems.
- High FAD.
  The screw compressor is the most economical type where high FAD is needed.
- Pulse-free volume flow.
  Through uniform compression the screw compressor can also be used for very sensitive consumer devices.
- Screw compressors operate economically with final compression pressures of between 5 and 14 bar.
  The normal maximum pressure $\mathbf{p}_{max}$ categories for screw compressors are 8 bar, 10 bar and 13 bar.

8.1.2 Piston compressors

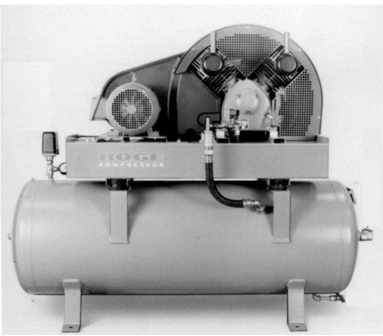


Fig. 8.2:
BOGE piston compressor with horizontal compressed air receiver

Piston compressors also have their special areas of application. They are an ideal supplement to those of the screw compressors.

- Intermittent requirement.
  Piston compressors are suitable for fluctuating consumption of compressed air with load peaks. They can be used as peak-load machines in a compressor group system. These compressors are the best choice for frequently changing loads.
- Small FAD quantities.
  When FAD quantities are small, the piston compressor is more economical than the screw compressor.
- Piston compressors can compress to high final pressures.

  The normal maximum pressure $\mathbf{p}_{max}$ categories for piston compressors are 8 bar, 10 bar, 15 bar, 30 bar and 35 bar.

8.2 Maximum pressure p_{max}

The next step in determining the size of a compressor with compressed air receiver and air treatment is to define the maximum pressure of the compressor $\mathbf{p_{max}}$.

The basis for the maximum pressure (cutout pressure $\mathbf{p_{max}}$) is the cycle difference **(p_{max} - p_{min})** of the compressor control, the maximum operating pressure of the consumer devices and the total pressure loss within the network.

8.2.1 Factors influencing cutout pressure p_{max}

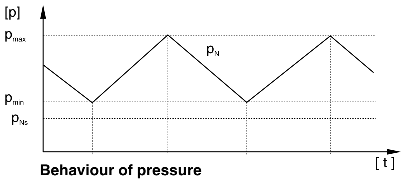


Fig. 8.3:
Behaviour of pressure in a compressed air receiver

The receiver pressure, which fluctuates between $\mathbf{p_{min}}$ and $\mathbf{p_{max}}$, must always be much higher than the operating pressures of the consumer devices in the network. Pressure loss always occurs in pneumatic systems. This is why the pressure loss caused by the various components of a pneumatic system must be taken into consideration.

The following values must be considered when defining the cutout pressure $\mathbf{p_{max}}$:

– Normal pneumatic networks **≤ 0,1 bar**
  The network should be designed so that the total pressure loss **Δp** of the entire network does not exceed 0.1 bar.
– Large pneumatic networks **≤ 0,5 bar**
  On widely branched networks, e.g., in mines, quarries or large building sites, a pressure loss **Δp** up t 0.5 bar can be allowed.
– Treatment of compressed air by dryer.
  Diaphragm compressed air dryer with Filter **≤ 0,6 bar**
  Refrigeration compressed air dryer **≤ 0,2 bar**
  Adsorption compressed air dryer with filter **≤ 0,8 bar**
– Compressed air treatment by filters and separators.
  Dust separator **≤ 0,05 bar**
  Filters generally **≤ 0,6 bar**
  The pressure loss **Δp** through filters rises from soiling. The latest time at which the filter must be changed is specified.
– The cycle difference of the compressor.
  Screw compressors **0,5 - 1 bar**
  Piston compressors **p_{max} - 20 %**
– Reserves.
  Unforeseen pressure loss occurs time and again in pneumatic systems. An adequate contingency reserve should always be planned for in order to avert performance loss.

8.3 Determining the volume of a compressed air receiver

Compressed air receivers are tanks used for storing compressed air, damping pulsation and separating condensate in the pneumatic system. The receiver must be of the correct size to be able in particular to fulfill its task of storing compressed air.

8.3.1 Recommendations for the volume of compressed air receivers

Determining receiver volume $\mathbf{V_R}$ is accomplished primarily by values gained from experience. BOGE recommends the following ratios of compressor FAD $\dot{\mathbf{V}}$ **[l/min]** to receiver volume $\mathbf{V_R}$ **[l]** :

– Piston compressors. $\mathbf{V_R = \dot{V}}$
  Intermittent running is aimed for due to the properties of the compressor.

– Screw compressors $\mathbf{V_R = \dot{V}/3}$
  Constant running is aimed for due to the properties of the compressor.

After defining the volume of the receiver, with piston compressors it is time to work out the cycle interval, which comprises the compressor running and idling times. The number of compressor cycles results from this.

8.3.2 Norm series and operating pressures for sizes of compressed air receivers

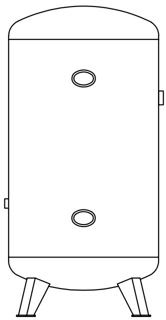


Fig. 8.4:
Vertical compressed air receiver

Compressed air receivers are available in sensibly graduated volume sizes. A standard size should always be chosen to save unnecessary costs for custom-made equipment.

The maximum pressure for which the receiver is designed is, for safety reasons, always at least 1 bar above the maximum pressure of the compressor. 10 bar compressors have, for instance, a compressed air receiver designed for 11 bar. The safety valve is adjusted for 11 bar.

The following table shows the sizes of compressed air receiver available for various operating pressures:

Compressed air	**Operating pressure up to**		
receiver vol. [l]	**11 [bar]**	**16 [bar]**	**36 [bar]**
18	•		
30	•		
50	•	•	
80	•		
150	•	•	•
250	•	•	•
350	•	•	•
500	•	•	•
750	•	•	•
1000	•	•	•
1500	•	•	•
2000	•	•	•
3000	•	•	•
5000	•	•	•

8.3.3 Volumes of compressed air receivers for compressors

The ideal capacity of compressed air receiver for a compressor can be defined more precisely with the aid of a formula.

The formula is ideal when long idle periods are planned with intermittent operation. The volume of the pneumatic system can be considered as a part of the receiver volume.

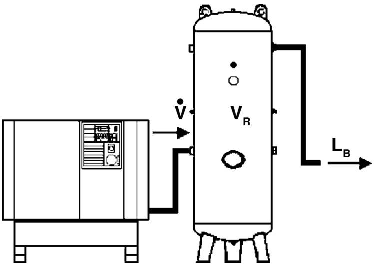


Fig. 8.5:
Compressor and compressed air receiver

$$V_R = \frac{\dot{V} \times 60 \times [\,L_B/\dot{V} - (L_B/\dot{V})^2\,]}{Al \times (p_{max} - p_{min})}$$

V_R = Volume of compressed air receiver [m^3]

$\dot{V}$ = FAD of compressor [m^3/min]

L_B = Required FAD [m^3/min]

Al = Allowed motor cycles / h [1/h]
(see chapter 8.4.3)

p_{max} = Cut-out pressure of compressor [bar_{op}]

p_{min} = Cut-in pressure of compressor [bar_{op}]

Despite taking all influencing factors into account, it is advisable to check the determined receiver size against the allowed motor cycles of the compressor.

Obviously, compressors with small receiver volumes $\mathbf{V_R}$ switch on and off more often. This is a strain on the motor. In contrast, with a large receiver volume $\mathbf{V_R}$ and a constant output the motor of a compressor switches on less often. This spares the motor.

Simple formulae for determining the size of the compressed air receiver

Piston compressor

$$V_R = \frac{Q \times 15}{Al \times \Delta p}$$

Screw compressor

$$V_R = \frac{Q \times 5}{Al \times \Delta p}$$

V_R = Volume of compressed air receiver [m^3]

Q = FAD of compressor [m^3/min]

15 or 5 = Constant factor

Al = Allowed motor cycles / h [1/h]
(see chapter 8.4.3)

Δp = Pressure differential ON/OFF

8.4 Compressor cycle intervals

The cycle interval is an important factor in a pneumatic system. To check the correct size of the receiver in relation to the FAD and compressed air consumption the cycle interval must first be calculated. This is done by calculating the compressor running time $\mathbf{t_R}$ and the compressor idle time $\mathbf{t_I}$, the sum of which provides the cycle interval.

8.4.1 Compressor idle times

During the compressor idle time $\mathbf{t_I}$ the compressed air requirement is covered from the volume of air stored in the receiver. The pressure in the receiver thus drops from the cutout pressure $\mathbf{p_{max}}$ to the cut-in pressure $\mathbf{p_{min}}$. During this time the compressor does not deliver compressed air.

The following formula is used to determine the compressor idle time $\mathbf{t_I}$:

$$t_I = \frac{V_R \times (p_{max} - p_{min})}{L_B}$$

t_I	=	Idle time of compressor	[min]
V_R	=	Volume of compressed air receiver	[l]
L_B	=	Required FAD	[l/min]
p_{max}	=	Cut-out pressure of compressor	[bar_{op}]
p_{min}	=	Cut-in pressure of compressor	[bar_{op}]

8.4.2 Compressor running times

During running time the compressor compensates the pressure loss in the receiver. At the same time the current compressed air requirement is covered. The output $\dot{\mathbf{V}}$ is higher than the actual consumption $\mathbf{L_B}$. The pressure in the receiver rises back to $\mathbf{p_{max}}$.

The following formula is used to determine the compressor running time $\mathbf{t_R}$:

$$t_R = \frac{V_R \times (p_{max} - p_{min})}{(\dot{V} - L_B)}$$

t_R	=	Running time of compressor	[min]
V_R	=	Volume of compressed air receiver	[l]
L_B	=	Required FAD	[l/min]
$\dot{V}$	=	FAD of compressor	[l/min]
p_{max}	=	Cut-out pressure of compressor	[bar_{op}]
p_{min}	=	Cut-in pressure of compressor	[bar_{op}]

8.4.3 Determining the motor cycle speed

The maximum motor cycle speed depends on the size of the drive motor. The drive motor can be damaged if the maximum number of cycles is exceeded.

To determine the number of expected motor cycles **A** for the compressor, the compressor running time t_R and idle time t_I are added together, and the reference time (normally 60 min) divided by the result.

The compressed air receiver must be larger if the result is above the maximum allowed number of cycles **AI**.

A second possibility would be to increase the cycle diferential **$(p_{max} - p_{min})$**.

$$A = \frac{60}{t_I + t_R}$$

A	=	Cycle speed	[1/h]
t_R	=	Running time of compressor	[min]
t_I	=	Idle time of compressor	[min]

The following table gives the allowed number of cycles for an electric motor per hour depending on the power rating of the motor.

Motor power rating [kW]	Allowed cycles/h AI [1/h]
4 - 7,5	30
11 - 22	25
30 - 55	20
65 - 90	15
110 -160	10
200 - 250	5

8.5 Examples for compressor configuration

8.5.1 Sample calculation for piston compressors

In chapter 7.2.5 the required FAD of **L_B = 2035 l/min** was determined for a number of consumer devices. The maximum required working pressure in this example is 6 bar_{op}. A piston compressor is dimensioned for this case of application.

8.5.1.1 Determining the maximum pressure p_{max}

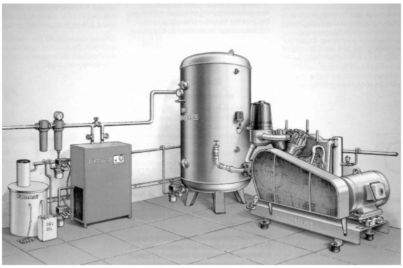


Fig. 8.6:
Compressor station with piston compressor, compressed air receiver, refrigeration compressed air dryer and filter system

The maximum compressor pressure **p_{max}** of the pneumatic system must now be determined. Starting from the working pressure of the consumer devices, all components in the pneumatic system must be taken into consideration:

– Maximum working pressure in the system		6 bar_{op}
– Pneumatic network	Pressure loss	0,1 bar
– Filters	Pressure loss	0,6 bar
– Refrig.compressed air dryer	Pressure loss	0,2 bar
Minimum pressure in receiver		**6,9 bar_{op}**

The cut-in pressure **p_{min}** must always be above this pressure.

– Cycle differential of piston compressors	approx. 2 bar
The cut-out pressure **p_{max}** is at least	**8,9 bar_{op}**
Selected maximum compressor pressure (cut-out pressure of compressor)	**10 bar_{op}**

8.5.1.2 Determining compressor size

Piston compressors are designed with reserves of approx. 40 %. Reserves are included from experience, in order to have a contingency for possible extensions to the system and to use the compressor intermittently. Intermittent operation means less wear.

The ideal usage rate UR for a piston compressor is around 60 %. BOGE piston compressors are designed for 100 % UR = continuous operation. When calculating the ideal compressor size this means: the required FAD $\mathbf{L_B}$ must be divided by 0.6 in order to obtain the minimum FAD $\dot{\mathbf{V}}_{min}$ of the piston compressor.

$$\dot{V}_{min} = L_B \ / \ 0{,}6$$
$$\dot{V}_{min} = 2035 \ / \ 0{,}6$$
$$\dot{V}_{min} = 3392 \text{ l/min}$$

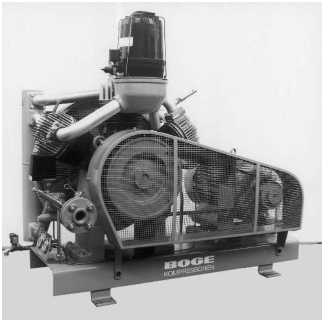


Fig. 8.7:
BOGE piston compressor, type RM 3650-213

The choice is:

Piston compressor Type RM 4150-213

Max. pressure	$\mathbf{p}_{max}$	:	10	bar		
FAD	$\dot{\mathbf{V}}$	:	3350	l/min		
Motor rating		:	30	kW	⇒	AI = 20

8.5.1.3 Volume of the compressed air receiver

The volume of the compressed air receiver should be determined using the BOGE recommendation, compressor FAD $\dot{\mathbf{V}}$ volume of compressed air receiver $\mathbf{V_R}$. The graduations among standardised sizes for receivers must be taken into consideration.

$$\dot{V} = 3350 \text{ l/min} \Rightarrow V_R = 3000 \text{ l}$$

8.5.1.4 Compressor cycle interval

After defining the size of the compressed air receiver it is necessary to determine the compressor running and idle times in order to check the motor cycle rate **C.**

The following formula is used to find the compressor idle time $\mathbf{t_I}$:

V_R = 3000 l

p_{max} = 10 bar_{op}

p_{min} = 8 bar_{op}

L_B = 2035 l/min

$$t_I = \frac{V_R \times (p_{max} - p_{min})}{L_B}$$

$$t_I = \frac{3000 \times (10 - 8)}{2035}$$

$$t_I = 2{,}95 \text{ min}$$

t_I	=	Idle time of compressor	[min]
V_R	=	Volume of compressed air receiver	[l]
L_B	=	Required FAD	[l/min]
p_{max}	=	Cut-out pressure of compressor	[bar_{op}]
p_{min}	=	Cut-in pressure of compressor	[bar_{op}]

The following formula is used to determine the compressor running time $\mathbf{t_R}$:

V_R = 3000 l

p_{max} = 10 bar_{op}

p_{min} = 8 bar_{op}

$\dot{V}$ = 3350 l/min

L_B = 2035 l/min

$$t_R = \frac{V_R \times (p_{max} - p_{min})}{(\dot{V} - L_B)}$$

$$t_R = \frac{3000 \times (10 - 8)}{(3350 - 2035)}$$

$$t_R = 4{,}56 \text{ min}$$

t_R	=	Running time of compressor	[min]
V_R	=	Volume of compressed air receiver	[l]
L_B	=	Required FAD	[l/min]
$\dot{V}$	=	FAD of compressor	[l/min]
p_{max}	=	Cut-out pressure of compressor	[bar_{op}]
p_{min}	=	Cut-in pressure of compressor	[bar_{op}]

8.5.1.5 Motor cycle rate of compressor

t_I = 2,95 min

t_R = 4,56 min

Motor output rating 22 kW ⇒ AI = 25

The motor cycle rate is calculated from the compressor running and idle time and compared with the allowed figure **AI**.

$$C = \frac{60}{t_I + t_R}$$

$$C = \frac{60}{2{,}95 + 4{,}56}$$

$$C = 8$$

C	=	Cycles	[1/h]
t_R	=	Running time of compressor	[min]
t_I	=	Idle time of compressor	[min]

Approx. 8 cycles per hour is well below the allowed number for the 30 kW motor (AI = 20). The compressed air receiver is of a good size. It could even be somewhat smaller because of the large reserve of motor cycles.

Note

If the exact compressed air consumption is not defined, 50% of the of the compressor FAD can be assumed as consumption when determining the motor cycle rate. In this case the idle and running times of the compressor are the same. This results in the maximum number of motor cycles.

8.5.2 Sample calculation for screw compressors

In chapter 7.2.5 the required FAD of L_B = 2,04 m³/min was determined for a number of consumer devices. The maximum required working pressure in this example is 6 bar_{op}. A screw compressor is sized for this application.

8.5.2.1 Example for determining the maximum pressure p_{max}

Fig. 8.8:
Compressor station with screw compressor, compressed air receiver, refrigeration compressed air dryer and filter system

The maximum compressor pressure $\mathbf{p_{max}}$ of the pneumatic system must now be determined. Starting from the working pressure of the consumer devices, all components in the pneumatic system must be taken into consideration:

– Maximum working pressure in the system		6 bar_{op}
– Pneumatic network	Pressure loss	0,1 bar
– Filters	Pressure loss	0,6 bar
– Refrig.compressed air dryer	Pressure loss	0,2 bar
Minimum pressure in tank		**6,9 bar_{op}**

The cut-in pressure $\mathbf{p_{min}}$ must always be above this pressure.

– Cycle differential of screw compressors	1 bar
The cut-out pressure $\mathbf{p_{max}}$ is at least	**7,9 bar_{op}**
Selected maximum compressor pressure (cut-out pressure of compressor)	**8 bar_{op}**

8.5.2.2 Determining compressor size

Fig. 8.9:
BOGE screw compressor

The ideal usage rate **UR** of a screw compressors is 100 %. This means, the required FAD $\mathbf{L_B}$ is equal to the minimum output $\dot{V}_{min}$ of the compressor.

$$\boxed{\mathbf{L_B = 2{,}04\ m^3/min = \dot{V}_{min} = ca.\ 2\ m^3/min}}$$

The choice is:

Screw compressor, Type S 21

Maximum pressure	$\mathbf{p_{max}}$:	8	bar	
FAD	$\dot{V}$:	2,42	m³/min	
Motor output rate	:	15	kW	⇒ AI = 25

8.5.2.3 Dimensioning the compressed air receiver

The volume of the compressed air receiver for screw compressors is calculated with the aid of the following formula. The usual sizes of standard compressed air receivers should be taken into account.

$\dot{V}$ = 2,42 m³/min

L_B = 2,04 m³/min

$^{LB}/\dot{V}$ = 0,843

Al = 25 1/h

p_{max} = 9 bar_{op}

p_{min} = 8 bar_{op}

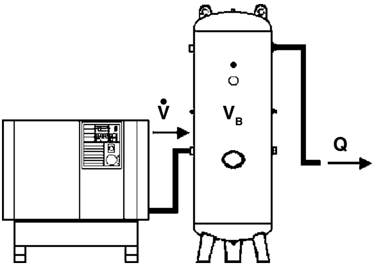


Fig. 8.10:
Compressor and compressed air receiver

$$V_R = \frac{\dot{V} \times 60 \times [\,^{LB}/_{\dot{V}} - (\,^{LB}/_{\dot{V}})^2\,]}{Al \times (p_{max} - p_{min})}$$

$$V_R = \frac{2{,}42 \times 60 \times [\,0{,}843 - 0{,}843^2\,]}{25 \times (9 - 8)}$$

$$V_R = 0{,}77\ m^3$$

Selected receiver volume:

$$V_R = 0{,}75\ m^3 = 750\ l$$

V_R	=	Volume of compressed air receiver	[m³]
$\dot{V}$	=	FAD of all compressors	[m³/min]
L_B	=	Required FAD	[m³/min]
Al	=	Allowed motor cycle rate	[1/h]
p_{max}	=	Cut-out pressure of compressor	[bar_{op}]
p_{min}	=	Cut-in pressure of compressor	[bar_{op}]

The volume of the compressed air receiver can also be defined according to the BOGE recommendation, compressor FAD to volume of compressed air receiver **$V_R = \dot{V}/3$**.

$$\dot{V} = 2{,}46\ m^3/min \Rightarrow V_R = 0{,}81\ m^3$$

8.5.2.4 Compressor cycle interval

The cycle intervals and maximum allowed cycle rate of the motor do not have to be checked with BOGE screw compressors because the microcontroller in the BOGE ARS control unit does not allow the maximum rate to be exceeded.

8.5.3 Summary on compressor selection

If a company expects fluctuating consumption of compressed air and is planning later extensions, it needs a compressor designed for intermittent operation. A piston compressor is the ideal choice. If the FAD of a compressor can cover the constant compressed air requirement then a screw compressor should be used.

Both compressor systems are available with full silencing. Both are supplied ready for use.

The choice of the right system should not depend on the purchase price, because the system pays for itself quickly if overhead operating costs are saved. Overhead operating costs are not only the energy costs to produce compressed air but also the costs of idling.

Piston compressor work in intermittent operation. The do not have an idle mode. Screw compressors must, because of their small cycle differential and relatively small compressed air receiver, automatically run in idle mode in order to avoid having too many motor cycles.

The ARS control unit aims for intermittent operation with minimum idling time.

8.6 Information on compressor configuration

8.6.1 Performance and working pressure

Fig. 8.11:
Impact wrench with pneumatic drive

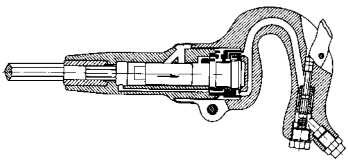


Fig. 8.12:
Valveless pneumatic hammer

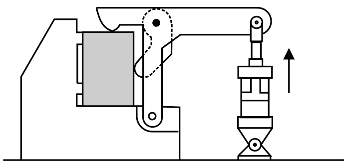


Fig. 8.13:
Pneumatic clamp

The working pressure of consumer devices should always be complied with. The performance of a pneumatic device always drops disproportionally if the network pressure $\mathbf{p}_N$ falls below its working pressure.

The following table shows the dependence of performance on working pressure using the average pneumatic tools and hammers as examples:

Effective pressure [bar]	Relative performance [%]		Relative air consumption [%]	
at connection	tool	drill hammer	tool	drill hammer
7	120	130	115	120
6	100	100	100	100
5	77	77	83	77
4	55	53	64	56

Example

The consequences of network pressure that is too low can be shown using a pneumatic cylinder as an example.

If the pneumatic cylinder of a clamping device is not supplied with the required working pressure, the clamping power of the cylinders falls and the workpiece is no longer held with the necessary force.

The workpiece falls loose from the clamp while being processed by a machine tool. This can result in the destruction of the workpiece and may also injure to the machine operator.

8.6.2 Varying working pressure of consumer devices

If the working pressure of the various consumer devices varies widely, the situation requires closer examination.

Some devices with a low compressed air requirement need a much higher working pressure than others.
In this case a second, small compressor station with a separate pneumatic network and an appropriately higher cut-out pressure $\mathbf{p}_{max}$ should be installed.

The unnecessary overcompression of the main volume flow of the pneumatic system causes considerable costs. In most cases, these additional costs justify the installation of a second system.

The second system usually amortises itself quickly by reducing operating costs.

8.6.3 Combined compressor systems

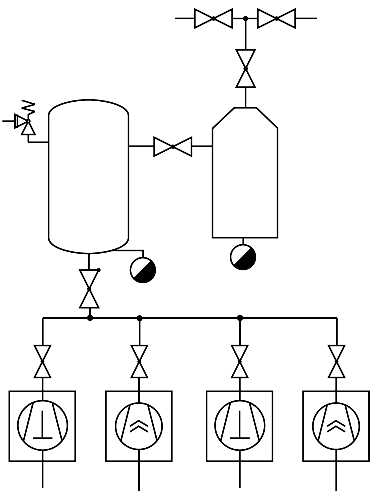


Fig. 8.14:
Diagram of a combined compressor system

For users of compressed air with high, heavily fluctuating consumption, a single, large compressor is not the best solution. The alternative is to have a combined compressor system consisting of several compressors. Greater operational reliability and greater economy are the arguments for this option.

One or more compressors cover the continuous basic requirement for compressed air (basic load). If the requirement rises, additional compressors are switched on (medium and peak load), until the output covers the requirement. If the requirement drops, the compressors are switched back off again, one after the other.

The configuration of individual compressors (free air delivered) in a combined compressor system is individually so different that no general statement are possible. The configuration depends on the pneumatic behaviour of all consumer devices connected to the system.

Advantages

– Operational reliability.
  Operations heavily reliant on compressed air can guarantee their supply at all times with a combined compressor system. If one compressor fails, or if servicing work needs to be done, the other compressors take over the work.

– Economy.
  Several small compressors are easier to adjust to compressed air consumption than one large compressor. This fact makes a system of this type more economical. If the system is only running at half-load, there are no high running costs for a large compressor but low idling costs for small compressors in readiness in a combined system.

9. The pneumatic system

9.1 The compressed air receiver

The size of the compressed air receiver is determined by the FAD of the compressor, the control system, and compressed air consumption. Compressed air receivers have various important tasks in a pneumatic system.

9.1.1 Storing compressed air

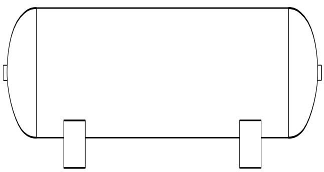


Fig. 9.1:
Horizontal compressed air receiver

The compressor builds up a store of compressed air inside the receiver. The compressed air requirement can be covered at intervals from this store. The compressor does not supply compressed air during this time. It is in readiness and does not use electricity. Additionally, fluctuating use of compressed air is compensated for and peak requirements covered. The motor is switched on less often and wear on it reduced.

In some circumstances several receivers are needed to build up an adequate store of compressed air. Very large pneumatic systems usually have an adequate storage capacity. In this case, smaller receivers can be used.

9.1.2 Pulsation damping

Due to the way they operate, piston compressors generate a pulsing volume flow. These pressure fluctuations impair the operation of various consumer devices. Process control and measuring equipment in particular react to pulsing volume flow by making errors. The compressed air receiver is used to balance out these fluctuations in pressure.

This is much less the case with screw compressors because they generate an almost even volume flow.

9.1.3 Condensate collection

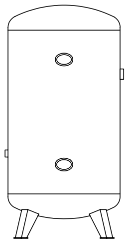


Fig. 9.2:
Vertical compressed air receiver

Compression causes the moisture in the air to form droplets of water (condensate). This water is usually drawn into the compressed air receiver with the volume flow. This is where compressed air is stored. Heat is given off to the cooler surrounding by the large surface of the receiver and the compressed air cools down. This causes a large part of the condensate to precipitate on the walls of the receiver. The condensate collects on the floor of the receiver and is removed by a suitable condensate collector.

Compressed air receivers that are only emptied at irregular intervals can be corroded by the condensate. One protection against corrosion is to galvanise the receiver in a dip-tank. It is not absolutely essential to galvanise the receiver if the condensate is drained regularly. Galvanising is also a useful option if the condensate contains a high concentration of aggressive components.

9.1.4 Operation of compressed air receivers

Compressed air receivers may only be continuously used for compressors with intermittent and idling modes. The area of pressure fluctuation $\mathbf{\Delta p}$ must not exceed 20 % of the maximum operating pressure (max. compressor pressure 10 bar, $\Delta p = 2$ bar). If pressure fluctuations are greater, the welding seams may break as a result of fatigue over the course of time. The compressed air receiver must then be specially designed for fluctuating stress.

9.1.5 Installation of compressed air receivers

The compressed air receiver should be installed in a cool place whenever possible. This will cause more condensate to form inside, which means that less will enter the pneumatic system and the pre-processing unit.

Compressed air receivers should be installed so that they are or can be made accessible for periodic inspections, and with the factory specification plate well visible.

The compressed air receiver should be installed on a suitable foundation with plenty of space for inspections. It should be taken into account that the stress on the foundation increases during pressure testing when the tank is filled with water.

Compressed air receivers must be installed so as not to be a hazard for the staff or other people. The necessary safety zones and distances must be observed.

The compressed air receiver and its ancillary equipment must be protected against mechanical influence from the outside (e.g., from vehicles), so that they are not a hazard for people or equipment.

9.1.6 Safety rules for compressed air receivers

Depending on their size and pressure, compressed air receivers are subject to a number of various rules and directives. Most important of them all is Pressure Equipment Directive 97/23/EC concerning the manufacture of compressed air receivers with a pressure content product exceeding 10,000 bar*l as well as the Simple Pressure Vessel Directive 87/404/EEC for compressed air receivers having a pressure content product of up to 10.000 bar*l. In addition, the user of a compressed air system is required to observe a number of national requirements. In Germany, the Industrial Safety Regulations (Betr SichV) apply for the most part.

Compressors as a whole are considered machines subject to EC machinery Directive 98/37/EC.

9.1.6.1 Registration and inspection obligations

Compressed air receivers are subject to various registration and inspection obligations. Directive 97/23/EC serves to divide such compressed air receivers into different groups in respect of their pressure content product [bar*l] and transported media. In conformity with their classification, requirements for inspection and control may vary, which particularly applies to repetitive inspection intervals.

Periodic inspections are to be repeated at max. 5 years intervals for interior inspection, and at max. 10 years intervals for strength testing. In order to comply with any particular requirements such intervals may be shortened by the competent technical inspection authorities. Not all inspections have to be carried out by the inspection authorities. Under certain circumstances inspections can be carried out by the operator’s own specially qualified and authorized personnel.

9.1.6.2 Approved inspection authorities and authorized personnel

The relevant approved inspection authorities are defined in § 14 of the general product safety directive. Those authorities must be both specified and accredited with their tasks traditionally being carried out by expert personnel. After a transitional period through the end of 2007, terminology will change and tasks will be conveyed to the specified authorities.

A qualified person within the meaning of the general product safety directive is a person who, by virtue of his/her industrial training, professional experience and up-to-date job training, has full command of expert knowledge as may be required for inspecting said equipment (§ 2 clause 7). This, in its broadest sense, corresponds to the previous definition of proficient persons.

9.1.6.3 Inspection prior to commissioning

Any compressed air system subject to control (i.e. compressed air systems as defined in the 97/23/EC and 87/404/EEC directives) may only be put into initial and/or post-modification operation after inspection by an approved authority under consideration of its intended use and with view to proper assembly, installation, setting-up and its safe functioning.

Smaller compressed air receivers as defined in the 87/404/EEC directive and having a pressure content product of less than 200 bar*l need not be inspected by an approved authority but may be inspected by any authorized personnel.

9.1.6.4 Registration

Based on a safety assessment analysis, the user is obliged to determine all applicable inspection intervals for the entire compressed air receiver system and its components. This has to be effected during inspection prior to commissioning of the system. Maximum intervals for interior inspection may not exceed 5 years whereas intervals for strength testing may not exceed 10 years.

Inspection delays for system components and/or the entire system are to be reported to the competent authority within six months after commissioning of the compressed air receiver system along with all applicable system related data.

9.1.6.5 Repetitive inspections

Any compressed air receiver systems and components thereof which are subject to inspection control may be required to be inspected by an approved authority for proper functioning and operation. Said inspections include technical testing of the system itself in conformity with the respective inspection rules as well as a proper functioning check.

As regards compressed air receivers with a pressure content product exceeding 1.000 bar*l, such inspections are to be repeated at max. 5 years intervals for interior inspection, and at max. 10 years intervals for strength testing.

Compressed air receivers having a pressure content product of less than 1.000 bar*l my be inspected by any authorized personnel.

Repetitive inspections are to include the following steps:

Interior inspection (max. 5 years intervals)

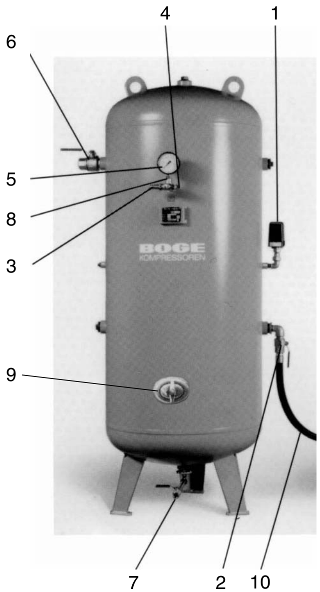
Disconnect compressed air receiver from network to make sure that it is not under pressure. Open inspection aperture and thoroughly clean inside of receiver. Walls must be metallically clean. After examination of the interior condition of the receiver a certificate of proper functioning is issued by the inspector.

Pressure test (max. 10 years intervals)

Disconnect compressed air receiver from network to make sure that it is not under pressure. Unscrew all fittings and plug any openings prior to completely filling receiver with water and connecting hand pump for pressure test. Use hand pump to generate 1.43 or 1.5 times the operating pressure, depending on receiver, before receiver is checked for leakage by the inspector.

As regards subject exterior and interior inspections, similar and equivalent methods may be employed for examination whereas with view to strength testing, equivalent non-destructive methods may be implemented in lieu of static pressure tests if, because of the design or the mode of operation of the receiver, such strength tests apparently do not fit the purpose.

9.1.7 Fittings on the compressed air receiver



1 = Pressure switch
2 = Non-return valve or ball shut-off valve
3 = Safety valve
4 = Control flange
5 = Pressure gauge
6 = Ball shut-off valve
7 = Condensate drain
8 = Mounting for fittings
9 = Inspection aperture
10 = High pressure hose

Fig. 9.3:
Compressed air receiver with fittings

The compressed air receiver is not simply a naked steel container. It needs a number of fittings to allow it to operate properly and assure the required safety.

– Pressure switch.
  The switch is for controlling the compressor.
– Non-return valve.
  A non-return valve must always be installed in the supply line from the compressor to the receiver. With piston compressors it prevents compressed air flowing back into the compressor during breaks in operation. With screw compressors the valve is integrated in the system.
– Safety valve.
  The installation of a safety valve on compressed air receivers is required by law. If the internal pressure of the receiver $\mathbf{p_N}$ (network pressure) rises 10 % ove the nominal pressure, the safety valve opens and blows out the excess pressure.
– Control flange.
  The control flange with aperture is used by the inspection authorities to connect a calibrated manometer for the pressure test.
– Pressure gauge.
  The manometer shows the internal pressure of the receiver.
– Ball shut-off valve.
  The ball shut-off valve isolates the receiver from the pneumatic system or the compressor.
– Condensate drain.
  Condensate precipitates inside the receiver and therefore it requires an appropriate connection for the condensate collector.
– Inspection aperture.
  The inspection aperture can take the form of a socket end or hand-hole flange. It is used to check and clean the inside of the receiver. The minimum size of the aperture is prescribed by law.
– High pressure hose.
  The high pressure hose connects the receiver with the compressor. It is used instead of a pipe so as not to transmit vibration from the compressor to the pneumatic system and to correct size deviations on connection to the system.

The pressure switch, high pressure hose and non-return valve are not typical fittings for compressed air receivers. But it it is sensible to have them installed.

9.1.7.1 Safety valve

Fig. 9.4:
Safety valve on combined compressed air-oil receiver of an oil-injection cooled screw compressor

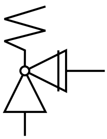


Fig. 9.5:
Diagram symbol for a safety valve

The installation of a safety valve on compressed air receivers is prescribed by law.

If the internal pressure of the receiver $\mathbf{p_N}$ (network pressure) rises to the maximum operating pressure of the tank (e.g., the maximum compressor pressure 10 bar, tank operating pressure 11 bar), the safety valve must slowly open.

If the system pressure rises to 1.1 times the nominal pressure (e.g., tank pressure 11 bar, safety valve 12.1 bar), the safety valve must open fully and blow off the excess pressure. Care must be taken that the cross-section of the outlet aperture of the safety valve is of a size that allows the entire output of all connected compressors to be blown off without the pressure in the receiver rising.

When an existing pneumatic system is extended at a later date the number of compressors increases. An appropriate upgrading of the safety valve can easily be overlooked when this happens. If the safety valve is no longer able to blow off the entire output of the compressors the operating pressure in the receiver will rise. This can cause the receiver to explode in extreme cases.

Safety inspection

The safety valve must be checked every time a compressor station is extended so as not to have a valve with too low a capacity.

The mains connection to the receiver must be shut off. The press switches must then be bridged, so that the compressors can no longer switch off automatically.

The pressure in the receiver rises until the safety valve switches. The receiver pressure must not exceed 1.1 times the limit (e.g., receiver pressure 11 bar, safety valve 12.1 bar). If this does happen, the safety valve is below par and must be replaced.

9.2 The compressed air circuit

A central compressed air supply needs a pipeline circuit to deliver compressed air to the individual devices. The circuit must meet various conditions in order to guarantee reliable and economical operation of the devices:

– Adequate volume flow.
  Each device in the circuit must be supplied with the required volume flow at all times.
– The necessary working pressure.
  Each device in the circuit must have the necessary air pressure at all times.
– Quality of compressed air.
  Each device in the circuit must have compressed air of the required quality at all times.
– Low pressure loss.
  The pressure loss in the circuit must be as low as possible for economic reasons.
– Secure operation.
  The supply of compressed air should be guaranteed as far as possible. If lines are damaged, repair and maintenance work must not put the entire circuit out of use.
– Safety rules.
  The relevant safety rules must be followed at all times in order to prevent accidents and the resulting rights of recourse.

9.2.1 The structure of a compressed air circuit

A pipeline circuit is made up of individual sections. This allows an ideal connection to be made between the compressor and dependent devices.

9.2.1.1 The main line

The main line connects the compressor station with the compressed air treatment and the compressed air receiver. Distribution lines are connected to the main line. The main line must be of a size that allows the entire output of the compressor station to be delivered now and in the near future, and with the minimum loss of pressure.

The pressure loss **Δp** in the main line should be no higher than 0.04 bar.

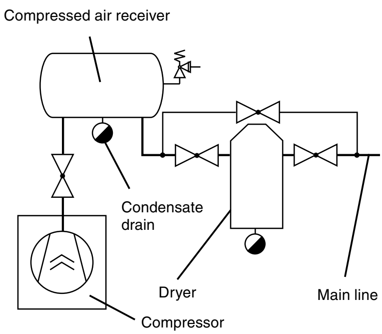
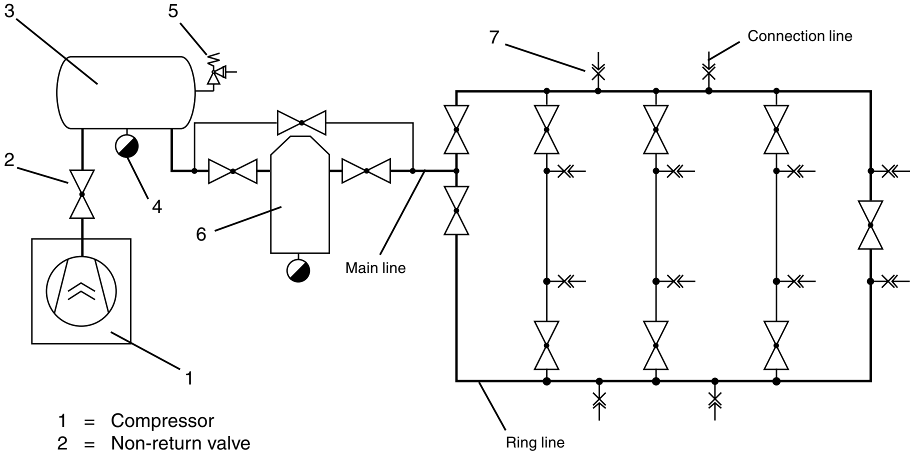


Fig. 9.6:
Main line of a compressed air circuit

9.2.1.2 The distribution line- ring line

The distribution lines are laid through the entire operation and bring compressed air to the devices. They should always take the form of a ring line wherever possible. This increases the economy and security of operation of the line as a whole.

Pressure loss **Δp** in the distribution lines should be no higher than 0.03.



1 = Compressor
2 = Non-return valve
3 = Compressed air receiver
4 = Condensate drain
5 = Safety valve
6 = Compressed air dryer
7 = Compressed air connections

Fig. 9.7:
Compressed air supply with ring line

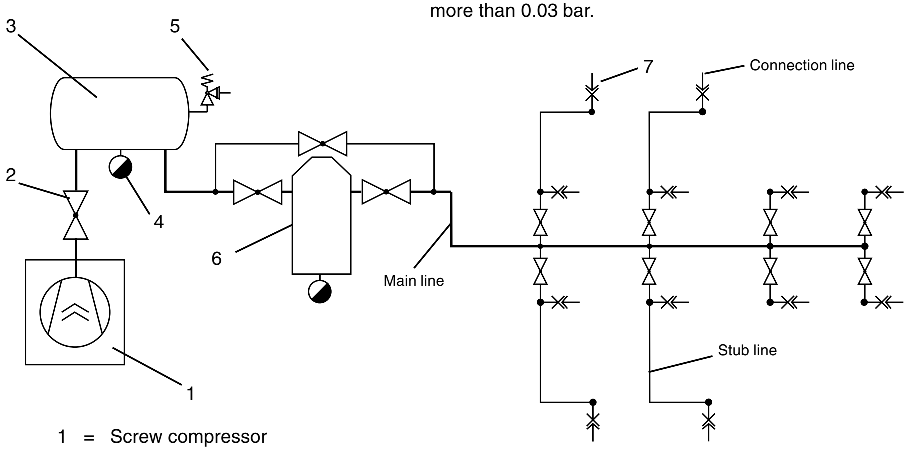
A ring line forms a closed distribution ring. It is possible to isolate individual sections of the network without interrupting the supply of compressed air to other areas. This provides assurance that compressed air will be available for most devices, even when servicing, repairs and extension work is being carried out.

If the compressed air is supplied through a distribution ring, the compressed air has a shorter route to travel than with stub lines. This means that lower pressure loss **Δp** is needed. When dimensioning the ring line one can calculate with half the flow pipe length and half the volume flow allowing smaller pipe dimensions.

9.2.1.3 The distribution line- stub line

The distribution lines are laid through the entire operation and bring compressed air close to the devices. They can also take the form of a stub line.

The pressure loss **Δp** in the distribution lines should be no more than 0.03 bar.



1 = Screw compressor
2 = Non-return valve
3 = Compressed air receiver
4 = Condensate drain
5 = Safety valve
6 = Compressed air dryer
7 = Compressed air connections

Fig. 9.8:
Compressed air supply with stub line

Stub lines branch off from larger distribution lines or the main line and end at the consumer device. Outlying consumers can be supplied through stub lines. It is also possible to supply a complete compressed air system with stub lines. They have the advantage of needing less material than ring lines. But they also have the disadvantage that they must be of larger size than ring lines and frequently cause high pressure losses.

Stub lines should always have a non-return valve which can isolate them from the system. This makes servicing and repair work easier.

9.2.1.4 The connection line

The connection lines come from the distribution lines. They supply consumer devices with compressed air. Since the devices operate with different pressures it is normally necessary to install a service unit with a pressure regulator in front of the device. The network pressure is reduced to the working pressure of the device by the regulator. Service units comprising filters, separators, regulators and oilers are not needed if the compressed air is pre-treated.
The pressure loss **Δp** in the connection lines should be no higher than 0,03 bar.

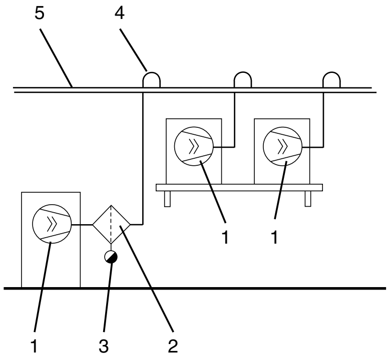
Note

For industrial applications the recommended pipe size is DN 25 (1"). This size has next to no cost disadvantages compared with smaller sizes and nearly always guarantees a reliable supply of compressed air. Consumer devices requiring up to 1800 l/min can be supplied through line lengths of up to 10 m with hardly any pressure loss.

9.2.1.5 Connecting to a collective line with multiple systems

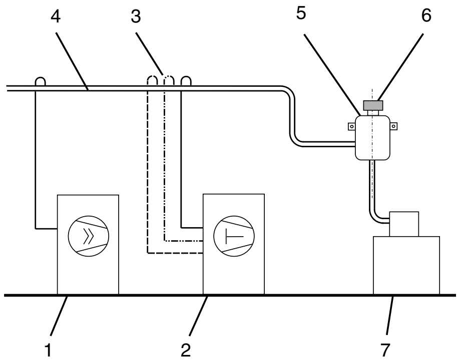
Attention must be paid to the following points when connecting several compressors to a common (collective) line.

Compressed air



1 = Screw compressor
2 = Water separator
3 = Condensate drain
4 = Connection line
5 = Collection line

Condensate



1 = Screw compressor
2 = Piston compressor
3 = Connection line
4 = Collection line
5 = Expansion vessel
6 = Vent silencer
7 = Oil-water-separator

Fig. 9.9:
Collective lines

Compressed air and condensate collection lines

1. Collection line with gradient.
   The line must be laid with a gradient of approx. 1.5 - 2 % in the direction of flow.
2. Connection line from above.
   The connection line must be connected to the collective line from above.

Compressed air collective lines

3. Water separator on longer rising lines.
   On longer lines that rise to a collective line a water separator with automatic drainage must be installed after the compressor in order to catch the water flowing back.

Vent collective lines

Points 1 and 2 also apply when vent lines are brought together in collective lines.
Vent collective lines must also have an expansion vessel and vent silencer installed.

9.3 Tips for planning pipe systems

9.3.1 General planning tips

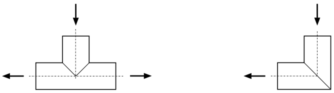


Fig. 9.10:
Unfavourable flow conditions: T and knee-piece

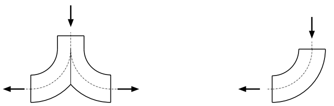


Fig. 9.11:
Favourable flow conditions: Y-tube and curved pipe

Compressed air lines should be straight wherever possible. On corners that can not be avoided, **do not use** knee and T-pieces. Long curves and Y-pieces provide better flow conditions and therefore less pressure loss Dp. Abrupt changes in the diameter of the line should also be avoided due to the high loss of pressure this causes.

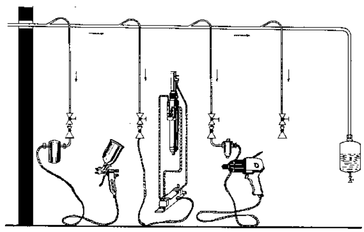
Large pipe systems should be subdivided into several sections, each of which should be equipped with a non-return valve. It is important to be able to isolate parts of the system, particularly for inspections, repairs and conversions.

In some situations it may be of benefit to have a second compressor to supply the system from a different point. This shortens the distance the compressed air has to travel. As a result, the pressure loss **Δp** is lower.

Main lines and large distribution lines should be welded with V-seams. This means there are no sharp edges and points inside the pipes. There is therefore less resistance in the pipes and the burden on filters and tools caused by detached particles of welded metal is reduced.

9.3.2 Pipeline without compressed air dryer

Pipeline with 1.5 - 2 ‰ gradient



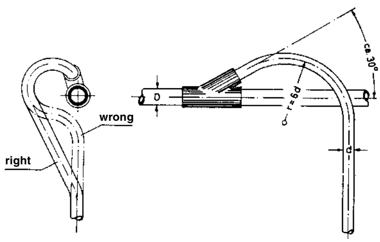


Fig. 9.12:
Examples of correctly laid piping

Compression causes the water in the air to form droplets (condensate). If the compressed air is not pre-processed by a compressed air dryer, water must be expected in the entire pipeline network.

In this situation there are various guidelines to be followed when installing the pipeline, in order to prevent damage to consumer devices.

- Temperature gradients.
  Where possible, the compressed air lines should be laid so that the air does not cool down when flowing through. The air should be heated gradually. If the absolute humidity is constant, the relative humidity will then fall. Condensate will then be unable to form.
- Pipelines with gradients.
  The pipelines must be laid with a gradient of approx. 1.5 - 2 % in the direction of flow. The condensed water in the pipeline will then collect at the lowest point of the line.
- Vertical main line.
  The main line directly behind the compressed air receiver should rise vertically. The condensate that occurs when cooling takes place can then flow back into the receiver.
- Condensate drain.
  Condensate drains must be installed at the lowest points of the system in order to drain off the condensate.
- Connection lines.
  The connection lines must branch off upwards in the direction of flow. The pipeline here must be as straight as possible to avoid unnecessary pressure loss.
- Fittings.
  A service unit with filter, water separator and pressure reduction valve should always be installed. A compressed air lubricator may also be needed, depending on the application.

9.3.3 Pipeline system with compressed air dryer

If there is a compressed air dryer with an appropriate filter installed in the system, many of the measures taken against condensate can be dispensed with.

– Pipelines.
  The lines can be laid horizontally because there is almost no water left in the system. The other measures concerning the way the lines are laid are also unnecessary.
– Condensate drain.
  Condensate drains are only fitted at the filters, the compressed air receiver and the dryer.
– Connection lines.
  The connection lines can be joined vertically downwards with T-pieces.
– Fittings.
  Only pressure reduction valves have to be fitted to the consumer devices. A lubricator may be required, depending on the application.

This considerably reduces the price of the installation. Sometimes, even the money saved here justifies installing a compressed air dryer.

9.4 Pressure loss Δp

Every pneumatic pipeline is resistance for compressed air in flow. This resistance is internal friction which occurs with the flow of all liquid and gaseous media. It results from the effect of force among the molecules(viscosity) of the flowing medium ad the walls of the pipeline. This is the cause of pressure loss in pipelines.

9.4.1 Type of flow

Quite apart from internal friction, the type of flow inside the line also affects pressure loss. Air can move in two completely different ways.

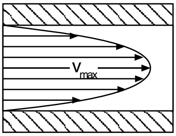


Fig. 9.13:
Flow and speed development with laminar flow

Laminar flow

Laminar flow is even-layered flow. The individual molecules of the compressed air move in parallel, adjacently flowing layers. This type of flow has two main properties:

- low pressure loss.
- low heat transition.

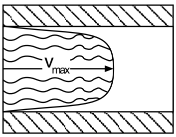


Fig. 9.14:
Image of flow and speed with turbulent flow

Turbulent flow

Turbulent flow is whirly and uneven. The axially directed flow is surrounded by constantly changing additional movement at all points. The paths of flow all have an effect on each other and form small whirls. This type of flow has two main properties:

- high pressure loss.
- high heat transition.

9.4.2 The Reynolds number Re

The type of flow can be defined using the Reynolds number **Re**. This gives the criterion for laminar and turbulent flow. The Reynolds number **Re** is influenced by various factors:

- The kinematic viscosity of the compressed air.
- The mean speed of the compressed air.
- The inside diameter of the pipe.

The flow in the pipeline remains laminar until what is known as the critical Reynolds number $\mathbf{Re}_{crit}$ is exceeded. It then takes on the condition of unevenness and turbulence.

Note

The high flow speeds that lead to $\mathbf{Re}_{crit}$ being exceeded do not normally occur in pneumatic networks. The prevailing flow in pneumatic networks is laminar. Turbulent flow only occurs at points where there are massive flow disturbances.

The speed of flow in compressed air lines must not exceed 20 m/s, since noise and turbulent flow will otherwise occur.

9.4.3 Pressure loss in the pipe system

Each change in the line hinders the flow of compressed air within it. The laminar flow is disturbed and higher pressure loss results.

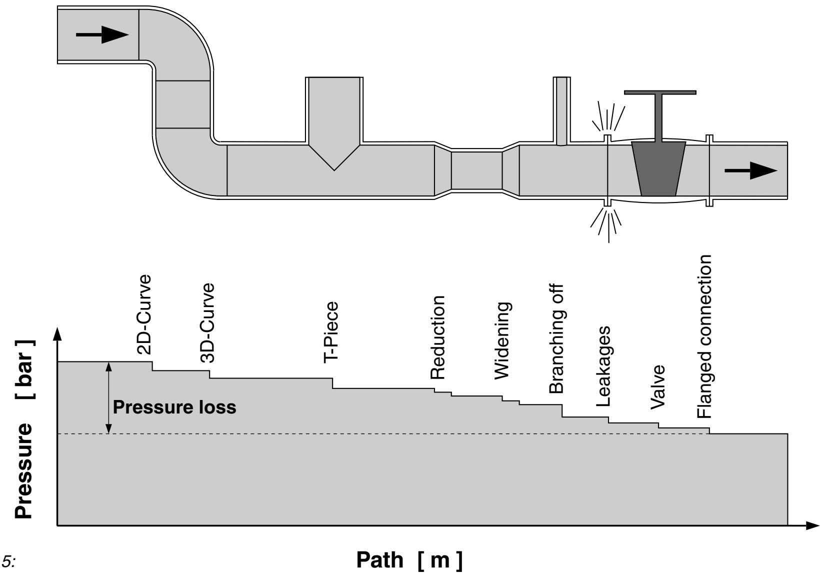


Fig. 9.15:
Pressure loss in a pipeline

The amount of pressure lost is influenced by several components and circumstances of the network:

- length of pipe.
- clear inside diameter of the pipe.
- pressure in the pipe network.
- branches and bends in the pipe.
- narrowing and widening.
- valves.
- fittings and connections
- filters and dryer.
- leakage points.
- surface quality of the pipelines.

These factors must be taken into account when planning the system, otherwise increased pressure loss will occur.

9.5 Dimensioning pipelines

Correct dimensioning of the pipes in a system is of great importance for economical operation. Pipes with too small a diameter cause high losses of pressure. These losses must be compensated for by high compression in order to guarantee the performance of consumer devices.

The main factors influencing the ideal inside diameter $\mathbf{d_i}$ of the pipe are:

- Volume flow $\dot{\mathbf{V}}$.
  The maximum throughput of air should be assumed when determining $\mathbf{d_i}$. Increased pressure loss has a greater impact when the requirement for compressed air is at a maximum.
- Effective flow length of pipeline.
  The length of the pipeline should be determined as accurately as possible. Fittings and bends are unavoidable in pipeline systems. When determining the effective flow length of the pipeline these must be taken into account as an equivalent section of pipe.
- Operating pressure.
  When determining $\mathbf{d_i}$ the compressor cut-out pressure $\mathbf{p_{max}}$ is to be assumed. At maximum pressure the pressure drop $\mathbf{\Delta p}$ is also highest.

9.5.1 Maximum pressure drop Δp

The pressure drop $\mathbf{\Delta p}$ in a pipeline with a maximum pressure $\mathbf{p_{max}}$ of 8 bar_{op} and above should not exceed a certain total loss by the time it reaches the consumer device:

- Pipe system **$\Delta p \leq 0.1$ bar**

The following values are recommended for the individual sections of the system:

- Main line **$\Delta p \leq 0.04$ bar**
- Distribution line **$\Delta p \leq 0.04$ bar**
- Connection line **$\Delta p \leq 0.03$ bar**

In pipe systems with lower maximum pressures (e.g., 3 bar_{op}) a pressure loss of 0,1 bar is higher in relative terms than in an 8 bar_{op} system. In this case, a different value is recommended for the system as a whole:

- Pipe system **$\Delta p \leq 1{,}5\ \%\ p_{max}$**

9.5.2 Nominal width of pipelines Comparison [DN – Inch]

Medium-weight threaded pipes made of standard structural steel (DIN 17100), which are often used for pipe systems, are made according to the DIN 2440 standard. This standard prescribes certain graduations of nominal width (inside diameter $\mathbf{d}_i$) and certain designations. For this reason, fittings and pipes are only available in the corresponding sizes.

The graduations of nominal diameter also apply for other pipe materials and standardisations.

The standard nominal widths must always adhered to when dimensioning pipelines. Other nominal widths are only available if specially made and are disproportionately expensive.

The following table contains standard graduations in DN (Diameter Nominal) mm and inches, and the most important basic data for pipes according to DIN 2440:

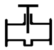
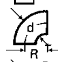
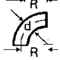
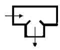
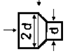
Nominal pipe width acc. to DIN 2440		Outside diameter	Inside diameter	Inside cross-section	Wall thickness
[Inches]	[DN]	[mm]	[mm]	[cm2]	[mm]
1/8"	6	10.2	6.2	0.30	2.00
1/4"	8	13.5	8.8	0.61	2.35
3/8"	10	17.2	12.5	1.22	2.35
1/2"	15	21.3	16.0	2.00	2.65
3/4"	20	26.9	21.6	3.67	2.65
1"	25	33.7	27.2	5.82	3.25
1 1/4"	32	42.4	35.9	10.15	3.25
1 1/2"	40	48.3	41.8	13.80	3.25
2"	50	60.3	53.0	22.10	3.65
2 1/2"	65	76.1	68.8	37.20	3.65
3"	80	88.9	80.8	50.70	4.05
4"	100	114.3	105.3	87.00	4.50
5"	125	139.7	130.0	133.50	4.85
6"	150	165.1	155.4	190.00	4.85

9.5.3 Equivalent pipe length

A major factor in dimensioning the inside diameter of a pipe $\mathbf{d_i}$ is the pipe length. Pipelines are not only made up of straight sections of pipe, the flow resistance of which can quickly be deduced. Installed bends, valves and other fittings considerably increase flow resistance inside the pipeline. This is the reason that the effective pipe length **L** must be determined, taking into account the fittings and bends.

For simplification, the flow resistance values of various fittings and bends have been converted into equivalent pipe lengths.

The following table gives the equivalent pipe length in dependency on pipe nominal width and the fitting:

Fittings		**Equivalent Pipe Length [m]**						
		Pipe and Fitting Nominal Width [DN]						
		DN 25	**DN 40**	**DN 50**	**DN 80**	**DN 100**	**DN 125**	**DN 150**
Check valve		8	10	15	25	30	50	60
Diaphragm valve		1.2	2.0	3.0	4.5	6	8	10
Gate valve		0.3	0.5	0.7	1.0	1.5	2.0	2.5
Knee bend 90°		1.5	2.5	3.5	5	7	10	15
Bend 90° R = d		0.3	0.5	0.6	1.0	1.5	2.0	2.5
Bend 90° R = 2d		0.15	0.25	0.3	0.5	0.8	1.0	1.5
T-Piece		2	3	4	7	10	15	20
Reduction piece D = 2d		0.5	0.7	1.0	2.0	2.5	3.5	4.0

These values must be added to the actual pipe length to obtain the effective pipe length **L**.

Note

Complete information about fittings and bends are not generally available at the start of planning a pipeline system. The effective pipe length **L** is therefore calculated by multiplying the straight pipe length by 1.6.

9.5.4 Determining the inside diameter d_i of the pipe by calculation

The following approach formula can be used to dimension the inside diameter of the pipe. It assumes the maximum operating pressure $\mathbf{p_{max}}$ (compressor cutout pressure), the maximum volume flow $\dot{\mathbf{V}}$ (required output $\mathbf{L_B}$) and the effective pipe length **L**. **Δp** is the target pressure loss.

$$d_i = \sqrt[5]{\frac{1{,}6 \times 10^3 \times \dot{V}^{1{,}85} \times L}{10^{10} \times \Delta p \times p_{max}}}$$

d_i	=	Inside diameter of pipeline	[m]
$\dot{V}$	=	Total volume flow	[m^3/s]
L	=	Effective pipe length	[m]
Δp	=	Target pressure loss	[bar]
p_{max}	=	Compressor cut-out pressure	[bar_{abs}]

Example

The inside pipe diameter $\mathbf{d_i}$ of a pneumatic connection line with a target pressure loss **Δp** of 0,1 bar is to be determined using the approach formula. The maximum operating pressure $\mathbf{p_{max}}$ (compressor cutout pressure) is 8 bar_{abs}. A volume flow $\dot{\mathbf{V}}$ of 2 m^3/min will flow through pipeline with an approximate length of 200 m.

$\dot{V}$	=	2	m^3/min	=	0,033 m^3/s
L	=	200	m		
Δp	=	0,1	bar		
p_{max}	=	8	bar_{abs}		

$$d_i = \sqrt[5]{\frac{1{,}6 \times 10^3 \times 0{,}033^{1{,}85} \times 200}{10^{10} \times 0{,}1 \times 8}}$$

$$d_i = 0{,}037 \text{ m} = 37 \text{ mm}$$

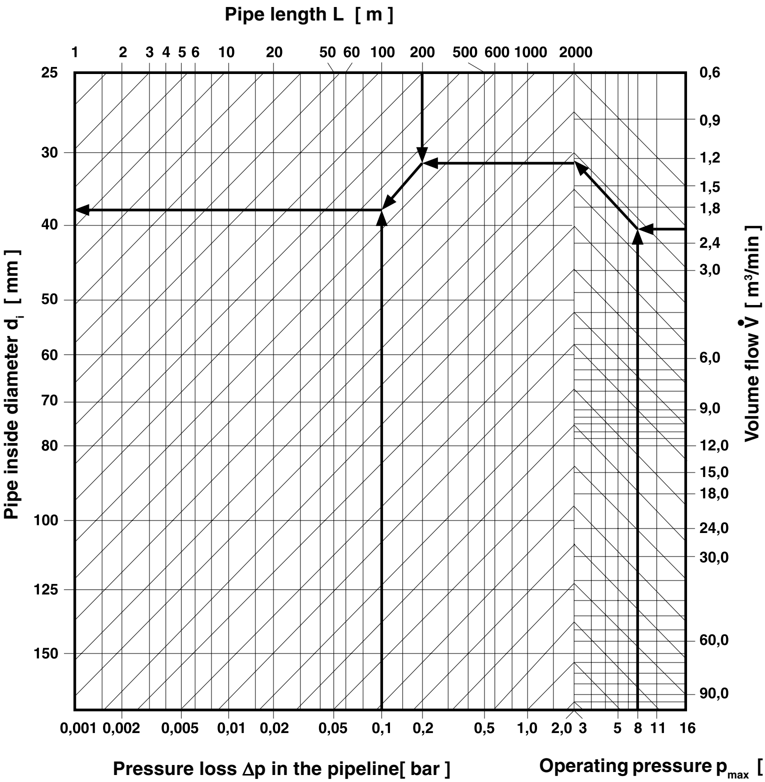
Nominal width selected: DN 40

The inside diameters of pipes are standardised in certain sizes. But it is rare to find a standard nominal width that matches the calculated inside diameter. In such cases the next largest standard nominal width is taken.

9.5.5 Determining the inside diameter of the pipe d_i by graphics

The pipe inside diameter $\mathbf{d_i}$ can be determined easier and faster with a nomogramme than by calculation. The major influencing factors are the same with calculation method as with the graphical method.

Start by reading the intersection of the volume flow $\dot{V}$ and the operating pressure $\mathbf{p_{max}}$. Proceed by following the thick line in the example in the direction of the arrow.



Example

Volume flow	$\dot{V}$	=	2	m³/min
Effective pipe length	L	=	200	m
Pressure loss	Δp	=	0,1	bar
Operating pressure	p_{max}	=	8	bar_{abs}
Pipe inside diameter	$\mathbf{d_i}$	**=**	**app. 38 mm**	

The nominal width selected for the pipeline is DN 40

9.5.6 Determining the inside diameter of the pipe d_i with the aid of a bar graph

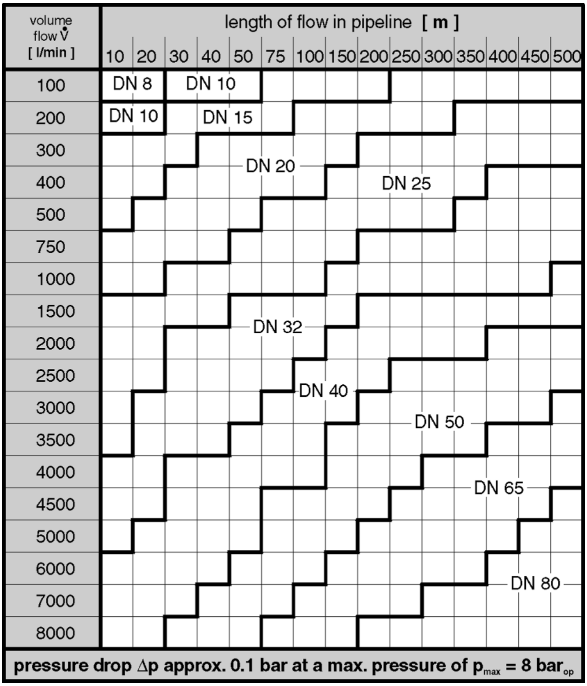
The third and simplest method of determining the pipe inside diameter $\mathbf{d_i}$ is the bar graph. However, this method is very limited in application. Two conditions must be met for the bar graph method to be used:

– A maximum pressure $\mathbf{p_{max}}$ in the network of 8 bar_{op}.

– A target pressure loss $\mathbf{\Delta p}$ of 0,1 bar.

The bar chart is very easy to use:

Take the determined maximum volume flow $\dot{V}$ and the effective pipe length and find the respective line or column in the graph. The resulting intersection indicates the correct pipe nominal width to meet the requirements.



Example

Pressure loss	Δp	=	0,1	bar
Operating pressure	p_{max}	=	8	bar_{op}
Effective pipe length	L	=	200	m
Volume flow	$\dot{V}$	=	2000	l/min

The nominal width of the pipe obtained is DN 40

9.6 Choosing the material for pipelines

System pipelines are normally made of steel, non-ferrous metal or plastic. They must meet various criteria, which limits the choice of material for some applications:

- Protection against corrosion.
  The question of resistance to corrosion is always a prime consideration unless the compressed air is dried in a pre-treatment unit. The pipes must not rust through over the course of time.
- Maximum operating temperature.
  Some materials lose tensile strength at high temperatures and become brittle at low temperatures.
- Maximum operating pressure.
  The maximum operating pressure drops with increasing thermal stress.
- Low pressure loss.
  Low pressure loss is obtained by high surface quality on the inside of the pipe.
- Low-cost installation.
  Installation prices can be reduced by a multitude of pre-shaped parts, fast and easy installation and cheap material.

9.6.1 Threaded pipes

Steel threaded pipes compliant with DIN 2440, DIN 2441 and DIN 2442 (medium-weight and heavyweight versions) are in widespread use in pneumatic systems. They are used particularly in small and medium-sized distribution and connection lines. Threaded pipes are used everywhere where the demands on the quality of compressed air are not high. They are available in black and galvanised metal.

- Size DN 6 - DN 150
- Maximum operating pressure max. 10 - 80 bar_{op}
- Maximum operating temperature 120°C

Advantages

Threaded pipes are inexpensive and quickly installed. There are many different and useful shaped parts and fittings to use with them. The joints can be disconnected and the individual parts reused.

Disadvantages

Threaded pipes have a high flow resistance and the joints tend to leak over time. An experienced fitter is needed to install them. Ungalvanised threaded pipes should not be used in networks without a dryer, because they corrode.

9.6.2 Seamless steel pipes

Seamless mild steel pipes compliant with DIN 2448 are chiefly mainly used in main and distribution lines with medium and large pipe diameters. They are available in black and galvanised finishes.

– Sizes 10.2 - 558,8 mm
– Maximum operating pressure max. 12.5 - 25 bar_{op}
– Maximum operating temperature 120°C

Advantages

Seamless mild steel pipes are available in sizes up to 558,8 mm. They are completely airtight if properly laid. Leakage is therefore practically zero. The pipes are cheap, and there are relatively many shaped parts to choose from.

Disadvantages

An experienced fitter is needed to lay seamless, mild steel pipes because they must be welded and flanged. Ungalvanised mild steel pipes should not be used in networks without a dryer, because they corrode.

9.6.3 Stainless steel pipes

Stainless steel pipes compliant with DIN 2462 and DIN 2463 are only used in pneumatic networks requiring the highest quality. They are also often used in the „wet“ sections of a conventional system between the compressor and the dryer.

– Sizes 6 - 273 mm
– Maximum operating pressure max. 80 bar_{op}, part. higher
– Maximum operating temperature 120°C

Advantages

Stainless steel pipes are completely corrosion-proof and have only low flow resistance (low pressure loss). They are absolutely airtight if properly laid. Leakage is therefore practically zero.

Disadvantages

An experienced fitter is needed to lay seamless, stainless steel pipes because they must be welded and flanged. The pipes are very expensive and the availability of shaped parts is limited.

9.6.4 Copper pipes

Copper pipes conforming to DIN 1786 and DIN 1754 are used for small and medium pipes as process control lines. The seamless pipes are available in hard, semi-hard and soft qualities.

– Sizes: soft 6 - 22 mm; semi-hard 6 - 54 mm; hard 54 - 131 mm
– Maximum operating pressure: max. 16 - 140 bar_{op}
– Maximum operating temperature: 100°C

Advantages

Copper pipes are available in long sections, they can be bent if the diameter is small, and they are easy to work with. It is therefore possible to use one piece for longer sections of the network. This reduces the number of joints. The occurrence of leaks is also lower.

Copper pipes are corrosion-proof and pressure loss is low due to the smooth surface of the inside walls.

Disadvantages

An experienced fitter is needed to install copper pipes since fittings are normally soldered to them. The joints can not be disconnected.

The material is expensive, but there are many shaped parts to choose from because copper pipes are also used in the sanitary area.

If the lines are longer, the expansion of copper due to heat must be taken into account. The coefficient of length expansion for copper is greater than that for steel.

If the compressed air contains moisture, particles of copper may form local galvanic elements in subsequent steel piping, leading to pitting. Copper vitriol can also arise.

9.6.5 Plastic pipes

There are plastic pipes as pipe systems from various makers and in various materials. There are also polyamide pipes for high pressures and polyethylene pipes for large diameters. This means that there are plastic pipes with the appropriate properties for almost every area of application. For this reason it is difficult to generally apply information about sizes, operating pressures and temperatures.

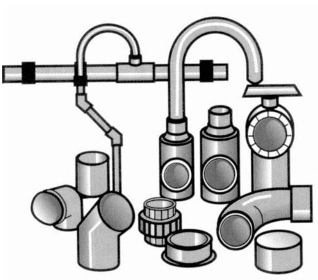


Fig. 9.16:
An assortment of plastic shaped parts and fittings

Advantages

Because plastic pipes do not corrode, there is no need for any protective surface material. They are up to 80 % lighter than steel. This simplifies installation and there are fewer demands placed on the pipefittings.

The inside surface is very smooth. Flow resistance is low (low pressure loss) and deposits such as calcium and rust etc. have practically no chance to build up. Plastic pipes are usually harmless from a toxicological and hygienic standpoint.

PVC pipe systems and the like have a large number of shaped parts and fittings available for them. Installation is very easy. The pipe sections are fitted together and given an airtight seal with special adhesive. No special knowledge is necessary for installation. Pressure loss and leakage is generally very low in plastic piping.

Disadvantages

The low-cost PVC pipe systems have a maximum operating pressure of only 12.5 bar at 25°C. It must also be carefully noted that the maximum operating pressure of these plastic pipes drops heavily if the temperature is increased. For this reason plastic pipes may not be used in the hot areas of a compressor station and must be protected from direct sunlight.

Plastic pipes have large coefficients of linear expansion and their mechanical stability is not particularly high.

Resistance to certain condensates and types of oil is not always guaranteed with some plastics. The composition of condensates in the network must therefore be checked beforehand.

Plastic pipes are not made in large quantities for high pressures or large diameters. This makes them expensive and the number of shaped parts available is limited. An experienced plastic welder is needed to install these pipes.

9.7 Marking pipelines

Pipelines must be marked clearly according to the type of medium they contain according to German law and DIN 2403. Unambiguous marking also eases correct installation, the planning of extensions and firefighting.

The marking should indicate possible dangers in order to avert accidents and physical injury. Appropriate marking also makes it easier to follow pipelines in a complicated network. For this reason, the direction of flow of the medium must always be indicated.

Pipes are marked with ID numbers (groups) and colours, both of which are defined in DIN 2403.

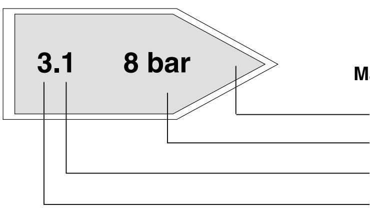
Medium	Group ID number	Colour	Colour number
Air	3	grey	RAL 7001
Water	1	green	RAL 6018
Combustible liquids	8	brown	RAL 8001
Gas	4/5	yellow	RAL 1013
Water steam	2	red	RAL 3003
Acids	6	orange	RAL 2000
Alkalis	7	violet	RAL 4001
Oxygen	0	blue	RAL 5015

Fig. 9.17:
Marking plates with cleartext

Colour markings and markings in writing must be applied at certain points:

– In writing at the start of the line.
– In writing at the end of the line.
– In writing at branches.
– In writing where the line passes through a wall.
– In writing at fittings and distribution points.
– In colour by way of rings or continuous paint for the entire length of the line.

Marking plates



Direction of flow.
Colour matches colour code for medium.
Sub-group number (different line networks).
Group number of medium.

Fig. 9.18:
Marking plates with ID numbers

10. The Installation Room

The installation room of a compressor must satisfy a number of conditions for correct operation to be assured. When considering the significance of a well-planned and well-kept installation room it is important to know that around 2/3 of all compressor malfunctions are caused by faulty installation, inadequate ventilation and a lack of servicing.

The general rules for accident prevention and environmental protection must also be adhered to.

10.1 Cooling the compressor

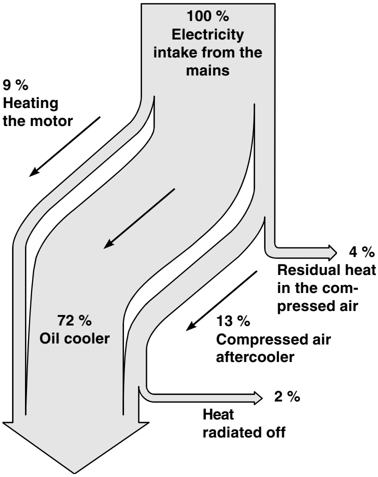


Fig. 10.1:
Heat distribution in a screw-type compressor with oil injection cooling.

When designing a compressor station it must be remembered that the compression process inside the compressor generates a large amount of waste heat. The main principle of thermo-dynamics applies, which states that the entire electrical power intake of the compressor is converted into heat.

The waste heat must be extracted reliably since there may otherwise be an accumulation of heat in the compressor. If the temperature inside the compressor is too high for too long it can lead to mechanical damage in the compressor stage and the drive motor.

The required cooling medium (air or water) can be supplied in two ways:

– Air-cooling.
  Air-cooling is the most common cooling method for all types of compressor. When it is used, ventilation of the installation room is of particular importance. It must be well planned and implemented. If not, thermal problems with the compressor are bound to occur.
– Water-cooling.
  Water cooling may be necessary in larger compressors if the heat can not be properly extracted by air-cooling. Water-cooling places fewer demands on ventilation inside the installation room.

This chapter deals primarily with the requirements and rules applying to installation rooms for air-cooled compressors. With the exception of information concerning ventilation, the material in this chapter can be used equally for water-cooled compressors.

10.2 Compressor installation

When installing compressors and the other components of a compressor station there are certain conditions to observe which, if not complied with, may lead to malfunctions. There are also certain accident prevention and environmental protection rules to be followed.

10.2.1 General information regarding the installation room

The installation room should be clean, free of dust, dry and cool. Strong sunlight must not be allowed to enter. The room should be located on the north side of a building wherever possible, or in a well-ventilated basement.

There should be no heat-emitting pipes or assemblies in the installation room of a compressor. If this can not be avoided, the pipes and assemblies must be adequately insulated.

Easy accessibility and good lighting should be provided for servicing work and periodic inspections of the compressed air receivers.

A compressor installation room must always be properly ventilated to prevent the ambient temperature from exceeding the maximum admissible levels.

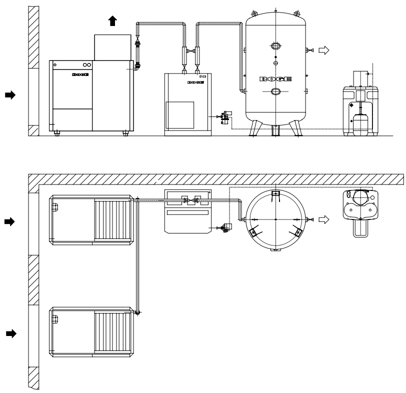


Fig. 10.2:
Compressor station with 2 screw-type compressors, refrigerant air dryer, compressed air receiver and oil/water separator.

10.2.2 Admissible ambient temperature

Compressors operate ideally at ambient temperatures between +20° and +25°C. The following ambient temperatures apply for piston and screw-type compressors:

- Minimum +5°C.
  If the temperature falls below +5°C, pipelines and valves can ice up. This can cause the compressor to malfunction. Screw-type compressors switch off automatically if the temperature is below the minimum admissible compression temperature.
  An additional anti-freeze facility allows ambient temperatures down to -10°C.
- Maximum +40°C.
  Maximum +35°C with sound-insulated piston-type compressors.
  If the ambient temperature rises above the maximum level, the compressed air outlet temperature may exceed the maximum statutory level. The quality of the compressed air deteriorates, the components of the compressor are subjected to more strain, and the servicing intervals are shorter. Screw-type compressors switch off automatically if the temperature is above the maximum admissible compression temperature.

10.2.3 Fire safety rules for installation rooms

The following rules apply for rooms where compressors with oil injection cooling are to be installed:

– The room must have special fire protection if the compressor motor rating is **over 40 kW**.
– Compressors with a motor rating of **over 100 kW** must be installed in a separate fire-protected room.

Requirements for fire-protected installation rooms:

– The walls, ceilings, floors and doors must be **Fire safety category F30** or better.
– No inflammable liquids may be stored in the installation room.
– The floor area around the compressor must not be made of combustible material.
– Leaking oil must not be allowed to spread on the floor.
– There must be no inflammable substances within a radius of at least three metres from the compressor.
– No combustible system parts, such as cable lines, may be laid over the compressor.

10.2.4 Disposal of condensate

The inducted air contains water in the form of vapour which turns into condensate during compression. This condensate contains oil. It may not be allowed into the public sewage network without being processed.

Always follow the appropriate drainage rules set by the local authority.

BOGE recommends the ÖWAMAT for processing the condensate. The purified water can be drained into the public sewage lines. The oil is caught in a catch pan and must then be disposed of in a responsible manner.

10.2.5 Compressor installation instructions

When installing compressors, the following general points must be observed, regardless of ventilation:

– When installing a compressor or compressed air receiver a flat industrial floor without foundation will suffice. Special mountings are not generally needed..

– Compressors should always be located on elastic mountings. This stops vibration being transmitted to the floor, and the compressor noise being carried to other parts of the building.

– The compressor should be connected to fixed lines with a BOGE high pressure hose of approx. 0.5 m in length. This prevents vibration from the compressor being transmitted to the compressed air line and compensates inaccurate lines.

– The compressor must be fitted with paper intake filters if there is a heavy dust occurrence at the installation point. This keeps wear on the compressor to a minimum.

– Compressor units must never be covered by hoods or cladding. Measures of this type always lead to thermal problems. An exception to this is the original BOGE sound insulation hood, which is specially designed for each individual compressor.

10.2.6 The space requirement of a compressor

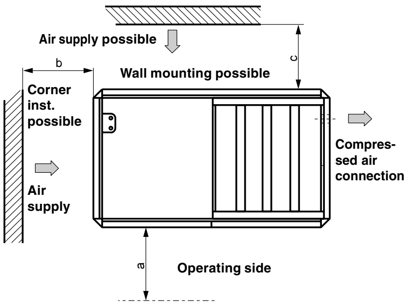


Fig. 10.3:
Space requirement plan for a sound-insulated screw-type compressor, model S 21 - S 30

A compressor requires a certain amount of space, and this depends on the construction and type of compressor concerned. From this arise compressor-specific minimum distances in all directions.

– The compressor must be installed to allow easy access for operation and servicing.

– For the cooling of a compressor to be assured, there must be a certain minimum distance between the ventilator or cooler and the neighbouring wall or other systems. If this is not provided for, the effect of the ventilator or cooler is much reduced and efficient cooling is no longer guaranteed.

– When several compressors are installed adjacently the heated cooling air of one compressor must not be used as the cooling air of another.

The minimum distances to walls and neighbouring equipment and machinery can sometimes vary greatly, depending on the types and versions of the compressors. These are to be taken from the respective operating instructions.

10.2.7 Conditions for installing compressed air receivers

Certain accident prevention rules must be followed when installing compressed air receivers.

- Compressed air receivers must be protected from external damage (e.g., falling objects).
- The receiver and its equipment must be able to be operated from a safe location.
- Safety areas and distances must be observed.
- The receiver must be safe where it stands. It must not move or tilt by the application of external force. This includes the additional weight during pressure testing! A reinforced foundation may be necessary for large compressed air receivers.
- The factory specification plate must be well visible.
- Compressed air receivers must have reasonable protection against corrosion.
- Vertical receivers are brought horizontally into the compressor rooms and then set up on their feet. The diagonal height of the receiver must therefore be taken into account in the dimensions of the room, otherwise it will be impossible to set up the receiver.

10.3 Ventilation of a compressor station

The most important requirement for operating air-cooled compressors is an adequate flow of cooling air $\dot{V}_c$. The waste heat generated by the compressor must be reliably extracted at all times. There are three different possibilities for ventilation, depending on the rooms available, and the type and model of the compressor:

- Natural ventilation.
  Ventilation through the air inlet and outlet apertures in the side walls or the ceiling by natural means i.e., without assistance from a ventilator.
- Artificial ventilation.
  Ventilation through the air inlet and outlet apertures in the side walls or the ceiling with the assistance of an outlet ventilator.
- Air inlet and outlet ducts.
  Ventilation by means of appropriate ducts, usually with the assistance of an exhaust ventilator.
- With water-cooled compressors the main heat is extracted by the cooling water. The residual heat (radiated from the motor) must be extracted by cooling air.

10.3.1 Factors influencing the flow of cooling air of a $\dot{V}_c$ of a compressor

A compressor generates a certain amount of waste heat depending on its drive rating. On air-cooled compressors this heat must be extracted by a flow of cooling air $\dot{V}_c$.

The volume of cooling air $\dot{V}_c$ is influenced by several factors as well as the drive rating of the compressor:

- Transmission heat
  A part of the heat generated is emitted as transmission heat by the walls enclosing the installation room (including the windows and doors). The constitution of the walls, the ceiling, the floor, doors and windows have a considerable influence on the flow of cooling air $\dot{V}_c$.
- Room temperature.
  The higher the temperature of the installation room, the greater the requirement for cooling air.
- Temperature gradient.
  The greater the difference **Δt** between the outside and inside temperature, the lower the requirement for cooling air.
- Room height and site.
  The greater the height and size of the room, the better the distribution of the generated heat, and the requirement for cooling air drops accordingly.

10.3.2 Definition of the factors influencing the flow of cooling air $\dot{V}_c$ to and from a compressor

To obtain generally applicable values for the flow of cooling air $\dot{V}_c$ the following outline conditions have been set that influence the volume of cooling air $\dot{V}_c$.

- Room temperature 35°C = 308 K
- Temperature gradient **Δt** 10 K
- Wall thickness 25 cm
  The surrounding walls are assumed to be homogenous brick walls without windows and doors.
- Room height and size.
  The room height is defined as being lower than 3 m and the area of the room less than 50 m².

The defined outline conditions above assume the least favourable admissible environment for operating the compressor. The values calculated for the flow of cooling air $\dot{V}_c$ are generally applicable because conditions in real installation rooms are normally better.

Thermal problems will not occur if the recommended flow of cooling air $\dot{V}_c$ for a compressor is assured.

10.3.3 General information for ventilation of compressor rooms

This chapter specifies the most important conditions concerning air supply and extraction that must be satisfied by the installation room of one or more air-cooled compressors. They are based on the requirements set forth in VDMA specification sheet 4363 „Ventilation of installation rooms for air-cooled compressors“.

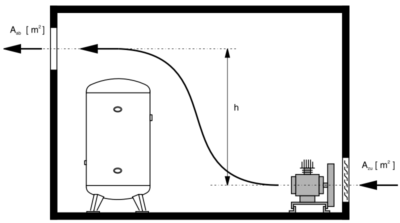


Fig. 10.4:
Arrangement of air inlet and outlet apertures

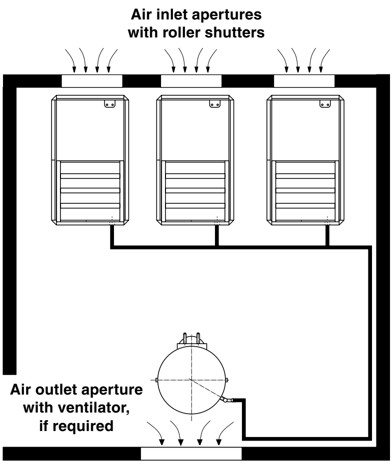


Fig. 10.5:
Installation room with three sound-insulated compressors

- Hot air always rises. To allow an effective exchange of heat, the (inlet) apertures for the supply of cold air must be located close to the ground and the (outlet) exhaust apertures must be in the ceiling or in a side wall **at the top**.
- The compressor must be installed next to the air inlet aperture $\mathbf{A}_{in}$ so that it draws fresh air for compression and cooling air for ventilation directly from the inlet aperture $\mathbf{A}_{in}$.
- The compressor must be positioned so that it can not reinduct its own heated exhaust air..
- The inlet apertures or ducts of the compressor must be arranged so that dangerous admixtures (e.g., explosive or chemically unstable substances) can not be inducted.
- The exhaust air should flow from the compressor via the compressed air receiver (if fitted) to the outlet aperture $\mathbf{A}_{out}$. The assemblies in the installation room should be arranged accordingly.
- Adjustable roller shutters must be installed in the air inlet apertures $\mathbf{A}_{in}$. This allows the flow of cold air from the outside to be reduced and the temperature should then not fall below the minimum admissible level in Winter. If this is not sufficient, the compressor must be equipped with its own heater. The accessories required can be obtained from BOGE.
- When installing several compressors in one room, care must be taken that there is no thermal interaction between them. If one compressor draws in the exhaust air from another compressor, the system will overheat. The ventilation must cater for the total cooling-air requirement for all compressors. Ideally, each compressor should have its own air inlet aperture of a size according to its need.

10.3.4 Natural ventilation

With natural ventilation, the circulation of air is controlled by an air inlet aperture $\mathbf{A}_{in}$ and an air outlet aperture $\mathbf{A}_{out}$ in the side walls of the installation room. Heat is exchanged by the natural circulation of air only, since hot air rises. For adequate ventilation to be provided, the air inlet aperture must be located as far as possible below the air outlet aperture.

Experience shows that this method of ventilation is only suitable for compressors with ratings of up to 22 kW. Even smaller compressors can have ventilation problems, depending on the conditions in the installation room.

10.3.4.1 Outlet air aperture required for natural ventilation

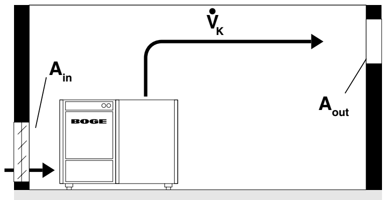


Fig. 10.6 :
Natural ventilation of a compressor installation room with a sound-insulated BOGE screw-type compressor

An adequate flow of cooling-air $\dot{\mathbf{V}}_c$ can only be obtained with natural ventilation if the air inlet and outlet apertures are of an appropriate size.

The figures in the following table are based on VDMA specification sheet 4363 „Ventilation of installation rooms for air-cooled compressors“.

Drive rating **P** **[kW]**	**Required flow of cooling-air** $\dot{V}_c$ **[m³/hr]**	**Required ventilation apertures** A_{in} **and** A_{out} **[m²]**
3,0	1350	0,20
4,0	1800	0,25
5,5	2270	0,30
7,5	3025	0,40
11,0	3700	0,50
15,0	4900	0,65
18,5	6000	0,75
22,0	7000	0,90

In principle, the air inlet $\mathbf{A}_{in}$ and outlet apertures $\mathbf{A}_{out}$ should be of equal size. The cooling air has to pass through both apertures. But taking into account the installation of roller shutters, grids and the like, the air inlet aperture should be approx. 20 % larger than the air outlet aperture $\mathbf{A}_{out}$. If this is not the case, the maximum admissible ambient temperature may be exceeded.

Note

When defining the flow of cooling-air $\dot{\mathbf{V}}_c$ for a compressor station, the cooling-air requirement of a refrigerant compressed air dryer or heat-generating absorption dryer must be included in the calculations.

10.3.5 Artificial ventilation

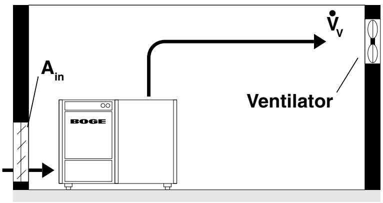


Fig. 10.7:
Artificial ventilation of a compressor room with a sound-insulated BOGE screw-type compressor

In many cases natural ventilation of the installation room is insufficient. Due to structural aspects or the high output of the installed compressor the flow of cooling air is inadequate for the task. In these cases, the hot air must be extracted with the aid of a ventilator.

Artificial ventilation increases the flow speed of cooling air inside the installation room and guarantees the required flow of air by forced ventilation. There are greater reserves when outside temperatures are high. The inlet air aperture must be modified to cater for the ventilator output.

The ventilator(s) should, for reasons of economy, be controlled in several stages by a thermostat. The control depends on the temperature in the installation room. The higher the temperature rises, the greater the output rate of the ventilator.

10.3.5.1 Required ventilator output with artificial ventilation

As with natural ventilation, the required flow of cooling air $\dot{V}_c$ is derived from the output of the installed compressor. The waste heat generated by the compressor must be reliably extracted. The ventilator output $\dot{V}_v$ is approx. 15 % greater than the required flow of cooling-air $\dot{V}_c$. This guarantees perfect cooling, even in high Summer.

The figures in the following table are based on VDMA specification sheet 4363 „Ventilation of installation rooms for air-cooled compressors“.

Drive rating P [kW]	**Required ventilator output $\dot{V}_v$ [m³/hr]**
4.0	1800
5.5	2270
7.5	3025
11.0	3700
15.0	4900
18.5	6000
22.0	7000
30.0	9500
37.0	11000
45.0	14000
55.0	17000
65.0	20000
75.0	23000
90.0	28000
110.0	34000
132.0	40000
160.0	50000
200.0	62000
250.0	70000

10.3.5.2 Required inlet air aperture with artificial ventilation

With artificial ventilation, the exhaust ventilator determines the size of the air outlet aperture.

The aperture needed for an exhaust ventilator is normally much smaller than that required for natural ventilation.

The size of the air inlet aperture $\mathbf{A_{in}}$ depends on the ventilator output $\dot{\mathbf{V}}_\mathbf{V}$ and the maximum flow speed $\mathbf{v_S}$ in the inlet aperture.

It is preferable to calculate with a flow speed of $\mathbf{v_S = 3\ m/s}$. However, if structural considerations do not permit the size of aperture resulting from this calculation, it is also possible to use a flow speed of $\mathbf{v_S = 5\ m/s}$.

The minimum size of the air inlet aperture is calculated with the aid of the following formula:

$$A_{in} = \frac{\dot{V}_V}{3600 \times v_S}$$

$$m^2 = \frac{m^3/hr}{3600\ s/h \times m/s}$$

A_{in}	=	minimum area of air inlet aperture	[m^2]
$\dot{V}_V$	=	Ventilator output	[m^3/h]
v_S	=	maximum flow speed	[m/s]

Note

It is to be remembered when choosing exhaust ventilators, that the flow of cooling-air is subject to the same laws of physics as the compressed air. Even when cooling-air flows through ducts and apertures, when flow speed increases the dynamic pressure **Δp** (pressure loss) rises. A ventilator can only overcome dynamic pressure that lies below its defined surface pressure. If the dynamic pressure is higher than the surface pressure of the ventilator, no volume flow can occur.

The maximum dynamic pressure is determined from the shape and size of the air inlet and outlet apertures together with the respective ducts (if fitted). The flow speed must also be taken into account.

A **Δp = 100 Pa (10 mm WH)** can be assumed for simple apertures without unfavourable diversion (ducting).

10.3.5.3 Example of artificial ventilation of a compressor station

A screw-type compressor, model S 21, is to be operated together with a refrigerant compressed air dryer D 27 in a small installation room. Structural considerations do not allow natural ventilation. Artificial ventilation with a ventilator is therefore required.

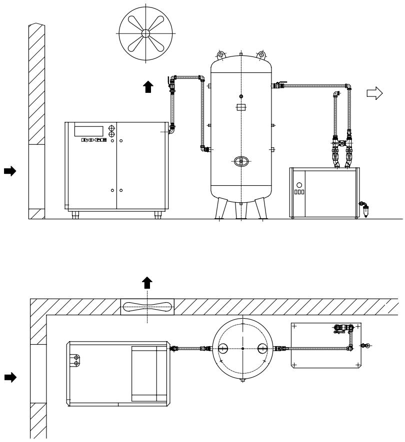


Fig. 10.8:
Compressor station with screw-type compressor, cooling compressed air dryer, compressed air receiver

BOGE screw-type compressor, model S 21

Output $\dot{V}$	:	2.42	m³/min
Motor rating	:	15	kW
Cooling-air req. $\dot{V}_{V1}$	:	4900	m³/hr

Refrigerant compressed air dryer, model D 27

Through-flow rate $\dot{V}$	:	2.66	m³/min	
Cooling air req. $\dot{V}_{V2}$	:	770	m³/min	(see data sheet)

The two flows of cooling air must be added together. The result is the required ventilator output that must be provided in the installation room.

Ventilator output $\dot{V}_{Vttl}$: 5670 m³/hr

The required size of air inlet aperture is calculated using ventilator output $\dot{\mathbf{V}}_{\mathbf{Vttl}}$ and the maximum flow speed $\mathbf{v_S = 3\ m/s}$:

$$A_{in} = \frac{\dot{V}_{Vttl}}{3600 \times v_S}$$

$$A_{in} = \frac{5670}{3600 \times 3}$$

$$A_{in} = 0.525\ m^2$$

A_{in}	=	minimum area of air inlet aperture	[m²]
$\dot{V}_{Vttl}$	=	Ventilator output	[m³/hr]
v_S	=	maximum flow speed	[m/s]

A ventilator with an output of 5670 m³/hr must be installed in the installation room (The dynamic pressure of the apertures must be taken into account when choosing the ventilator). The air inlet aperture $\mathbf{A_{in}}$ should be at least 0.525 m² in size.

10.3.6 Circulation of cooling-air with inlet and outlet ducts

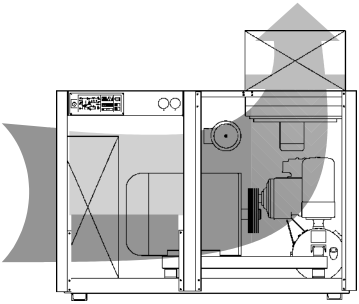


Fig. 10.9:
Circulation of cooling-air in a BOGE screw-type compressor from series S 21 - S 150

The circulation of cooling air through inlet and outlet ducts is an elegant solution to thermal problems in a compressor installation room.

Duct ventilation is possible with sound-insulated compressors. The cool air is directed over the compressor and kept together for extraction. BOGE screw-type compressors are fitted with a cooling ventilator that generates a surface pressure of approx. 60 Pa (approx. 6 mm WH). This means that it can force exhaust air through a straight outlet duct of approx. 5 m in length and with the recommended cross-section.

The ducts can be connected to the apertures in the sound insulation hood without difficulty. There is normally no need for an additional exhaust ventilator inside the duct.

The cool-air ducts direct the air out into the open. But they can also be fitted with flap controls to use the heated air for room heating in Winter. If the compressor rooms are unheated, it may be desirable in Winter to use an air circulation system with part of the heated cooling air being released into the compressor room.

10.3.6.1 Air inlet ducts

It is also possible in principle to supply cooling air to compressors by way of ducts. However, an air inlet duct reduces the induction volume flow (dynamic pressure) and thus has a negative effect on the output of the compressor. For this reason, cooling air should only be supplied through ducts in the following situations:

– Unclean environment.
  The induction air at the location of the compressor contains a high proportion of dirt, dust, chemical impurities or it contains too much moisture. Under these conditions the air supply should be drawn from a cleaner part of the building.
– High ambient temperature.
  The temperature at the compressor's location is distinctly higher than that in neighbouring rooms or outside the building. This is possible if a lot of heat is given off by systems and machinery in the compressor room.

10.3.6.2 Extraction of air through a cool-air duct

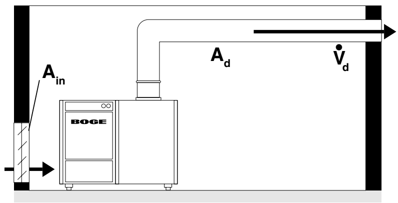


Fig. 10.10:
Extraction of air from a compressor room with a BOGE screw-type compressor, emitting the air into the open

Compressor rooms containing individual units can usually be cooled by a appropriately arranged exhaust ventilator or by natural ventilation. When there are several compressors set up in one installation room, the use of cool-air ducts is always recommended.

When ducts are fitted, the installation room is not heated as much by waste heat from the compressor.

The difference in temperature **Δt** between the inlet and outlet air is approx. 20 K. The flow speed in the outlet ducts should not exceed 6 m/s. The cross section (radius) required for the duct is therefore much smaller than the wall aperture when using natural or artificial ventilation.

10.3.6.3 Required flow of cooling-air $\dot{V}_d$ and cross-section of duct A_d when using a cool-air duct

The figures for the required flow of cooling air $\dot{V}_d$ with ducts given in the following table are based on VDMA specification sheet 4363 „Ventilation of installation rooms for air-cooled compressors“. An increase in the temperature of the cooling-air of **Δt = 20 K** is assumed.

The calculation used to determine the required free cross-section of the duct $\mathbf{A_d}$ are based on a maximum dynamic pressure in the duct of 50 Pa (5 mm WH). This corresponds to approx. 5 m of straight outlet duct, with no bends, tapering or objects inside, a flow speed of 4 – 6 m/s.

Drive rating P [kW]	Required flow of cooling-air with exhaust duct $\dot{V}_d$ [m³/hr]	Required free cross-section for duct A_d [m²]
4.0	800	0.08
5.5	1000	0.10
7.5	1300	0.13
11.0	1700	0.13
15.0	2900	0.15
18.5	4500	0.23
22.0	4500	0.26
30.0	4500	0.33
37.0	6500	0.41
45.0	6500	0.48
55.0	8000	0.59
65.0	8600	0.64
75.0	9200	0.68
90.0	16000	0.85
110.0	16000	1.11
132.0	24400	1.24
160.0	24400	1.61
200.0	27800	2.06
250.0	33600	2.49

10.3.6.4 Information concerning ventilation by ducting

All objects or features inside ducts, such as diversions, filters, roller-shutter flaps, curvatures, T-pieces and silencers cause an increase in flow resistance and thus an obstacle to the flow of air. If the duct has many such features and is very long, the size of the recommended free cross-section (radius) of the duct must be checked by an expert.

There are appropriate fire safety measures prescribed to prevent fire from spreading through ventilation ducts. DIN 4102, part 6 requires the installation of automatic fire safety flaps whenever ventilation ducts pass through a wall.

If the duct is long or unfavourably laid, the dynamic pressure can be over 50 Pa (5 mm WH). In this case there is a risk that the cooling ventilator of a screw-type compressor can not overcome the dynamic pressure in the duct. This means that cooling air stops flowing and the entire cooling effort for the compressor collapses. In this case an auxiliary ventilator will have to be installed.

The air inlet and outlet flaps as well as the ventilators should, for economical reasons, be controlled by a thermostat in the installation room.

The cooling-air ducts must never be mounted directly on the compressor housing. Compensators that remove tension and stop the transmission of vibration must always be used.

A cooling-air duct with sound-insulation cladding radiates less heat to the surroundings and also suppresses noise that comes out of the compressor with the exhaust air.

BOGE generally recommends that the task of installing the ducts and any associated construction work be given to a specialist company.

With multiple units, each compressor must have its own air inlet and outlet duct.

When using a collective duct for multiple units, automatic check flaps must be used to prevent heated cooling-air flowing over a compressor that is switched off in the installation room and heating the inlet air.

10.3.6.5 Dimensioning the air inlet aperture when using an outlet duct

The size of the air inlet aperture $\mathbf{A_{in}}$ is dependent on the flow of cooling-air $\mathbf{V_d}$ and the maximum flow speed $\mathbf{v_s}$ in the aperture itself.

It is preferable to calculate with a flow speed of $\mathbf{v_s = 3\ m/s}$. However, if structural considerations do not permit the size of aperture resulting from this calculation, it is also possible to use a flow speed of $\mathbf{v_s = 5\ m/s}$.

The minimum size of the air inlet aperture is calculated with the aid of the following formula:

$$A_{in} = \frac{\dot{V}_d}{3600 \times v_S}$$

$$m^2 = \frac{m^3/h}{3600\ s/h \times m/s}$$

A_{in}	=	minimum area of outlet aperture	[m²]
$\dot{V}_d$	=	Ventilator output	[m³/h]
v_S	=	maximum flow speed	[m/s]

10.3.6.6 Variations of duct-type ventilation

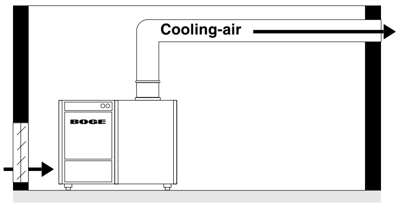


Fig. 10.11:
Extraction of air into the open using an outlet duct

The duct directs the hot exhaust air directly into the open. This method is recommended if there are high temperatures in the compressor room.

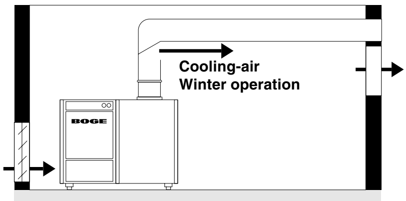


Fig. 10.12:
Outlet duct with circulation flap

The outlet duct directs the hot cooling-air directly into the open. When temperatures in the installation room are cold, hot exhaust air is added to the cold room air through a circulation flap. The circulatory ventilation prevents the unit from freezing when outside temperatures are below zero. It is also recommended to have auxiliary heating to prevent a cold compressor from freezing during the start-up phase.

When this method is used, it is necessary to have an air outlet aperture dimensioned according to the flow of cooling-air in addition to the outlet duct itself.

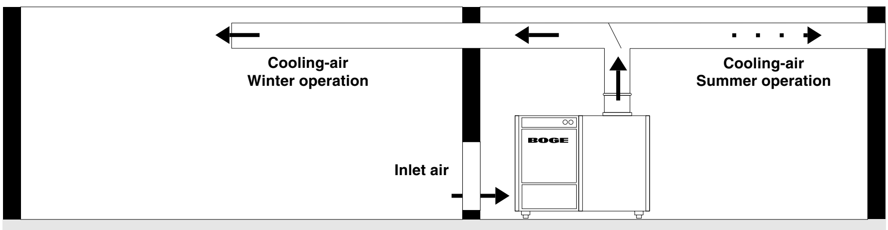


Fig.10.13:
Using hot cooling-air for heating

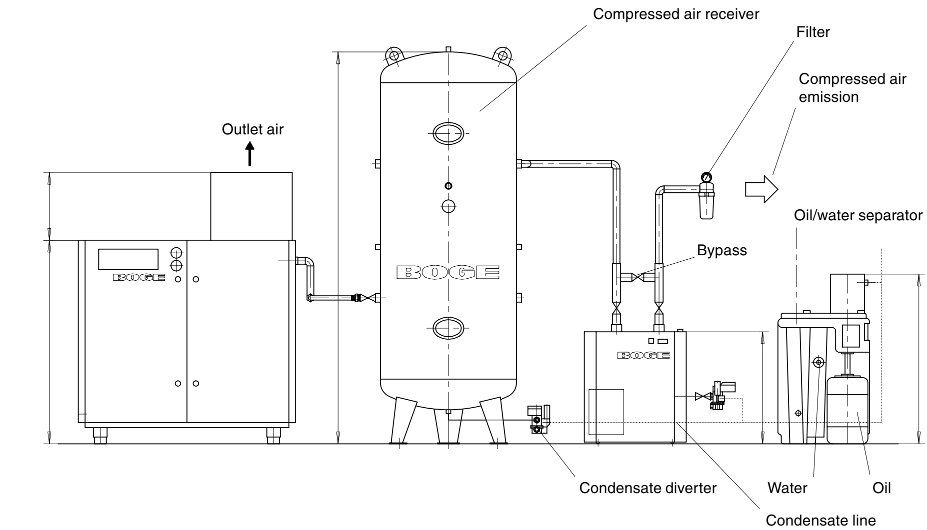
When the outdoor temperature is cold (in Winter) duct directs all or some of the heated cooling-air from the compressor into other rooms in the building in order to heat them. When outdoor temperatures are hotter (in Summer) the duct emits the air directly into the open.

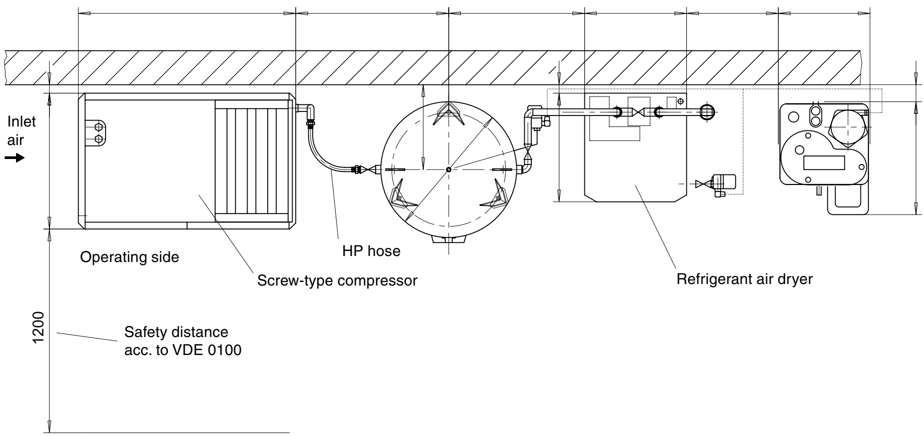
With this configuration, the inlet air is mostly drawn from heated rooms. This guarantees that the cooling-air is warm enough when ambient temperatures are low. The compressor then always operates above the minimum admissible temperature.

Air filters and silencers should be installed in the outlet duct in order to reduce dust and noise in the rooms heated.

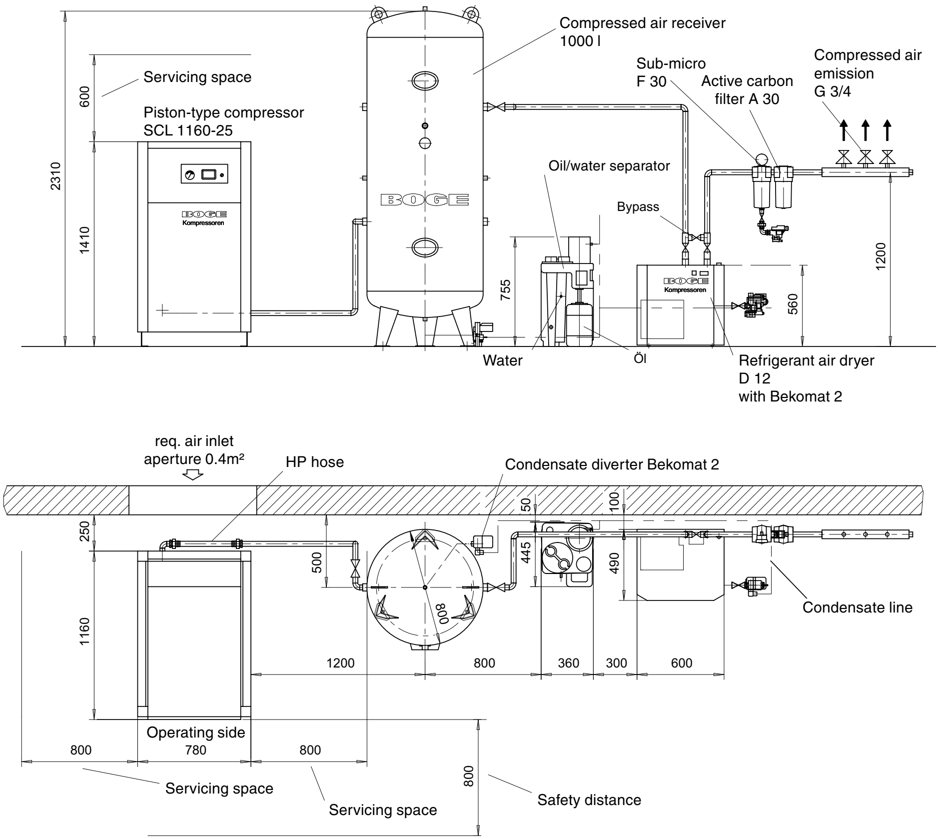
10.4 Example installation plans

10.4.1 Installation of a screw-type compressor: an example





10.4.2 Installation of piston-type compressor: an example



11. Heat recovery

Rising energy costs and increasing environmental awareness led many compressor-users to the view that the enormous potential of compressor heat must not longer be allowed to escape unused. They approached the compressor makers who developed high performance heat recovery systems. Since then, the heat given off by compressors has been utilised. It serves to heat rooms, and to heat utility and heating water.

11.1 The heat balance of a compressor station

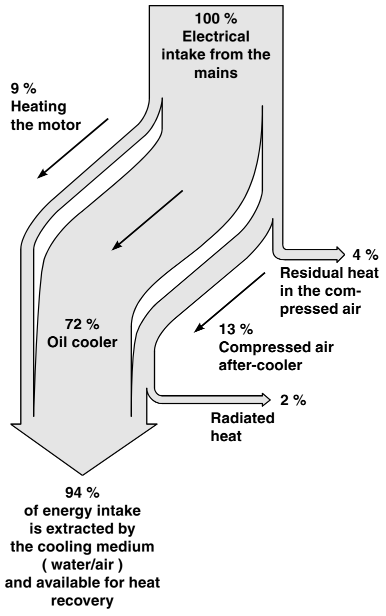


Fig. 11.1:
Distribution of heat in a screw compressor with oil injection cooling

To be able to appreciate the possibilities of heat recovery from compressors it must be taken into account that on the basis of the first principle of thermodynamics the entire electricity intake of a compressor is converted into heat. In order to make this heat useful one must know where it occurs and what proportion of it can be economically reclaimed for further use.

The heat is always discharged with the aid of a coolant. This coolant contains approx. 94 % of the electrical energy entering the compressor in the form of heat. Approx. 4 % remains in the compressed air as residual heat and approx. 2 % is lost to the atmosphere by radiation.

When drawing up the balance sheet, assumptions should not be based only on the output from the motor that the compressor needs to compress the air. The electric motor itself converts energy into heat. One must also consider the efficiency rate of the motor, which according to the drive rating lies between 80 % and 96 %. This again increases the amount of heat emitted.

11.2 Room heating

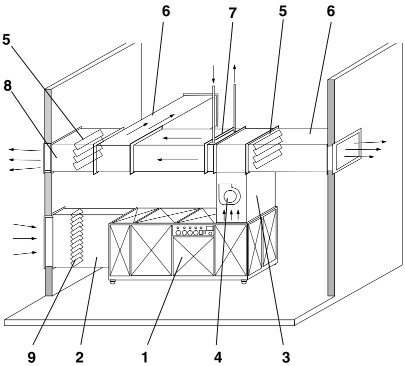
The most obvious use for compressor heat is to heat rooms.

With the simplest method of room heating the compressor is installed in the room to be heated. This means that the compressor is installed directly in the workshop or storeroom, usually close to workplaces.

In this case, the only ducts required are those to discharge hot air into the open during Summer. The heating air does not require to be transported over long distances.

Of course, there must be adequate cooling for the compressor. Sound insulation is normally required by safety rules.

11.2.1 Room heating through ducting



1 = Silenced compressor
2 = Inlet duct
3 = Outlet duct
4 = Additional exhaust ventilator
5 = Control flaps (thermostatically controlled)
6 = Air outlet duct (Room heating)
7 = Heat exchanger
8 = Air outlet duct (into the open for Summer operation)
9 = Air inlet flap

Fig. 11.2: Op. diagram of ducting

To utilise the heat emitted by a central compressor station the heated flow of cooling air must be brought through ducts into the rooms to be heated. This is only recommended for larger compressors since smaller ones do not provide enough usable heat.

The flow of cold air passes over the compressor and drive motor. The cooling air absorbs the emitted heat and is drawn into an outlet duct with the aid of a ventilator. In this process the cooling air normally heats up to +50° / +60°C.

One use of compressor heat for room heating requires a silenced compressor with ducted cooling air. BOGE screw compressors are all silenced and fitted with an internal ventilator. For this reason they can be connected to a ducting system without difficulty. Non-silenced compressors (e.g., most piston compressors) can not be upgraded later for utilisation of emitted heat, even if an adjusted sound-insulation hood is fitted.

11.2.2 Operation of room heating

Insulated ducts conduct the warm cooling air of a compressor or compressors at low outside temperatures into the building. This heats the respective rooms. If the outside temperatures are high, a duct directs the cooling air directly into the open.

The flow of cooling air is directed by inlet and control flaps. These flaps and the ventilators should be controlled by an adjustable room thermostat which monitors the temperature in the heated rooms.

Fire safety measures are prescribed to prevent fire spreading through the ventilation ducts. DIN 4102, Part 6 requires that self-closing fire safety flaps be installed if the ventilation flaps pass through a wall.

It is possible to heat exchangers in the ducts. With the aid of these heat exchangers, water can be heated to a temperature of approx. +40°C. This hot water can assist a central heating system or be used as utility water.

11.2.3 Economy of room heating

The installation costs of room heating can be very high in proportion to to energy costs saved. Before installing an expensive system, it should be checked that enough heat is generated to justify the expense of a ducting system. It should be taken into account that the flow of hot air inevitably cools down if it has to travel long distances through a ducting system. The investment must be in the correct proportion to the heating costs saved.

The cost savings increase the more the compressor is used. The more the compressor runs, the more effective the room heating is.

11.3 The Duotherm heat exchanger

For screw compressors with oil injection cooling there are special heat recovery systems for heating utility water or heating water. A heat exchanger is installed in the main flow path of hot oil in the compressor. Utility or heating water is heated by this hot compressor oil.

The Duotherm heat exchangers operate independently of the type of compressor cooling because the heat exchanger is installed as a pre-cooler before the actual air and water cooler.

11.3.1 Duotherm BPT

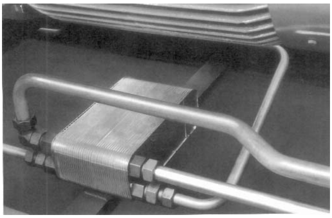
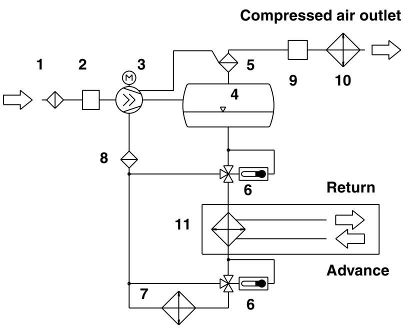


Fig. 11.3:
The heat reclamation system BOGE-Duotherm BPT



1 = Intake filter
2 = Suction controller
3 = Compressor stage
4 = Combined compressed air/oil vessel
5 = Oil separator
6 = Thermostatic oil control valve
7 = Oil cooler
8 = Oil filter
9 = Min. pressure non-return valve
10 = Compressed air aftercooler
11 = Heat exchanger

Fig. 11.4:
Flow diagram of BOGE-Duotherm BPT

The Duotherm BPT-System is used for heating water or hot production water. The heart of this system is a plate heat exchanger consisting of a number of profiled, stainless steel plates. The piled plates form a mutually isolated two channel system. A Special process of hard-soldering connects these layered plates together. Seals, which have the inherent risk of leaks, are not required. The resulting heat exchanger works very effectively and reliably.

Operating principle

The oil heated to approx. +90°C by the compressor circuit flows through the plate heat exchanger. The water coming in reverse flow through the exchanger is heated up to +70°C. The heated quantity of water is independent of the temperature difference in this process.

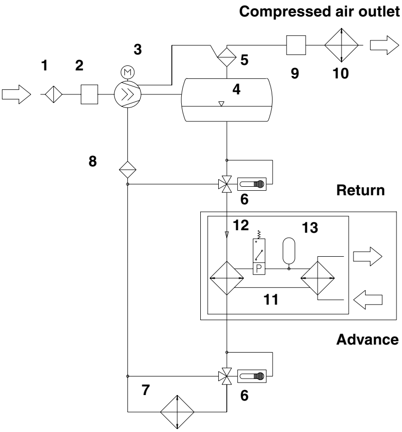
There is a thermostatic oil control valve before and after the heat exchanger. Depending on the oil temperature the flow of oil is either sent through the oil cooler and also the heat exchanger or through a bypass.

Features

– When the stop valves in the water inlet and outlet are closed an enclosed space is formed at the same time. When the water heats in this space it expands and the pressure rises. An expansion vessel and safety valve must be installed in order to prevent damage to the plate heat exchanger.
– If the water is very dirty, a dirt pan with a maximum pore width of 0.6 mm should be installed in the line.
– Flush connections for cleaning the heat exchanger must be fitted.
– This heat exchanger is normally integrated in the compressor cabinet. It can be set up separately or fitted on site later.

11.3.2 Duotherm BSW

Fig. 11.5:
The heat reclamation system BOGE-Duotherm BSW



1 = Intake filter
2 = Suction controller
3 = Compressor stage
4 = Combined compressed air/oil vessel
5 = Oil separator
6 = Thermostatic oil control valve
7 = Oil cooler
8 = Oil filter
9 = Min. pressure non-return valve
10 = Compressed air aftercooler
11 = Safety heat exchanger
12 = Pressure monitor for aperture
13 = Expansion vessel

Fig. 11.6:
Flow diagram of BOGE-Duotherm BSW

The Duotherm BSW-System is used to heat drinking and utility water. Since other rules apply in the sanitary area, this is a safety heat exchanger. Two independent circuits are kept apart by a separation liquid.

The BSW-System is a pipe bundle heat exchanger in which one pipe is inserted into another without making contact. The safety area in this double pipe is filled with a non-toxic separation liquid. The liquid transmits the heat and in the event of damage it prevents the water from mixing with the oil. The drinking water can therefore not be contaminated.

A pressure monitor switches immediately in the event of pipe breakage. The emitted impulse can be processed elsewhere (e.g., for an alarm or to shut down the system).

Operating principle

The oil from the compressor circuit heated to approx. +90°C flows through a pipe bundle. The separating liquid transmits the heat to the utility water in the second bundle. The water coming in reverse flow through the second pipe bundle can be heated to approx. 55°C. The quantity of water heated depends on the temperature difference. The heated water is subsequently directed to a appropriate container (boiler) from where it can be transported to the hot water circuit.

There is a thermostatic oil control valve before and after the heat exchanger. Depending on the oil temperature the flow of oil is either sent through the oil cooler and also the heat exchanger or through a bypass.

Features

– The pressure monitor must be set to a value that is at least 20 % below the minimum pressure of the media used.
– Conditions for use

Minimum water pressure	0,5 bar
Maximum water pressure	16 bar
Maximum oil pressure	16 bar
Maximum pressure of separating liquid	10 bar
Maximum temperature (oil and water)	+100°C

If the maximum temperature is exceeded, malfunctions will follow and an alarm will be actuated.
– Because of its size, the BSW safety heat exchanger is integrated in the compressor cabinet. It can also be set up separately or fitted later on site.

11.3.3 How much energy is it possible to save ?

The Duotherm-System makes available 75 % of the electrical power taken into the compressor. This takes in the form of heat discharged by the compressor oil.

The values given in the table for the quantity of heat and water have been calculated on the basis of energy retention and the general laws of heat transfer. They are in principle applicable for both Duotherm systems. When using a Duotherm BWT system it is not economical to heat utility water to above +55°C because the amount of water heated is too small.

The values given assume continuous compressor operation, and heat loss is not taken into account because local conditions vary. The calculation of savings for heating costs is based on conventional oil heating :

- Specific heating value H for heating oil 38.0 MJ/l
- Price of heating oil 0.20 €/l
- Heating efficiency 75 %
- Operating hours 1000 hrs

Drive-rating [kW]	Discharged power [kW/h]	Usable quantity of heat [MJ/h]	Quantity of water at Δt 25 K 313 → 338 K [m³/h]	Quantity of water at Δt 35 K 293 → 328 K [m³/h]	Quantity of water at Δt 50 K 293 → 343 K [m³/h]	Cost savings at 1000 hrs [€]
11.0	8.9	32.0	0.305	0.217	0.152	225.–
15.0	12.3	44.2	0.420	0.300	0.210	310.–
18.5	14.8	53.2	0.509	0.363	0.255	373.–
22.0	17.7	63.7	0.609	0.435	0.305	447.–
30.0	24.4	87.8	0.835	0.596	0.417	616.–
37.0	30.3	109.0	1.040	0.743	0.520	765.–
45.0	37.7	135.7	1.295	0.925	0.647	952.–
55.0	45.5	163.8	1.565	1.118	0.782	1149.–
65.0	54.9	197.6	1.885	1.346	0.942	1387.–
75.0	63.1	227.1	2.170	1.550	1.085	1594.–
90.0	74.0	266.4	2.545	1.818	1.272	1869.–
110.0	90.0	324.0	3.095	2.210	1.547	2274.–
132.0	110.5	397.0	3.800	2.714	1.900	2786.–
160.0	133.5	480.6	4.590	3.278	2.295	3373.–
200.0	168.3	605.8	5.790	4.136	2.895	4251.–
250,0	208,9	752,0	7,180	5,128	3,590	5277.–

11.4 Closing remarks concerning heat recovery

Compressors offer enormous possibilities for saving energy and costs through exploitation of heat emission. However, it is not wise to attempt to force heat from a small compressor. It is normally only worth the expense with large screw and piston compressors and combined systems. The usable energy rises with the capacity of the compressor.

The investment costs for a heat recovery system depend much on local conditions. They must be taken into account because they are a big influence on the amortisation time of the system.

A principle decision must be made whether to use the emitted heat for room heating or for utility and heating water. Remember that room heating is seldom used in Summer.

Compressor usage is also a major factor. The longer it operates, the more heat there is, and it is available continuously and in ample supply.

Before such a system is installed, a needs analysis should always be made for the heat requirement. This analysis can then be compared with the average running time of the compressor.

This comparison then allows the true value of the heat recovery system to be seen. It will also show whether reclamation can cover the demand for heating or whether a second heating system is needed.

12. Sound

12.1 The nature of sound

Sound waves are mechanical vibrations of an elastic medium. Starting from a sound source, a vibrating body, they spread in solid bodies, liquids and gases in the form of pressure fluctuations(pressure waves). The study of sound is called acoustics.

Vibrating bodies of all aggregate conditions can transmit sound waves.These are known as sound sources.These can be strings, rods, plates, columns of air, membranes, machines etc.

If the vibrations are emitted from the ambient air they are known as **airborne sound**.

The vibrating bodies, gases and liquids can transmit the vibrations to solid objects. In this case they are known as **structure-borne sound**.

12.1.1 Sound perception

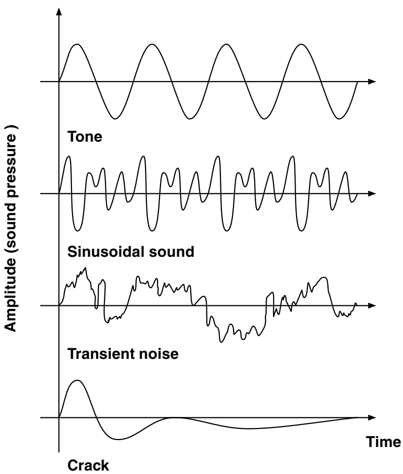


Fig. 12.1: Impressions of sound

There are the following connections between the vibrations of airborne sound coming from the vibrations of a sound source and the human perception of sound:

Amplitude of vibration

The amplitude is the periodic deviation of pressure that occurs in a sound wave.

It corresponds to the impression of loudness perceived by human beings.

Frequency of vibration

The frequency is the number of pressure fluctuations during a unit of time. It is normally measured in **Hz** (vibrations per second).

This corresponds to the impression of tone perceived by human beings.

Vibration form

A distinction is made between different forms of vibration which cause the different impressions of sound:

- Tone.
  A tone (pure tone) is a sinus vibration.
- Sinusoidal sound.
  This is the superimposition of several tones. Several sinusoidal vibrations superimpose and form a non-sinusoidal vibration. The tone with the lowest frequency defines the overall perception of the sound. The other tones (top tones) give the impression of sound colour.
- Transient noise.
  Transient noise is an irregular vibration. It is a mixture of very many frequencies or different magnitudes.
- Crack.
  A crack is a single, short and sharp report.

12.2 Important terminology in acoustics

12.2.1 Sound pressure

Sound pressure $\tilde{\mathbf{p}}$ is the periodic pressure deviation (over and under pressure and alternating pressure) that occurs in a sound wave. It is measured in **Pa** (10^{-5} bar).

In gaseous media sound pressure is superimposed over the existing gas pressure **p**. Sound pressure is heavily dependent on various factors e.g., the sound output of the source, the spatial circumstances etc.

Sound pressure moves between approx. 2×10^{-4} Pa with the ticking of a clock and approx. 65 Pa with the start of an aircraft in the direct vicinity.

12.2.2 Sound level

To be able to handle acoustic sizes better, the value is set in proportion with a reference size put in a logarithm. The levels as logarithm of a proportional size are dimensionless. The designation **dB** (Decibel) is added.

The sound pressure level is set in proportion to the reference pressure $\mathbf{p_0 = 2 \times 10^{-5}}$ **Pa** and pu in a logarithm. The following applies for the sound pressure level:

$$\boxed{L_p = 20 \lg \frac{\tilde{p}}{p_0} \text{ dB}}$$

L_P	=	Sound pressure level	[dB]
$\tilde{p}$	=	Sound pressure	[Pa]
p_0	=	Reference sound pressure	[2×10^{-5} Pa]

The other sizes in acoustics are treated in similar fashion. Acoustics uses almost only levels to indicate sizes.

12.2.3 Sound intensity

The sound intensity indicates the sound energy radiated by a sound source per second. It is a machine-specific size (emission size) and can be influenced by sound insulation measures among other methods.

Using the sound intensity of a machine, it is possible to calculate the sound pressure level of a certain location, taking into account the distance, the structural conditions and other sound sources near to the sound intensity. There is often no need to carry out extensive measuring.

12.3 Human perception of sound

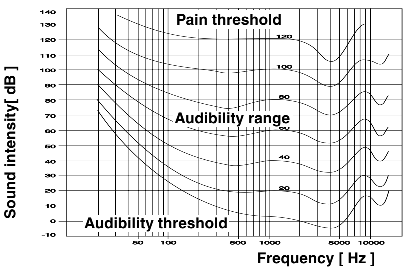


Fig. 12.2:
The human hearing range

The human ear can normally only hear frequencies from 16 to 20000 Hz . Higher frequencies are described as supersonic, lower ones as infrasonic. The perceptible sound pressure is between 10^{-5} Pa and 100 Pa, whereby a sound pressure of 100 Pa nearly always leads to the immediate loss of hearing in humans.

The human sense of hearing does not perceive the various sound pressures and frequencies with the same intensity. The audibility range offers a summary of the sound pressure and frequency ranges perceptible to humans. The bottom limit of the curve shows the **audibility threshold** and top curve the **pain threshold**. The largest range of sound pressure perceptible to the human ear is at around 1000 Hz.

12.3.1 The sound intensity level

Sound pressure is a physical size and can therefore be measured. The intensity at which a person perceives sound pressure is a physiological size that depends on the sense of hearing.

The level of loudness is an empirically determined size. The perception of loudness has been tested in series of experiments with different people and an average value formed. The level of loudness is given in **Phon**.

At 1000 Hz the sound intensity level matches the unassessed sound pressure level. The sound intensity level can not be measured with technical instruments. This is why comparative measurements are very difficult, if not impossible.

12.3.2 Assessed sound level dB (A)

Acoustic sizes must be adapted to the perception range of the human ear in a way that they make also technical sense. Depending on the frequency, the real sound pressure level is adjusted with certain values to the sensitivity of the ear. There are valid international evaluation curves for these adjustment values.

Some areas of application for different evaluation curves are given below.

A – Evaluation curve for L_N = 30 - 60 Phon.

B – Evaluation curve for L_N = 60 - 90 Phon.

C – Evaluation curve for linear audibility range.

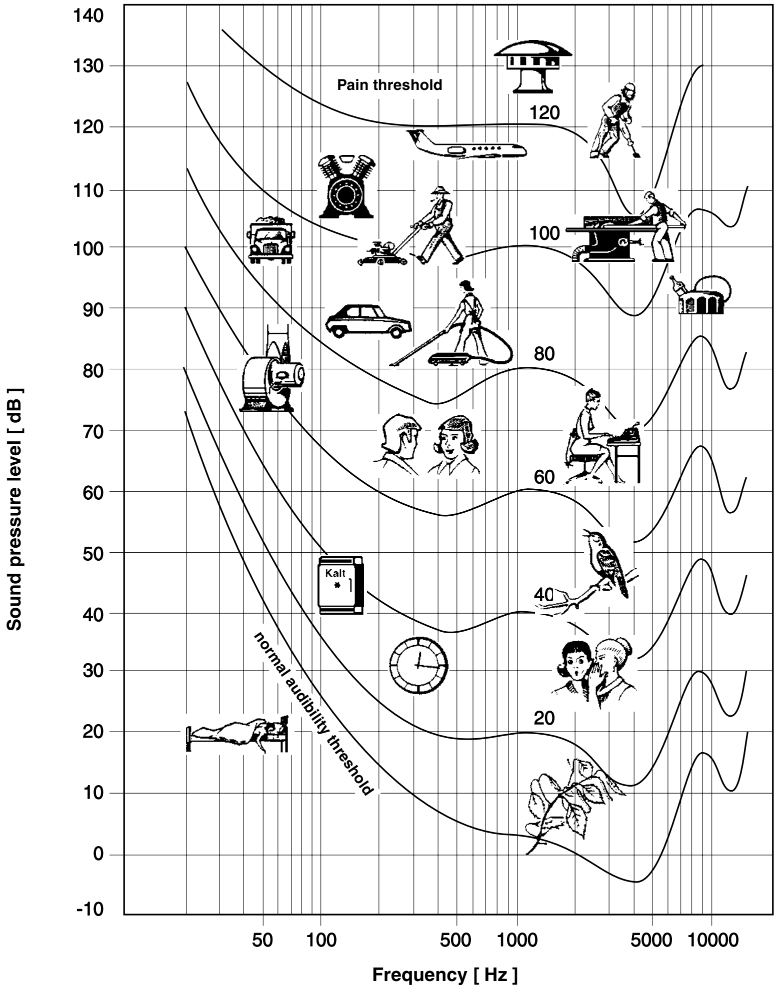
D – Evaluation curve for aircraft noise.

An evaluated sound level is indicated by having the letter of the evaluation curve e.g., dB (A) suffixed.

The A-evaluation curve is the one primarily used in measuring the noise of compressors and other machinery. Sound measurement as standardised in DIN 45635 uses A-evaluated sound pressure levels.

12.3.3 Loudness in comparison

The following diagram shows the hearing range of an average person, which lies between the audibility threshold and the pain threshold, together with various examples of differing loudness.



The ticking of a clock corresponds to a sound pressure level of approx. 20 dB (A).

Normal conversation at a distance of around 1 m corresponds to a sound pressure level of approx. 70 dB (A).

12.4 Behaviour of sound

The dissemination and general behaviour of sound depends on various factors. It must also be taken into account that the sound output of a machine (the sound source) remains constant.

12.4.1 Distance from the sound source

The sound pressure generated from the source always diminishes with increasing distance. The constant sound output of a source disseminates over a greater area (dispersion) with increasing distance. The form of the sound wave plays an important part in this. Machinery and compressors nearly always radiate sound energy in the form of a semisphere because they are normally on a firm base.

The sound pressure level then goes down, with reference to the 1 m distance value, as shown in the following table:

Distance from the sound source [m]	1	2	5	10	25	50	100
Sound pressure level reduction[dB (A)]	0	5	12	16	23	28	32

These starting values refer to an unrestricted dissemination of sound over an open area. A certain amount of reflection from normal, reverberant ground is taken into account.

Example

An ultra-silenced BOGE screw compressor S 21 is installed in a large hall. It generates according to DIN 45635 a sound pressure level of 69 dB (A). At a distance of 10 m the sound pressure generated by the compressor is only around 53 dB (A).

12.4.2 Reflection and Absorption

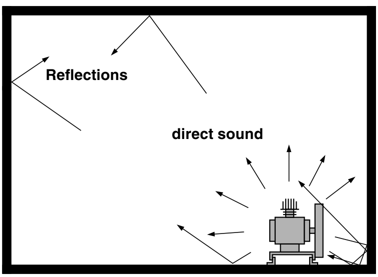


Fig. 12.2:
The dissemination of sound in an enclosed space

A part of the sound is reflected by the walls and other objects. In rooms, reflection causes a diffuse field of undirected sound waves. The general level of sound pressure in the room is increased by reflected sound. This reflected sound is known as reverberation.

Reverberant materials with smooth surfaces, such as brick walls, reflect a large amount of occurrent sound. The shape of the surface heavily influences the reflections. If a room is padded with specially arranged insulative pyramids the result is an acoustically dead room without reflection. Rooms of this type are used to measure sound pressure and the like with scientific accuracy.

The sound not reflected is absorbed by walls or objects. The material conducts the absorbed sound further and damps it. It is usually transmitted back to the air at another point. Materials with a high elasticity module, such as steel, conduct sound very well. The damping effect is usually low.

12.4.3 Damping sound

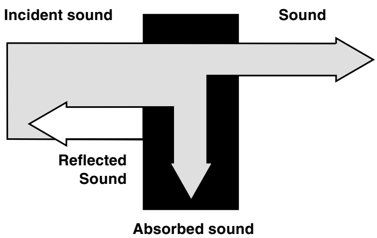


Fig. 12.3:
Sound insulation (damping) by walls

Damping is the conversion of sound energy into heat generated by the friction of particles against each other. The sound is absorbed in this process. Damping of airborne sound is achieved by porous or fibrous absorption materials with a low elasticity module and a large area mass (kg/m^2). The extent to which sound is damped by appropriate materials also depends on the frequency spectrum of the sound. Some frequencies are affected more and others less.

Sound damping by the air depends much on the temperature and humidity of the air. Under normal conditions it is only perceptible from a distance of 200 m. When humidity is high e.g., in fog, the damping effect is greater.

12.4.5 Dissemination of sound in pipes and ducts

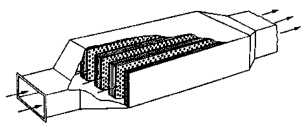


Fig. 12.4:
Absorption silencer with straight elements

Special laws apply for the dissemination of sound in pipes and ducts. A flowing medium and the reflections in a narrow duct assist the dissemination of sound. Measures must be taken against the unrestricted dissemination of sound in ducts, particularly when the hot outlet air of a compressor is being used for room heating.

Coming from a silenced compressor, a sound wave is directed into the air outlet duct. The sound, which here is not affected by the silencer, continues through the duct system. It proceeds unimpeded through the ventilation apertures and into the heated rooms.

There are various measures that can be taken to reduce the continuation of sound in ducts or pipes:

– Linear insulation.
  The ducts are lined with strongly absorbent materials. This reduces the sound energy and the sound pressure level in the duct.
– Absorption insulation.
  A part of the duct is loosely filled with sound absorbent material (e.g., rock wool). This absorbs a large part of the sound energy, similar to walls. The great drawback of this form of insulation lies in its high resistance to flow. Insulation of this type is not recommended in duct systems without a big exhaust ventilator.

12.4.6 Sound pressure level from many sound sources

If there are several sources of sound in one room, the sound pressure level will rise. The more sound energy emitted, the higher the sound pressure. The perceived intensity of the sound increases. The correlations are not linear. They depend much on the structure of the room, the sound pressure levels of the individual sources and their frequency spectrum. Therefore, when looking at the correlations, only the two simplest cases are given here.

The numbers given here should be seen as reference values only. They may deviate sharply in individual cases because many influencing factors are not taken into consideration.

12.4.6.1 Several sound sources with the same level

When there are two or more sound sources with the same sound pressure level in a large room, the correlation is relatively simple. The following table shows the increase of the overall sound pressure level without taking possible reflection or transient noise into account:

Number of sound sources	2	3	4	5	10	15	20
Increase of sound pressure level [dB (A)]	3	5	6	7	10	12	13

To obtain the overall sound pressure level the increase in sound pressure must be added to the sound pressure levels of the individual sources.

Example

There are three ultra-silenced BOGE screw compressors S 21 in a large hall. Each generates according to DIN 45635 sound pressure of 69 dB (A). The overall sound pressure level is therefore at 74 dB (A) [69 + 5].

12.4.6.2 Two sound sources with different levels

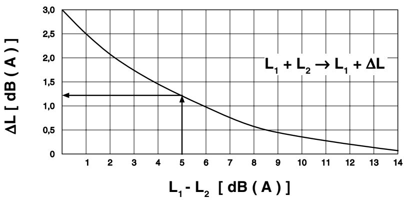


Fig. 12.5:
Sound strengthened by two sources with different levels

The total sound pressure **of two** different sound pressures **($L_1 + L_2$)** can be determined with the aid of a diagram. When there are several sound sources with different levels the correlations are very complicated.

The diagram shows by how many Decibels (**ΔL**) the higher of the two sound levels **L_1** rises in dependency on the difference between the two levels**(L_1 - L_2)**.

Example

A compressor with a sound pressure according to DIN 45635 of 69 dB (A) and a compressor with a sound pressure of 74 dB (A) are installed in the same room. The total sound pressure in this case is approx. 75.3 dB (A).
[74 – 69 = 5 → 74 + 1.3 = 75.3]

12.5 The effects of noise

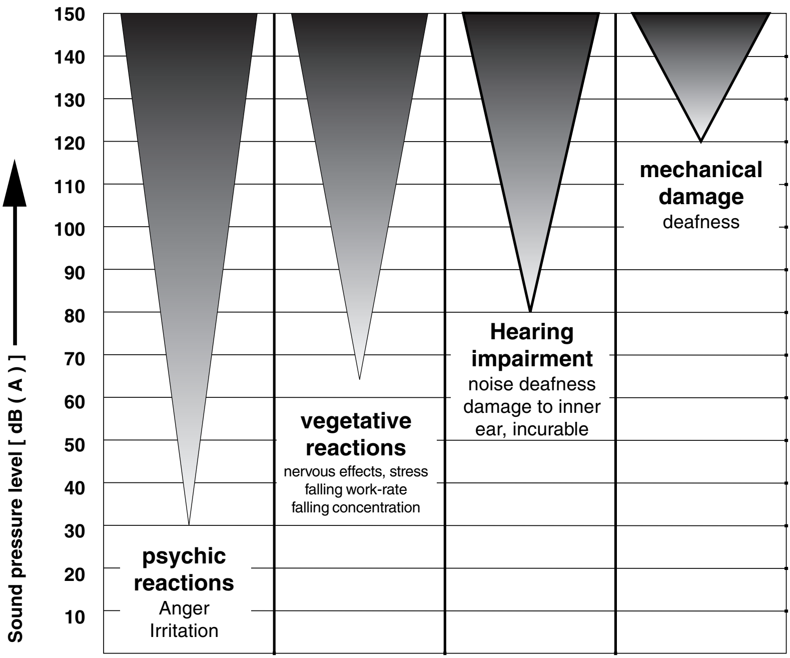


Fig. 12.6:
Noise as a health hazard

One form of sound is noise. This is undesired, annoying or painful sound. Noise has various adverse effects depending on its sound pressure:

– Disturbed concentration
– Sound pressure of approx. **70 dB (A)** disturbs speech communication.
– Sound pressure of **85 dB (A)** usually leads to a temporary reduction of hearing after an 8-hour shift. If this acoustic stress continues for several years it can cause permanent damage to hearing.
– Sound pressure of **110 dB (A)** leads to a reduction of hearing in a very short time. If this stress continue for several hours it is very likely to result in permanent damage to hearing.
– Sound pressure of **135 dB (A)** and above causes immediate deafness in most cases.

12.6 Noise measurement

When measuring noise at compressors and similar machinery the main method used is the enveloping surface method of DIN 45635 or other norms like Cagi-Pneurop or PN 8 NTC 2.3. These norms define the conditions for measuring the noise emitted by compressors and machinery to the outside air (noise output) according to standard methods, thus making the results comparable.

Noise is mainly measured at compressors and machinery to find out whether certain requirements are being met. The results determined are useful for:

– Comparing similar machinery.
– Comparing different machinery.
– Estimating sound levels at a distance.
– Checking noise emissions with respect to safety laws.
– Planning noise protection measures.

12.7 Silencing on compressors

Fig. 12.7:
Silenced BOGE screw compressors

Compressors sometimes emit sound levels of over 85 dB (A) when in operation. This can be much higher if there are several unsilenced compressors in one room. Since the Work Safety Act recommends the wearing of protective equipment from 85 dB (A) upwards and prescribes it from 90 dB (A), it is often beneficial to install silenced compressors.

Silenced compressors can be installed close to workplaces. This avoids the cost of long lines and separate compressor rooms, and reduces pressure loss in pneumatic lines.

Certain demands are placed on sound-insulation materials:

– Not combustibility.
– Insensitivity to dust.
– Insensitivity to oil.

The silencing material used for compressors is therefore usually mineral cotton (rock wool or fibreglass) and fluorocarbon-free, hardly flammable, self-extinguishing foam material, that is installed in the steel sheet case.

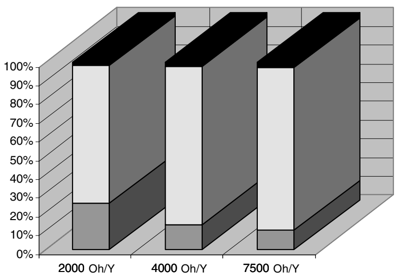
13. Costs of compressed air

13.1 Composition of compressed air costs

The operating costs for compressed air comprise three factors:

- Servicing and maintenance costs.
  The servicing costs are for the wages of the fitter, spare parts and consumed materials such as lubrication and cooling oil, air filters, oil filters and the like.
- Energy costs.
  The energy costs include the costs for electricity and fuel. These are needed to heat the compressor.
- Capital service.
  Capital service includes the interest and repayment on the items invested in (compressor, pre-processing and pipeline). These are the depreciation and interest costs.

13.1.1 Cost factor ratios



Servicing and maintenance costs

Energy costs

Capital service

Fig. 13.1:
Composition of compressed air costs with differing operating hours per year

The individual factors can vary in size, depending on the hours of operation per year. With single shift operation this is normally 2000 hrs/yr, 4000 hrs/yr with 2-shift operation, and 7500 hrs/yr with 3-shift operation.

In determining the cost ratios, calculations are based on electricity costs of 0,10 €/kWh and a depreciation period of 5 years with an interest rate of 8 %.

Cost factors	Hours of operation per year 2000 Oh/y [%]	4000 Oh/y [%]	7500 Oh/y [%]
Servicing and maintenance	2	2.5	2.7
Energy costs	73	84	87
Capital service	25	13.5	10.3

It is easy to see that energy is the greatest cost factor. The servicing and maintenance costs are more or less negligible, and the costs for service of capital equipment are hardly a major item in the long term. The main criterion in acquiring a compressor system must therefore be energy consumption.

There is a breakdown of energy costs on the following page.

13.2 Cost-effectiveness calculation for energy costs

	Maker		**BOGE**	
	Type		**Screw compressor**	
	Model		**S 40**	
(1)	FAD of complete system **($\dot{V}$)**	m³/h	303	
	acc. to PN2 CPTC2			
	Ambient temperature t = 20° C			
	Operating pressure	bar	8	
(2)	Electrical power requirement			
	of compressor	kW		
	of drive belt	kW		
	of transmission	kW		
	of fan	kW		
	of overall system **(P_e)**	kW	31.89	
(3)	Motor efficiency rating **(η)**		92.5	
	with IP 54 protection			
(4)	Total intake **(P_i)**	kW	34.47	
	from electricity supply			
	$P_i = P_e$ (2) × 100 / η (3)			
(5)	Electricity price **(c)**	€/kWh	0.10	
(6)	Electricity costs per hour	€/h	3.45	
	$C = P_i$ (4) × c (5)			
(7)	Costs per m³ compressed air	€/m³	0.0114	
	C_V = C (6) / $\dot{V}$ (1)			
(8)	Costs per year			
	Compressed air requirement **(A_R)**	m³/h	300	
	Hours of operation per year	Bh	2000	
	Compressed air requirement per year	m³	600.000	
	$A_R/Y = Oh \times A_R$			
(9)	Total costs per year	€/Year	6840,–	
	$C_Y = A_R/Y$ (8) × C_V (7)			
(10)	Additional costs per year			

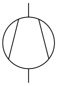
The energy cost calculation takes no account of possible idling times.

A.1 Symbols

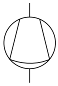
A.1.1 Picture symbols defined by DIN 28004

The following picture symbols are standardised by DIN 28 004, part 3. Only the parts of the norm relevant for compressed air generation are reproduced here.
These picture symbols are used for standard representation in flow diagrams for process systems.
Flow diagrams are used for communication among all persons involved with the development, planning, installation and operation of process systems, and to show the procedure used.

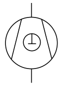
Compressors and pumps



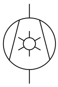
Compressor, general



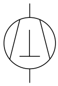
Diaphragm compressor



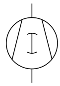
Rotary piston compressor



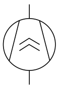
Liquid ring compressor



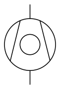
Reciprocating compressor



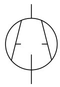
Roots Compressor



Screw compressor

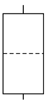


Turbo-Compressor

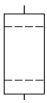


Rotary vane compressor
Rotary Compressor

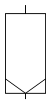
Filters



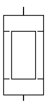
Fluid filter, general
Filter apparatus, general



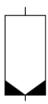
Liquid filter, general



Gas filter, general
Air filter, general

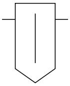


Active carbon filter

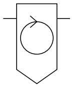


Gas-sorption filter

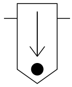
Separators



Separator, general

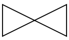


Centrifugal separator, Rotation separator
Dust separator

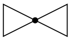


Gravity separator
Deposit chamber

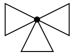
Fittings



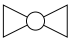
Shut-off fitting, general



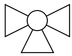
Shut-off through valve



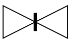
Shut-off 3-way valve



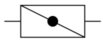
Shut-off through cock



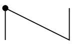
3-way cock



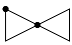
Main slide valve



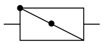
Butterfly valve



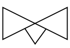
Non-return fitting, general



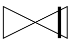
Non-return through-valve



Non-return flap

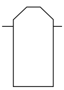


Fitting with constant setting action

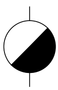


Fitting with safety function

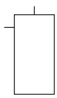
Miscellaneous



Dryer, general



Condensate drain

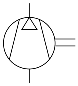


Vessel/receiver, general

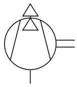
A.1.2 Symbols for contact units and switching devices as per ISO 1219

The following symbols are standardised by ISO 1219 (8.78). Only excerpts from the norm are reproduced here.
The symbols are used to make pneumatic and hydraulic circuit diagrams for describing the operation of respective controls and systems.

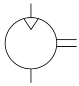
Energy transformation



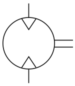
Compressor



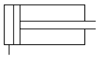
Vacuum pump



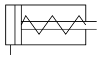
Pneumatic motor with one direction of flow



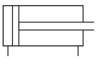
Pneumatic motor with two directions of flow



Single-action cylinder, return stroke by external power

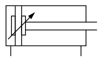


Single-action cylinder, return stroke by spring power



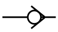
Double-action cylinder

Double-action cylinder with single-side, non-adjustable damping



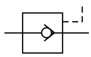
Double-action cylinder with two-side, adjustable damping

Non-return valves



Non-return valve without spring

Non-return valve with spring

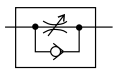


Controlled non-return valve

Flow control valves

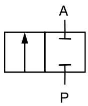
Throttle valve with constant restriction

Throttle valve, adjustable

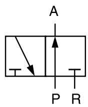


One-way restrictor

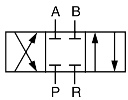
Direction valves



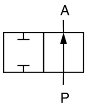
2/2-Way valve
with shut-off neutral position



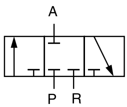
3/2-Way valve
with open neutral position



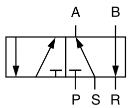
4/3-Way valve
with shut-off middle position



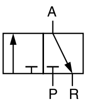
2/2-Way valve
with open neutral position



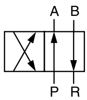
3/3-Way valve
with shut-off middle position



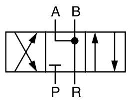
5/2-Way valve



3/2-Way valve with
shut-off neutral position

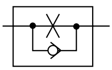


4/2-Way valve

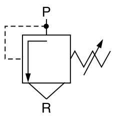


4/3-Way valve
Middle position Work direction
vented

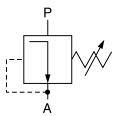
Pressure valves



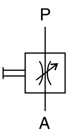
Diaphragm non-return valve



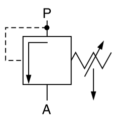
Pressure relief valve, adjustable



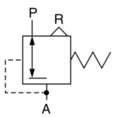
Pressure control valve without
drain aperture, adjustable



Throttle valve
adjustable, manually operated



Emergency valve
adjustable, with air vent

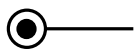


Pressure control valve with drain
aperture, adjustable

Short description of connections

A, B, C	Work line	R, S, T	Drain, vent
P	Pneumatic connection	X, Y, Z	Control lines

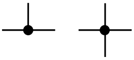
Energy transmission



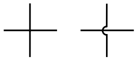
Compressed air source

Work line

Control line



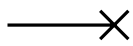
Line connection (fixed)



Line intersection

Flexible line

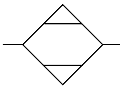
Drain with pipe connection



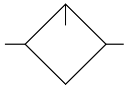
Pressure connection
(closed)

Pressure connection
(with connecting line)

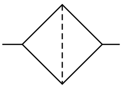
Compressed air receiver



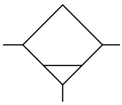
Dryer



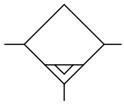
Lubricator



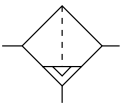
Filter



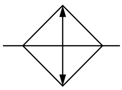
Water separator, hand-operated



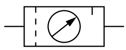
Water separator, automatic emptying



Filter with automatic water separator

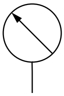


Cooler

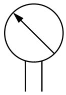


Service unit
(simple representation)

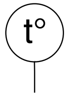
Miscellaneous devices



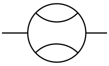
Pressure measuring device



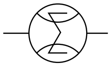
Differential pressure measuring device



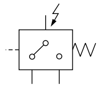
Temperature measuring device



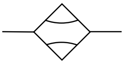
Compressed air measuring device
Ammeter



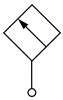
Flow measuring device
Volumemeter



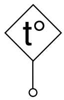
Pressure switch



Flow probe



Pressure probe



Temperature probe

Length

from	x	to • from	x	to
mm	0,03937	inch	25,4	mm
m	3,281	foot	0,3048	m
m	1,094	yard	0,914	m

Surface

from	x	to • from	x	to
mm^2	1,55 x 10-3	sq.inch	645,16	mm^2
cm^2	0,155	sq.inch	6,452	cm^2
m^2	10,76	sq.ft.	0,0929	m^2

Volume

from	x	to • from	x	to
cm^3	0,06102	cu.inch	16,388	cm^3
dm^3(litre)	0,03531	cu.ft.	28,32	dm^3(litre)
dm^3(litre)	0,22	gallon(U.K.)	4,545	dm^3(litre)
dm^3(litre)	0,2642	gallon(US)	3,785	dm^3(litre)
m^3	1,308	cu.yard	0,764	m^3

Volume flow

from	x	to • from	x	to
l/min	0,0353	cfm	28,3	l/min
m^3/min	35,31	cfm	0,0283	m^3/min
m^3/h	0,588	cfm	1,7	m^3/h

Pressure

from	x	to • from	x	to
bar(abs)	14,5	psia	0,07	bar(abs)
bar(abs)	14,5+Atm.	psig	0,07+Atm.	bar(abs)

Force

from	x	to • from	x	to
N	0,2248	pound force(lbf)	4,454	N
kW	1,36	HP	0,736	kW

Temperature

from	x	to • from	x	to
°C	(°C x 1,8) + 32	°F	(°F -32) / 1,8	°C

A

Absorption 85
Adsorption 86
Active carbon Adsorber 100
ARS 59
Atmospheric humidity 72
Autotronic 60

B

Basic units 6
Blaise Pascal law 3
Boyle-Mariotte law 7

C

Choice of compressor 139
Collective line 152
Combined compressor systems 141
Compressed air
- Advantages 14
- Applications 2, 21
- Composition 7
- Costs 204
- Energy costs 205
- Filters 93
- History 1
- Impurities 68, 71
- Loss 122
- Possible applications 18
- Properties 7
- Quality 79

Compressed air consumption
- Cylinders 114
- Cylindrical nozzles 111
- Nozzles 110
- Spray nozzles 113
- Spray paint guns 112
- Tools 115
- Total 120

Compressed air quality
- Planning tips 69

Compressed air receiver 142
- Compressed air storage 142
- Condensate separation 143
- Determining volume 129
- Fittings 147
- Installation 143
- Norm series 129
- Pulsation damping 140
- Set-up 173

Compressed air requirement 110, 117
- Allowances 121
- Mean operation time 117
- Simultaneity factor 118

Compressor 24
- Ambient temperature 170
- Cooling air flow $\dot{V}_K$ 174
- Cycle interval 131
- Diaphragm 29
- Free piston 30
- Heat balance 188
- Liquid ring 32
- Lubricant 50
- Reciprocating piston 27
- Rotary vane 31
- Running time 131
- Screw 33
- Space requirement 172
- Stop time 131
- Summary 26
- Notes for installation 172
- Types of construction 25

Compressor installation 170
Compressor layout
- Piston compressor 133
- Screw compressor 137

Compressors
- Axial 35
- Displacement 24
- Dynamic 24
- Radial 36
- Roots 34

Condensate 102
- Disposal 171

Condesate drain 103
Condensate quantity 74
Condensate separator
- Compressed air receiver 143
- Dust separator 95

Condensate treatment 108
Connection line 151
Control 51
Control unit
- ARS concept 59

Cooling air flow$\dot{V}_K$ 174
Costs
- Compressed air 204
- Compressed air loss 122

D

Dew point 73
Distribution line 150, 151
Drive motor 48
Dryer
- Arrangement 91
- Operating conditions 81

Duotherm heat exchanger
- Cost savings 193
- Duotherm BSW 191, 192

Dust separator 95
Dying 80
- Absorption 85
- Adsorption 86
- Membrane drying 84
- Over-compression 82
- Refrigerated drying 83

F

Filter
- Active carbon 99
- Micro- 97
- Operating pressure 94
- Pre- 96
- Pressure loss Δp 94
- Sterile 101

Filter mechanisms 98
Filter separation rate 93
Fire safety rules 171
Flow 13, 156
Fluidics 5
Frequency control 57

H

Harmful area 38
Heat exchanger 191
Heat reclamation 188

I

Idling control 54
Idling mode (L1) 52
Inspection 144, 146
Installation room 169
Intake filter 49
Intermittent control 54
Intermittent control, delayed 55
Isobar 8
Isochor 8
Isotherm 8

L

Laws
- Pressure Equipment Directive 144

Leakage 123

Leakage quantity 123
- Determining 123, 124

Loudness 198
- level 197

M

Main line 149

MCS 62

Motor cycles
- Determining 132
- Allowed 132

Multiple systems 152

N

Noise
- Safety directives 203
- Effects 202

Norms
- DIN 28004, Part 3 206
- ISO 1219 (8.78) 208

O

ÖWAMAT 109

Oil-water separator 109

Operating modes 52

Operation mode (L2) 53

Output 38

P

Part-load 53

Part-load control 56

Physical fundamentals 8

Picture symbols 206

Piston compressor 37
- Area of application 125
- Assemblies 41
- Control 40
- Cooling 39
- Example installation 187

Pipe inside diameter di
- Bar diagramme 163
- Determining by calculation 161
- Determining by graph 162

Pipe length, equivalent 160

Pipeline
- Dimensioning 158
- Marking 168
- Material 164
- Nominal width 159

Pipeline material 164
- Copper pipes 166
- Plastic pipes 167
- Seamless steel pipes 165
- Stainless steel pipes 165
- Threaded pipes 164

Pipe system 149
- Pressure loss Δp 157
- with compressed air dryer 155
- without dryer 154

Pneumonics 5

Pressure 10

Pressure definitions 51

Pressure dew point 73
- Determining 77
- on pressure relief 78

Pressure loss Δp 156

Pressure ranges 17

Q

Quality classes 79

R

Ratiotronic 60

Refrigeration drying 81

Regeneration 86
- Cold 87
- External hot 89
- Internal hot 88
- Vacuum- 90

Reynolds number Re 156

Ring main 150

Room heating 189
- Economy 190

S

Safety rules
- Compressed air receiver 142

Safety valve 49, 148

Screw compressor 42
- Area of application 127
- Assemblies 47
- Compression process 42
- Example installation 186
- Method of operation 43

SI-System 6

Sound 199

Sound dissemination 200

Sound intensity level 196

Sound level 196
- Assessed, dB (A) 197

Sound perception 195

Sound pressure 196

Stopped/stationary (L0) 52

Stub line 151

Suction rate 38

Supertronic 61

Switching symbols 208

T

Temperature 9

Tongue valve 49

Treatment 68

Types of control 54, 60

V

Vacuum pumps 24

Ventilators 24

Ventilation 174
- Air inlet ducts 181
- Artificial 178
- Compressor rooms 176
- Cool air duct 182
- Ducting 181
- Natural 177

Volume 9

Volume flow $\dot{V}$ 11

Compendium online: www.drucklufttechnik.de

BOGE
Otto-Boge-Str. 1-7
D-33739 Bielefeld
info@boge.com
www.boge.com