Compressed Air Manual

Atlas Copco

Compressed Air Manual

6th Edition

This Manual is published by:
Atlas Copco Compressor AB
Uddvägen 7
S-131 82 Nacka
Sweden

During the production of this material we have gratefully received pictures and contributions from our customers and suppliers, of which we would especially like to name: ABB, Siemens, Vattenfall, AGA and Hamrin Adsorptions & Filterteknik.

Atlas Copco Compressor AB
Project manager: Paul Lindeberg
Production: In Red Design AB
Technical Production: Blueprint AB

ISBN: 91-630-7342-0

Introduction

The Compressed Air Manual is a resource for everyone who wishes to know more about compressed air. This edition, the sixth, is in many respects extended, updated and improved compared to previous editions, of which the last was issued in 1976. Naturally a great deal has happened during these twenty or more years, nevertheless the fundamentals remain and make up the core of this Manual, which has been desired and requested by many.

The Manual addresses the essentials of theoretical and practical issues faced by everyone working with compressed air on a day-to-day basis, from the fundamental theoretical relations to more practical advice and tips. The main addition to this edition is an increased concentration on environmental aspects, air quality issues, energy savings and compressed air economy. Furthermore, we conclude with calculation examples as well as diverse, helpful table information and a complete keyword index. The Manual's contents have been produced by our leading compressed air technicians and I hope that the different sections act both as a textbook for newcomers and a reference book for more experienced users.

It is my belief that the Manual will be useful and perhaps even an enjoyment to many within the industry. Many questions can surely be answered with its help, while others require further investigation. In the case of the latter, I believe the reader can also receive help through the support and structure for continued discussion provided by the Manual. With this in mind, each reader is always more than welcome to contact us for answers to unresolved questions.

Stockholm, September 1998
Atlas Copco Compressor AB

Robert Robertson, MD

Chapter 1
Theory

Chapter 2
Compressors and auxiliary equipment

Chapter 3
Dimensioning and installation

Chapter 4
Economy

Chapter 5
Calculation example

Chapter 6
Appendices

Chapter 1 Theory

1.1 Physics General

1.1.1 The structure of matter

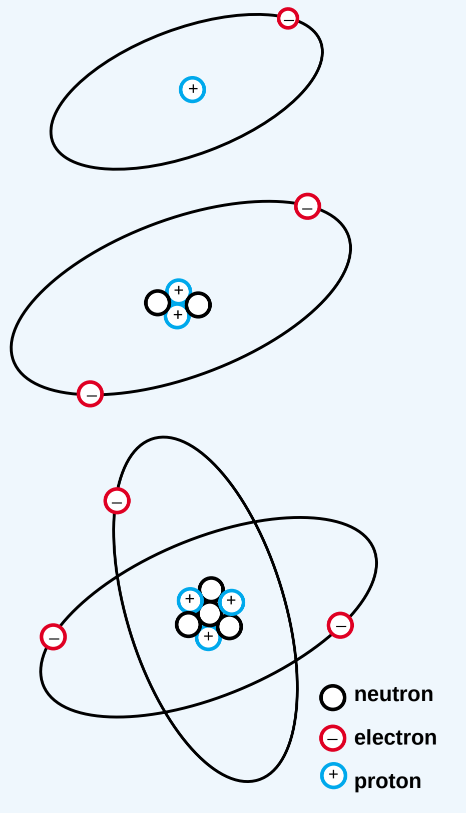
Matter primarily consists of protons, neutrons and electrons. There are also a number of other building blocks however these are not stable.

All of these particles are characterised by four properties: their electrical charge, their rest mass, their mechanical momentum and their magnetic momentum. The number of protons in the nucleus is equal to the atom's atomic number.

The total number of protons and the number of neutrons are approximately equal to the atom's total mass.

This information is a part of the data that can be read off from the periodic system. The electron shell contains the same number of electrons as there are protons in the nucleus. This means the atom is electrically neutral.

1:1



The electron shell gives elements their chemical properties. Here are a few simple examples. Hydrogen (top) has one electron in an electron shell. Helium (middle) has two electrons in an electron shell. Lithium (bottom) has a third electron in a second shell.

The Danish physicist, Niels Bohr, produced a theory as early as 1913 that proved to correspond with reality where he, among others, demonstrated that atoms can only occur in a so-called, stationary state with a determined energy. If the atom transforms from one energy state to another a radiation quantum is emitted, a photon.

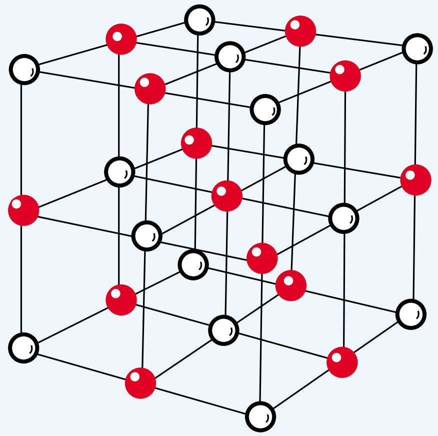
It is these different transitions that makes themselves known in the form of light with different wavelengths. In a spectrograph they appear as lines in the atom's spectrum of lines.

1.1.2 The molecule and the different states of matter

Atoms held together by chemical bonding are called molecules. These are so small that, for example, 1 mm^3 of air at atmospheric pressure contains approx. 2.55 x 10^{16} molecules.

All matter can in principle exist in four different states: solid state, liquid state, gaseous state and plasma state. In the solid state the molecules are tightly packed in a lattice, with strong bonding. At all temperatures above absolute zero a certain degree of molecular movement occurs, in the solid state as a vibration around a balanced position, the faster the greater the temperature becomes. When a substance in

1:2



A salt crystal has a cubic structure. Common table salt (NaCl) is a typical example. The lines show the bonding between the sodium (red) and the chlorine (white).

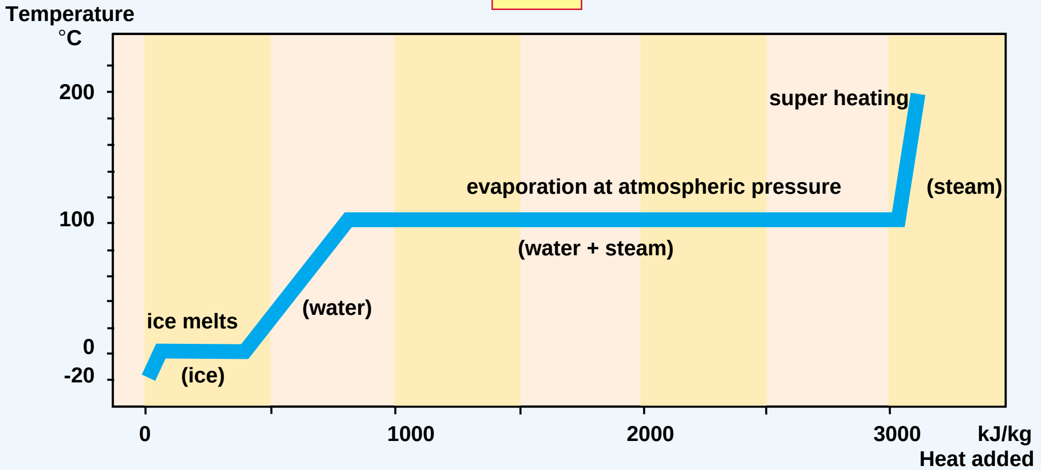
a solid state is heated so much that the movement of the molecules cannot be prevented by the rigid pattern (lattice), they become loose the substance melts and transforms to a fluid. If the liquid is heated more, the bonding of the molecules is broken, and it transforms into a gaseous state during expansion in all directions and mixes with the other gases in the room. When gas molecules are cooled, they loose speed and bond to each other again, and condensation starts. However, if the gas molecules are heated further, they are broken down into individual particles and form a plasma of electrons and atomic nuclei.

1.2 Physical units

1.2.1 Pressure

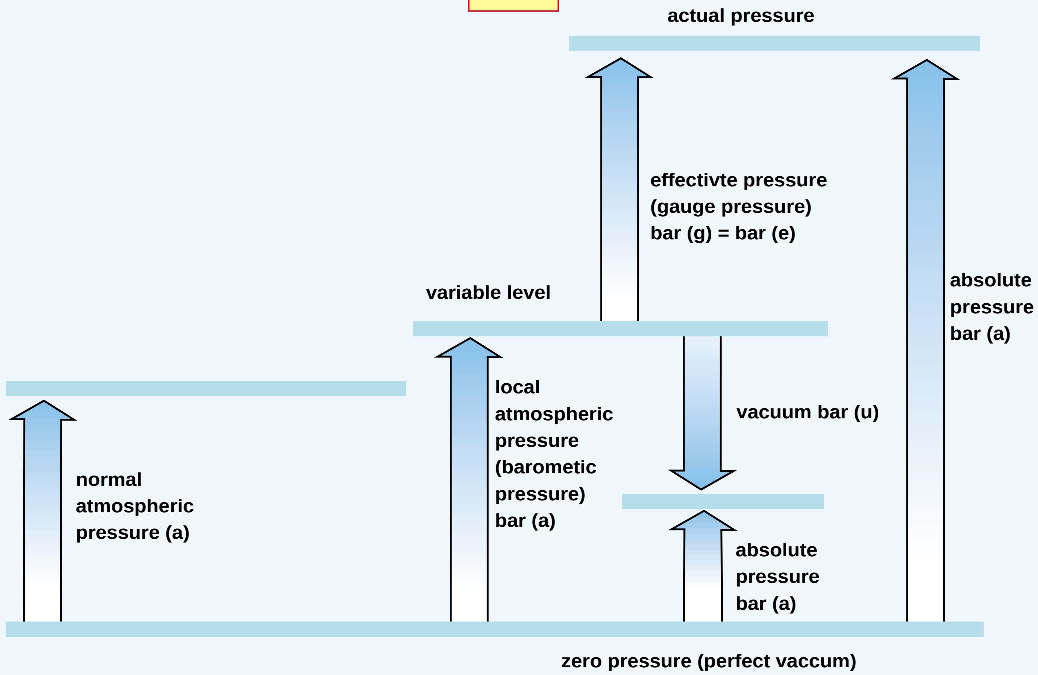
The force on a square centimetre area of an air column, which runs from sea level to the edge of the atmosphere, is about 10.13 N. Therefore the absolute atmospheric pressure at sea level is approx. 10.13 x 10^4 N per square metre, which is also called 1 Pa (Pascal), the SI unit for pressure. A basic dimension analysis shows that 1 bar = 1 x 10^5 Pa. The higher above sea level you are the lower the atmospheric pressure and visa versa.

1:3



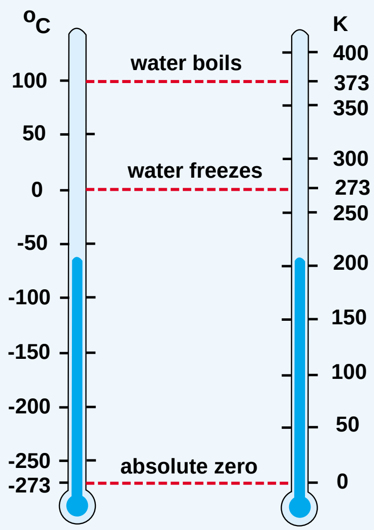
By leading off or applying thermal energy the state of a substance changes. This is how the relation for common water appears.

1:4



Most pressure gauges register the difference between the pressure in a vessel and the local atmospheric pressure. Therefore to find the absolute pressure the value of the local atmospheric pressure must be added.

1:5



This is how the relation between Celsius and Kelvin scales appear. For Celsius, 0° is set at water's freezing point; for Kelvin, 0° is set at absolute zero.

1.2.2 Temperature

The temperature of a gas is more difficult to clearly define than the pressure. Temperature is an indication of the kinetic energy in the molecules. They move more rapidly the higher the temperature and the movement stops at absolute zero. The Kelvin scale is based on this, but otherwise is graduated the same as the Celsius scale. The relation is:

$$T = t + 273.2$$

T = absolute temperature (K)
t = temperature (°C)

1.2.3 Thermal capacity

Thermal capacity refers to the quantity of heat required to increase the temperature of 1 kg of a substance by 1 K. Accordingly, the dimension of the thermal capacity will be J/kg x K. Consequently, the molar thermal capacity is dimensioned J/mol x K. The designations commonly used are:

c_p = thermal capacity at a constant pressure
c_v = thermal capacity at a constant volume
C_p = molar thermal capacity at a constant pressure
C_v = molar thermal capacity at a constant volume

The thermal capacity at a constant pressure is always greater than thermal capacity at a constant volume. The thermal capacity for a substance is however not constant, but rises in general with the temperature.

For practical usage a mean value can frequently be used. For liquids and solid substances it is $c_p \approx c_v \approx c$. The power consumed to heat a mass flow from t_1 to t_2 will then be:

$$Q \approx m \times c \times (t_2 - t_1)$$

Q = heat power (W)
m = mass flow (kg/s)
c = specific thermal capacitivity (J/kg x K)
t = temperature (K)

The explanation as to why c_p is greater than c_v is the expansion work that the gas at a constant pressure must perform. The relation between c_p and c_v called kappa κ, is a function of the number of atoms in the molecule.

$$\kappa = \frac{c_p}{c_v} = \frac{C_p}{C_v}$$

1.2.4 Work

Mechanical work can be defined as the product of a force and the distance over which the force affects a body.

Exactly as for heat, work is an energy that is transferred from one body to another. The difference is that it is a question of force instead of temperature.

An example is the compression of a gas in a cylinder by a moving piston. Compression takes place through a force moving the piston. At the same time energy is transferred from the piston to the enclosed gas. This energy transfer is work in a thermodynamic meaning. The sum of the applied and transmitted energy is always constant. Work can give different results, for example, changes to the potential energy, kinetic energy or the thermal energy.

The mechanical work associated with changes in the volume of a gas or gas mixture is one of the most important processes within thermodynamics. The SI unit for work is Joule. 1 J = 1 Nm = 1 Ws.

1.2.5 Power

Power is work per time unit. SI unit for power is Watt. 1 W = 1 J/s.

For example, the power or energy flow to a drive shaft on a compressor is numerically similar with the heat emitted from the system plus the heat applied to the compressed gas.

1.2.6 Volume rate of flow

The SI unit for volume rate of flow is m^3/s. However, the unit litre/second (l/s) is frequently used, when speaking about the volume rate of flow, for example, given by a compressor. This volume rate of flow is called the compressor's capacity and is either stated as normal litre/second (Nl/s) or as free output air rate (l/s). With the unit normal litre/second (Nl/s) the air flow rate is recalculated to "the normal state", i.e. 1.013 bar and 0°. The unit is primarily used when you wish to specify a mass flow.

With free output air rate the compressor's output air rate is recalculated to its standard intake condition (intake pressure and intake temperature). Accordingly, you state how many litres of air would fill if it once again were allowed to expand to the ambient condition. The relation between the two volume rates of flow is (note that the formula below does not take the humidity into consideration):

$$Q_i = \frac{Q_n \times (273 + T_i) \times 1{,}013}{273 \times p_i}$$

Q_i = volume rate of flow as free output air flow rate (l/s)
Q_n = volume rate of flow as normal litres / second (Nl/s)
T_i = intake temperature (°C)
p_i = intake pressure (bar)

1.3 Thermodynamics

1.3.1 Main principles

Thermodynamics' first main principle is a law of nature that can not be proved, but is accepted without reservation. It says that energy can neither be created nor destroyed and from that it follows that the total energy in a closed system is constant. Thermodynamics' second main principle says that heat can never of "its own effort" be transferred from one source to a hotter source. This means that energy can only be available for work if it can be converted from a higher to a lower temperature level. Therefore in, for example, a heat engine the conversion of a quantity of heat to mechanical work can only take place if a part of this quantity of heat is simultaneously led off without being converted to work.

1.3.2 Gas laws

Boyle's law says that if the temperature is constant, so the product of pressure and volume are constant. The relation reads:

$$p_1 \times V_1 = p_2 \times V_2$$

p = absolute pressure (Pa)
V = volume (m^3)

This means that if the volume is halved during compression, then the pressure is doubled.

Charles's law says that the volume of a gas changes in direct proportion to the change in temperature. The relation reads:

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \Rightarrow \Delta V = \frac{V_1}{T_1} \times \Delta T$$

V = volume (m_3)
T = absolute temperature (K)
ΔV = volume difference
ΔT = temperature difference

The general law of state for gases is a combination of Boyle's and Charles's laws. This states how pressure, volume and temperature affect each other. When one of these variables is changed, this affects at least one of the other two variables. This can be written:

$$\frac{p \times v}{T} = R = \text{gas constant}$$

p = absolute pressure (Pa)
v = specific volume (m^3/kg)
T = absolute temperature (K)
R = $\bar{R}/M$ = individual gas constant (J/kg x k)

The constant R is called the individual gas constant and only concerns the properties of the gas. If the mass m of the gas takes up the volume V, the relation can be written:

$$p \times V = m \times \bar{R} \times T$$

p = absolute pressure (Pa)
V = volume (m^3)
m = mole mass (kmol)
$\bar{R}$ = universal gas constant
= 8314 (J/kmol x K)
T = absolute temperature (K)

1.3.3 Heat transfer

Each heat difference within a body, or between different bodies, always leads to the transfer of heat, so that a temperature balance is obtained. This heat transfer can take place in three different ways: through conductivity, convection or radiation. In reality heat transfer takes place in parallel, in all three ways.

Conductivity takes place between solid bodies or between thin layers of a liquid or gas. Molecules in movement emit their kinetic energy to the adjacent molecules.

Convection can take place as free convection, with the natural movement that occurs in a medium or as forced convection with movement caused by, for example, a fan or a pump. Forced convection gives significantly more intense heat transfer.

All bodies with a temperature above 0°K emit heat radiation. When heat rays hit a body, some of the energy is absorbed and transforms to heat. Those rays that are not absorbed pass through the body or are reflected. Only an absolute black body can theoretically absorb all radiated energy.

In practice heat transfer is the sum of the heat transfer that takes place through conductivity, convection and radiation.

Generally the relation below applies:

$$q = k \times A \times \Delta T \times t$$

q = the quantity of heat (J)
k = total heat transfer coefficient ($W/m^2 \times K$)
A = area (m^2)
ΔT = temperature difference
t = time (s)

Heat transfer frequently occurs between two bodies, separated by a wall. The total heat transfer coefficient is dependent on the heat transfer coefficient on respective sides of the wall and the coefficient of thermal conductivity for the wall. For such a clean, flat wall the relation below applies:

$$1/k = 1/\alpha_1 + d/\lambda + 1/\alpha_2$$

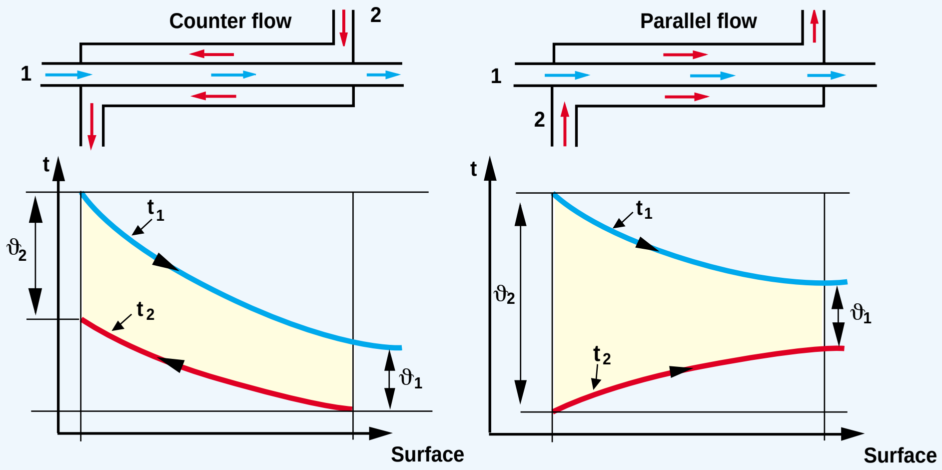
α = heat transfer coefficient on respective sides of the wall ($W/m^2 \times K$)
d = thickness of the wall (m)
λ = coefficient of thermal conductivity for the wall ($W/m \times K$)
k = total heat transfer coefficient ($W/m^2 \times K$)

The transferred quantity of heat, for example, in a heat exchanger, is at each point a function of the prevailing heat difference and the total heat transfer coefficient. Applicable to the entire heat transfer surface is:

$$Q = k \times A \times \vartheta_m$$

Q = transferred quantity of heat (W)
k = total heat transfer coefficient ($W/m^2 \times K$)
A = heat transferring surface (m^2)
ϑ_m = logarithmic mean temperature difference (K)

1:6



This is how the temperature process appears in counter flow respective parallel flow heat exchangers.

The logarithmic mean temperature difference is defined as the relation between the temperature differences at the heat exchanger's two connection sides according to the expression:

$$\vartheta_m = \frac{\vartheta_1 - \vartheta_2}{\ln \dfrac{\vartheta_1}{\vartheta_2}}$$

ϑ_m = logarithmic mean temperature difference (K)

ϑ = the temperature differences (K) according to figure 1:6.

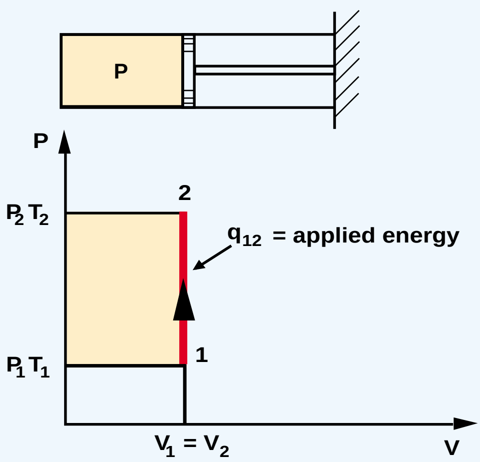
1.3.4 Changes in state

You can follow the changes in state for a gas from one point to another in a p/V diagram. It should really need three axes for the variables p, V and T. With a change in state you move along a curve on the surface in space that is then formed. However, you usually consider the projection of the curve in one of the three planes, usually the p/V plane. Primarily a distinction is made between five different changes in state:

Isochoric process (constant volume), isobaric process (constant pressure), isothermic process (constant temperature) isentropic process (without heat exchange with the surroundings) and polytropic process (where the heat exchange with the surroundings is stated through a simple mathematical function).

1.3.4.1 Isochoric process

1:7



Isochoric change of state means that the pressure increases, while the volume is constant.

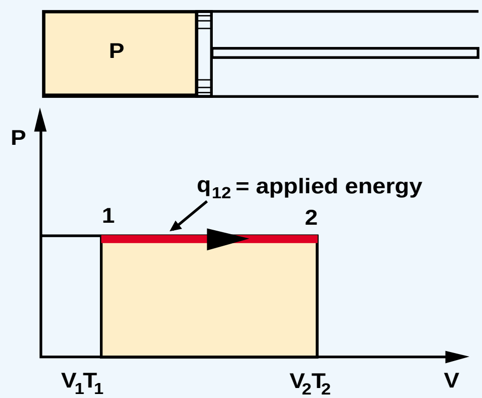
Heating a gas in an enclosed container is an example of the isochoric process. The relation for the applied quantity of heat is:

$$q = m \times c_v \times (T_2 - T_1)$$

q = quantity of heat (J)
m = mass (kg)
c_v = the heat capacity at constany volume (J/kg x K)
T = absolute temperature (K)

1.3.4.2 Isobaric process

1:8



Isobaric change of state means that the volume increases, while the pressure is constant.

Heating of a gas in a cylinder with a constant loaded piston is an example of the isobaric process. The relation for the applied quantity of heat is:

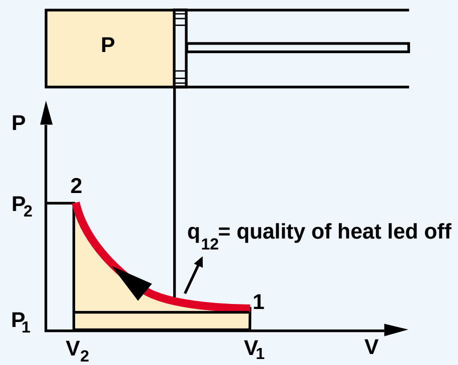
$$q = m \times c_p \times (T_2 - T_1)$$

q = quantity of heat (J)
m = mass (kg)
c_p = the heat capacity at constant pressure (J/kg x K)
T = absolute temperature (K)

1.3.4.3 Isothermic process

If a gas in a cylinder is compressed isothermally, a quantity of heat that is equal to the applied work must be gradually led off. This is practically impossible, as such a slow process can not be realised.

1:9



Isothermic change of state means that the temperature of a gas mixture is constant, when the pressure and volume are changed.

The relation for the quantity of heat led off is:

$$q = m \times R \times T \times 1n \left(\frac{p_2}{p_1}\right)$$

$$q = p_1 \times V_1 \times 1n \left(\frac{V_2}{V_1}\right)$$

q = quantity of heat (J)
m = mass (kg)
R = individual gas constant (J/kg x K)
T = absolute temperature (K)
V = volume (m^3)
p = absolute pressure (Pa)

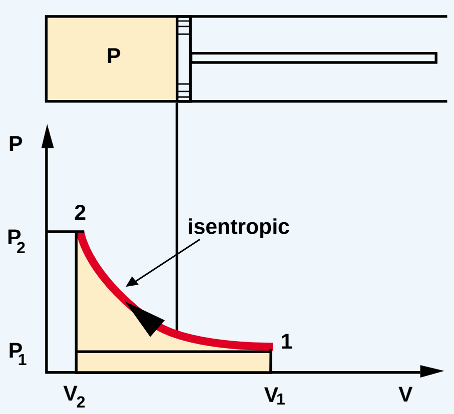
1.3.4.4 Isentropic process

An example of an isentropic process is if a gas is compressed in a fully insulated cylinder without heat exchange with the surroundings. Or if a gas is expanded through a nozzle so quickly that no heat exchange with the surroundings has time to take place. The relation for such a process is:

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{\kappa} \Rightarrow \frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\kappa}{\kappa-1}}$$

p = absolute pressure (Pa)
V = volume (m^3)
T = absolute temperature (K)
$\kappa = \frac{c_p}{c_v}$

1:10



When the entropy in a gas that has been compressed or expanded is constant, no heat exchange with the surroundings takes place. This change in state follows Poisson's law.

1.3.4.5 Polytropic process

The isothermic process involves full heat exchange with the surroundings and the isotropic process involves no heat exchange at all. In reality all processes are something between these extremes and this general process is called polytropic.

The relation for such a process is:

$p \times V^n$ = konstant

p = absolute pressure (Pa)
V = volume (m^3)
n = 0 means isobaric process
n = 1 means isothermic process
n = κ means isentropic process
n = ∞ means isochoric process

1.3.5 Gas flow through a nozzle

The gas flow through a nozzle depends on the pressure ratio on respective sides of the nozzle. If the pressure after the nozzle is lowered the flow increases, however, only until its pressure before the nozzle is approximately double so high. A further reduction of the pressure after the opening does not bring about an increase in flow.

This is the critical pressure ratio and it is dependent on the gas's isentropic exponent (κ). The critical pressure ratio occurs when the flow velocity is equal to the sonic velocity in the nozzle's narrowest section.

The flow becomes supercritical if the pressure after the nozzle is reduced further, under the critical value. The relation for the flow through the nozzle is:

$$G = \alpha \times \psi \times p_1 \times 10^5 \times A \times \sqrt{\left(\frac{2}{R \times T_1}\right)}$$

G = mass flow (kg/s)
α = nozzle coefficient
ψ = flow coefficient
A = minimum flow area (m^2)
R = individual gas constant (J/kg K)
T_1 = absolute temperature before the nozzle (K)
p_1 = absolute pressure before the nozzle (bar)

1.3.6 Flow through pipes

Reynold's number is a dimensionless ratio between inertia and friction in a flowing medium. It is defined as:

$$Re = D \times w \times \eta / \rho \times = D \times w / v$$

D = a characteristic measurement (for example the pipe diameter) (m)
w = mean flow velocity (m/s)
ρ = the density of the flowing medium (kg/m^3)
η = the flowing medium's dynamic viscosity (Pa x s)
v = η/ρ = the flowing medium's kinematic viscosity (m^2/s).

In principal there are two types of flow in a pipe. With Re<2000 the viscous forces dominate in the media and the flow becomes laminar. This means that different layers of the medium move in relation to each other in good order. The velocity distribution across the laminar layers is usually parabolic shaped. With Re≥4000 the inertia forces dominate the flowing medium and the flow becomes turbulent, with particles that move randomly in the flow's cross section. The velocity distribution across a layer with turbulent flow becomes diffuse.

In the critical area, between Re≤2000 and Re≥4000, the flow conditions are undetermined, either laminar or turbulent or a mixture of the both. The conditions are governed by factors such as the surface smoothness of the pipe or other disturbances.

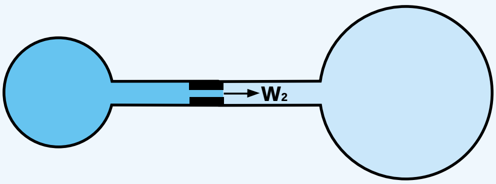
To start a flow in a pipe requires a specific pressure difference or pressure drop, to overcome the friction in the pipe and couplings. The size of the pressure drop depends on the diameter of the pipe, its length and form as well as the surface smoothness and Reynold's number.

1.3.7 Throttling

When an ideal gas flows through a restrictor, with a constant pressure before and after the restrictor, the temperature remains constant. However, there occurs a pressure drop across the restrictor, through the inner energy transforming into kinematic energy, which is why the temperature falls. However, for real gases this temperature change becomes lasting, even if the gas's energy content is constant. This is called the Joule Thomson effect. The temperature change is equal to the pressure change across the throttling multiplied by the Joule Thomson coefficient.

If the flowing medium has a sufficiently low temperature (≤+329°C for air) a temperature drop occurs across the restrictor, but if the flow medium is hotter, a temperature increase occurs. This condition is used in several technical applications, for example, in refrigeration technology and the separation of gases.

1:11



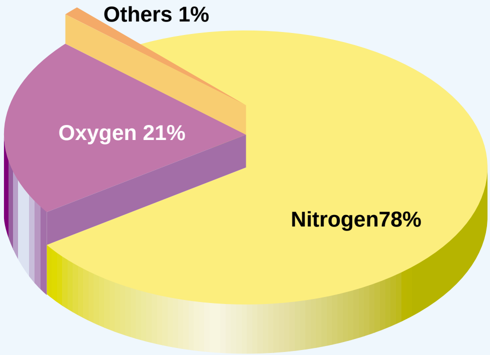
When an ideal gas flows through a small opening between two large containers, the energy becomes constant and no heat exchange takes place. However, a pressure drop occurs with the passage through the restrictor.

1.4 Air

1.4.1 Air in general

Air is a colourless, odourless and tasteless gas mixture. It consists of many gases, but primarily oxygen and nitrogen. Air can be considered a perfect gas mixture in most calculation contexts. The composition is relatively constant, from seal level and up to an altitude of 25 kilometres.

1:12



Air is a gas mixture that primarily consists of oxygen and nitrogen. Only approx. 1% is made up of other gases.

Air is always more or less contaminated with solid particles, for example,dust, sand, soot and salt crystals. The degree of contamination is higher in populated areas, less in the countryside and at higher altitudes.

Air is not a chemical substance, but a mechanically mixed substance. This is why it can be separated into its constituent elements, for example, by cooling.

1.4.2 Moist air

Air can be considered as a mixture of dry air and water vapour. Air that contains water vapour is called moist air, but the air's humidity can vary within broad limits. Extremities are completely dry air and air saturated with moisture. The maximum water vapour pressure that air can hold increases with rising temperatures. A maximum water vapour pressure corresponds to each temperature.

Air usually does not contain so much water vapour that maximum pressure is reached. Relative vapour pressure (also known as relative humidity) is a state between the actual partial vapour pressure and the saturated pressure at the same temperature.

The dew point is the temperature when air is saturated with water vapour. Thereafter with a fall in temperature the condensation of water takes place. Atmospheric dew point is the temperature at which water vapour starts to condense at atmospheric pressure. Pressure dew point is the equivalent temperature with increased pressure. The following relation applies:

$$(p - \varphi \times p_s) \times 10^5 \times V = R_a \times m_a \times T$$
$$\varphi \times p_s \times 10^5 \times V = R_v \times m_v \times T$$

p = total absolute pressure (bar)
p_s = saturation pressure at the actual temperature (bar)
φ = relative vapour pressure
V = total volume of the moist air (m^3)
R_a = gas constant for dry air = 287.1 J/Kg x K
R_v = gas constant for water vapour = 461.3 J/Kg x K
m_a = mass of the dry air (kg)
m_v = mass of the water vapour (kg)
T = absolute temperature of the moist air (K)

1.5 Types of compressors

1.5.1 Two basic principles

There are two basic principles for the compression of air (or gas), the displacement principal and dynamic compression. Among displacement compressors are, for example, piston compressors and different types of rotary compressors. They are the most common compressors in most countries.

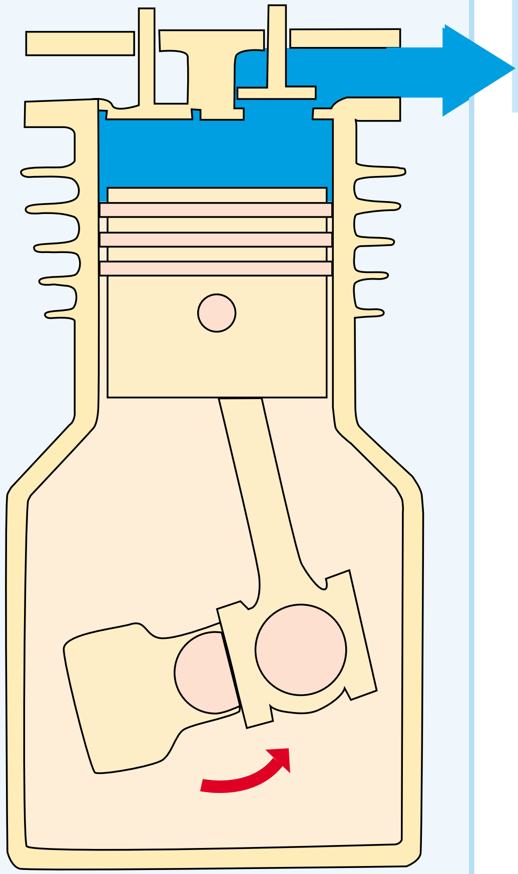
On a piston compressor for example, the air is drawn into a compression chamber, which is closed from the inlet. Thereafter the volume of the chamber decreases and the air is compressed. When the pressure has reached the same level as the pressure in the outlet manifold, a valve is opened and the air is discharged at a constant pressure, under continued reduction of the compression chamber's volume.

In dynamic compression air is drawn into a rapidly rotating compression impeller and accelerates to a high speed. The gas is then discharged through a diffuser, where the kinetic energy is transformed to static pressure. There are dynamic compressors with axial or radial flow. All are suitable for large volume rates of flow.

1.5.2 Displacement compressors

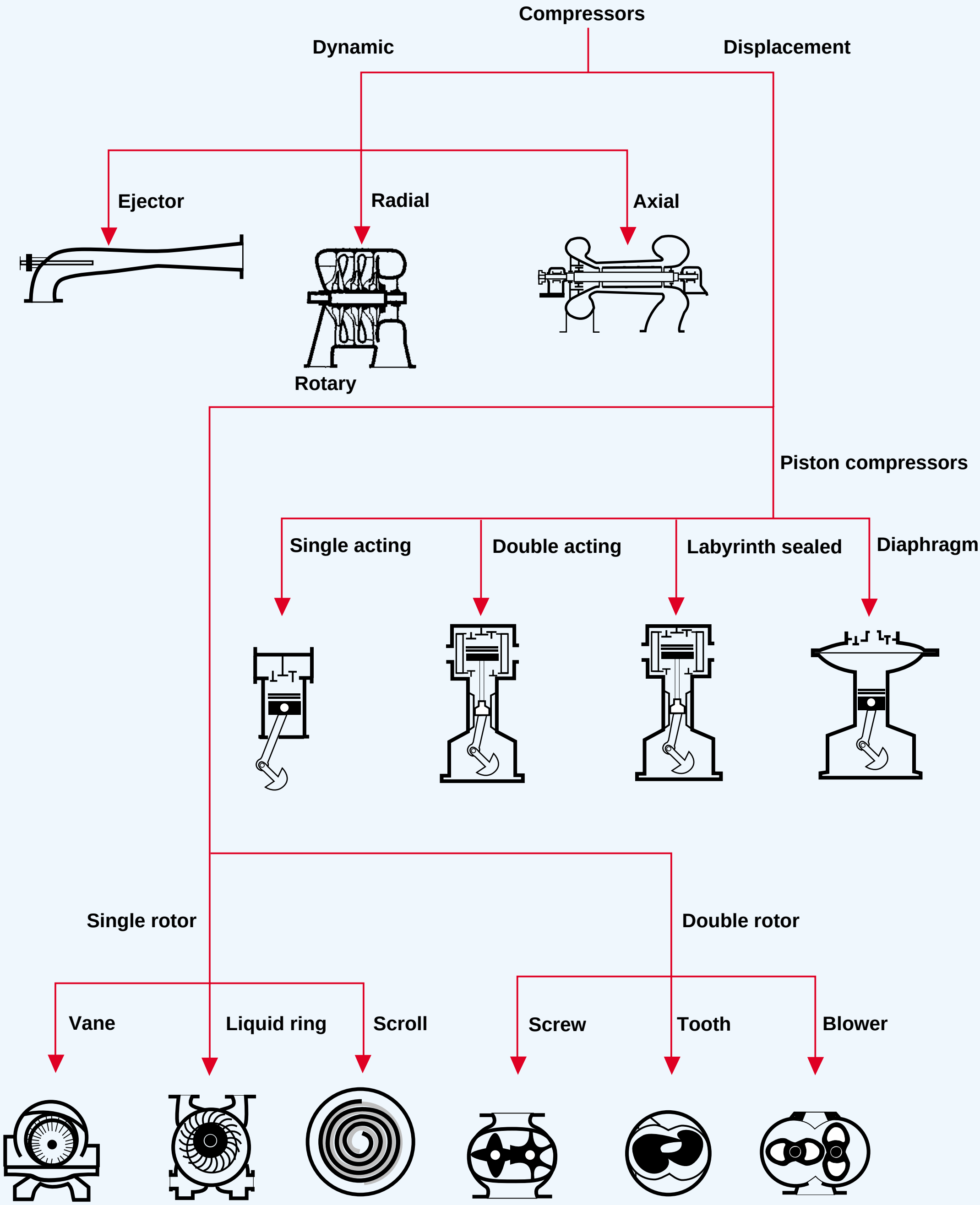
A bicycle pump is the simplest form of a displacement compressor, where air is drawn into a cylinder and is compressed by a moving piston. The piston compressor has the same operation principle, with a piston whose forward and backward movement is accomplished by a connecting rod and a rotating crankshaft. If only one side of the piston is used for compression this is called single acting. If both the piston's top and undersides are used the compressor is called double acting. The difference between the pressure on the inlet side and the pressure on the outlet side is a measurement of the compressor's work.

1:13



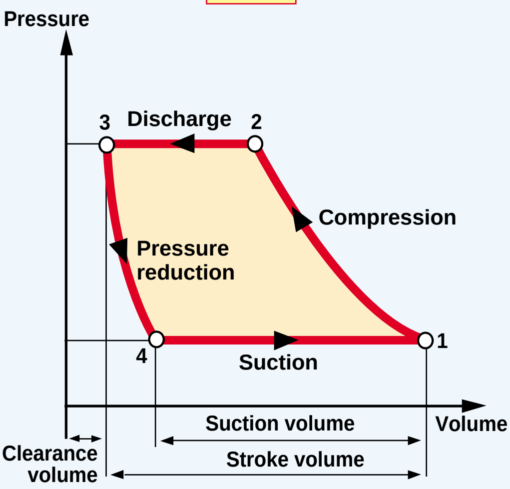
Single stage, single acting piston compressor.

The pressure ratio is the relation between absolute pressure on the inlet and outlet sides. Accordingly, a machine that draws in air at atmospheric pressure and compresses it to 7 bar overpressure works with a pressure ratio of $(7 + 1)/1 = 8$.



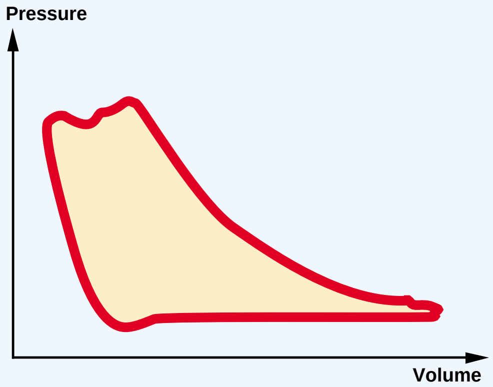
The chart shows the most common types of compressor, divided according to their working principles. They can also be divided according to other principles, for example, air or liquid cooled, stationary or portable, etc.

1:15



This is how a piston compressor works in theory with self-acting valves. The p/V diagram shows the theoretical process, without losses with complete filling and emptying of the cylinder.

1:16



This is how the actual p/V diagram can appear for a piston compressor. The pressure drop on the inlet side and the overpressure on the discharge side are minimised primarily by giving the compressor sufficient valve area.

1.5.3 The compressor diagram for displacement compressors

Figure 1:15 illustrates a theoretical compressor diagram and figure 1:16 illustrates a real compressor diagram for a piston compressor. The stroke volume is the cylinder volume that the piston travels during the suction stage. The clearance volume is the area that must remain at the piston's turning point for mechanical reasons, together with the area required for the valves, etc.

The difference between the stroke volume and the suction volume is due to the expansion of the air remaining in the clearance volume before suction can start. The difference between the theoretical p/V diagram and the real diagram is due to the practical design of a compressor, e.g. a piston compressor. The valves are never fully sealed and there is always a degree of leakage between the piston and the cylinder wall. In addition, the valves can not open and close without a delay, which results in a pressure drop when the gas flows through the channels. Due to reasons of design the gas is also heated when it flows into the cylinder.

Compression work with isothermic compression becomes:

$W = p_1 \times V_1 \times 1n(p_2/p_1)$

Compression work with isentropic compression becomes:

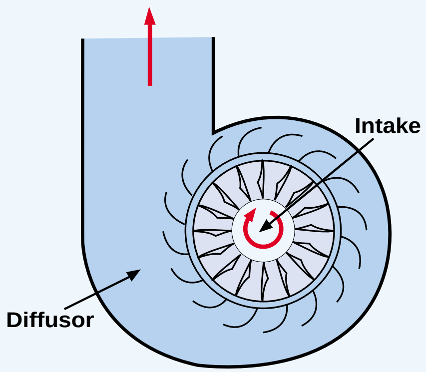
$$W = \frac{\kappa}{\kappa\text{-}1} \times (p_2V_2 - p_1V_1)$$

W = compression work (J)
p_1 = initial pressure (Pa)
V_1 = initial volume (m^3)
p_2 = final pressure (Pa)
κ = isentropic exponent in most cases x $\kappa \approx 1{,}3 - 1{,}4$ applies.

These relations show that more work is required for isentropic compression than with isothermic compression. In reality the requisite work lies between the limits ($\kappa \approx 1{,}3 - 1{,}4$).

1.5.4 Dynamic compressors

1:17



Radial compressor

A dynamic compressor is a flow machine where the pressure increase takes place at the same time as the gas flows. The flowing gas accelerates to a high velocity by means of the rotating blades, after which the velocity of the gas is transformed to pressure when it is forced to decelerate under expansion. Depending on the main direction of the flow they are called radial or axial compressors.

In comparison with displacement compressors, dynamic compressors have a characteristic where a small change in the working pressure results in a large change in the capacity. See figure 1:19.

Each speed has an upper and lower capacity limit. The upper limit means that the gas's flow velocity reaches sonic velocity. The lower limit means that the counter pressure is greater than the compressor's pressure build-up, which means return flow in the compressor. This in turn results in pulsation, noise and the risk for mechanical damage.

1.5.5 Compression in several stages

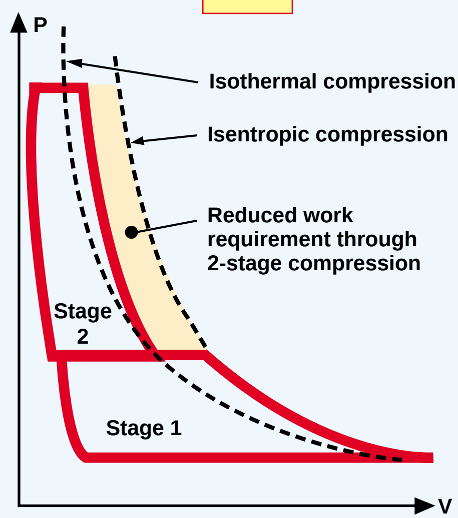
Theoretically a gas can be compressed isentropically or isothermally. This can take place as a part of a reversible process. If the compressed gas could be used immediately, at its final temperature after compression, the isentropic process would have certain advantages. In reality the gas can rarely be used directly without being cooled before use. Therefore the isothermal process is preferred, as this requires less work.

In practice attempts are made to realise this process by cooling the gas during compression. How much you can gain by this is shown, for example, with an effective working pressure of 7 bar that theoretically requires 37% higher output for isentropic compression compared with isothermal compression.

A practical method to reduce the heating of the gas is to divide the compression into several stages. The gas is cooled after each stage, to then be compressed further. This also increases the efficiency, as the pressure ratio in the first stage is reduced. The power requirement is at its lowest if each stage has the same pressure ratio.

The more stages the compression is divided into the closer the entire process

1:18



The shaded area represents the work saved by dividing the compression into two stages.

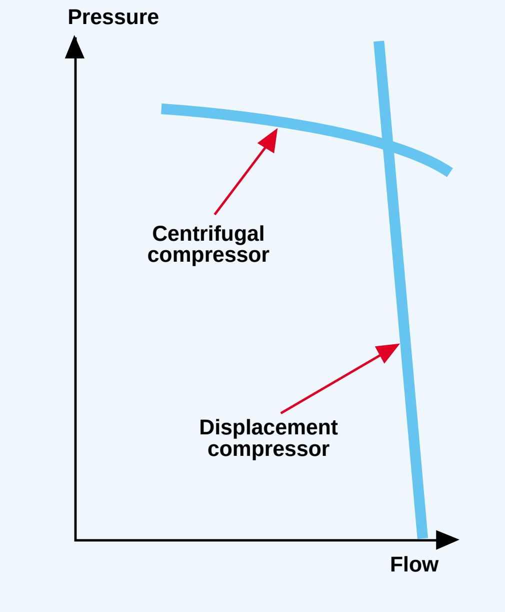
gets to be isothermal compression. However there is an economic limit for how many stages a real installation can be designed with.

1.5.6 Comparison between displacement and centrifugal compressors

The capacity curve for a centrifugal compressor differs significantly from an equivalent curve for a displacement compressor. The centrifugal compressor is a machine with a variable capacity and constant pressure. On the other hand a displacement compressor is a machine with a constant capacity and a variable pressure.

Examples of other differences is that a displacement compressor gives a higher pressure ratio even at a low speed, unlike the more significantly higher speed centrifugal compressors. The centrifugal compressors are well suited to large air flow rates.

1:19



This is how the load curves for centrifugal respective displacement compressors appear when the load is changed at a constant speed.

1.6 Electricity

1.6.1 Basic terminology and definitions

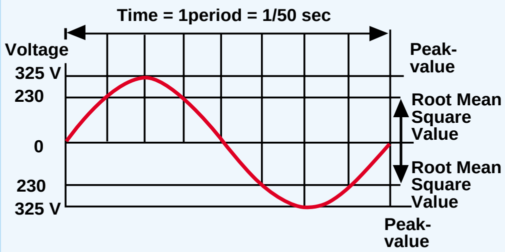
The alternating current used for example to power lighting and motor operations regularly changes strength and direction in a sinusoidal variation. The current strength grows from zero to a maximum value, then falls to zero, changes direction, grows to a maximum value in the opposite direction to then become zero again. The current has then completed a period. The period T is the time in seconds under which the current has gone through all its values. The frequency states the number of complete cycles per second.

$f = 1/T$
f = frequency (Hz)
T = time for a period (s)

When speaking about current or voltage it is usually the root mean square value that is meant. With a sinusoidal current the relation for the current's respective voltage's root mean square value is:

$$\text{root mean square value} = \frac{\text{peak value}}{\sqrt{2}}$$

1:20



This is how a period of a sinusoidal voltage appears (50 Hz).

Voltage under 50V is called extra low voltage. Voltage under 1000V is called low voltage. Voltage over 1000V is called high voltage. Standard voltages at 50 Hz are 230/400V and 400/690V.

1.6.2 Ohm's law for alternating current

An alternating current that passes a coil gives rise to a magnetic flow. This flow changes strength and direction in the same way as the current. When the flow changes an emf (electromotive force) is generated in the coil, according to the laws of induction. This emf is counter directed to the connected pole voltage. The phenomenon is called self-induction.

Self-induction in an alternating current unit partly gives rise to phase displacement between the current and the voltage, and partly to an inductive voltage drop. The unit's resistance to the alternating current becomes apparently greater than that calculated or that measured with direct current.

Phase displacement between the current and voltage is represented by the angle φ. Inductive resistance (reactance) is represented by X. Resistance is represented by R. Apparent resistance in a unit or conductor is represented by Z.

Applicable for impedance is:

$$Z = \sqrt{R^2 + X^2}$$

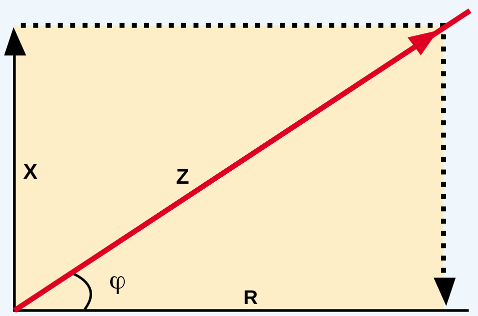
Z = impedance (Ω)
R = resistance (Ω)
X = reactance (Ω)

Ohm's law for alternating current reads:

U = I x Z

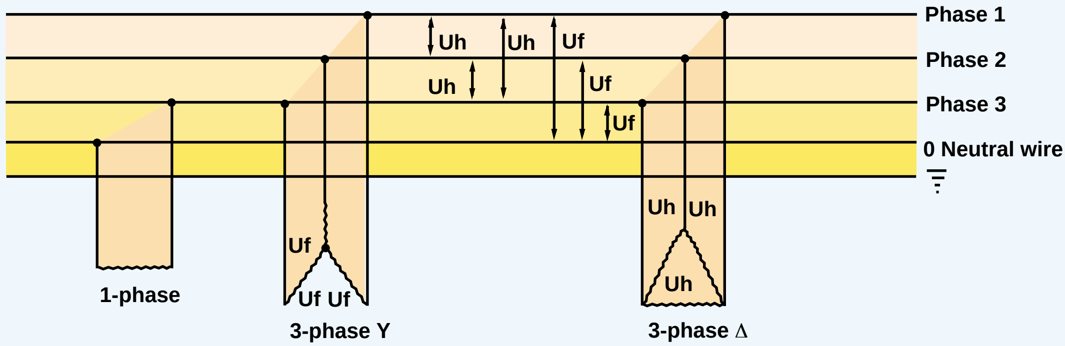
U = voltage (V)
I = current (A)
Z = impedance (Ω)

1:21



Relation between Reactance (X) – Resistance (R) – Impedance (Z) – Phase displacement (φ).

1:22



This is how the different connection options for a three-phase system appears. The voltage between the two phase conductors is called the main voltage (Uh). The voltage between one phase conductor and the neutral wire are called phase voltage (Uf). The phase voltage = Main voltage/$\sqrt{3}$.

1.6.3 Three-phase system

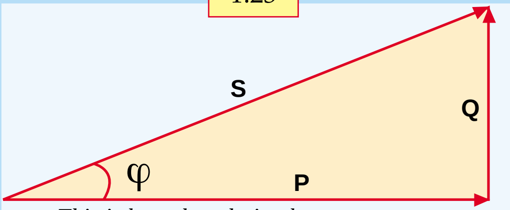
Three-phase alternating current is produced in a generator with three separate windings. All values on the sinusoidal voltage are displaced 120° in relation to each other.

Different units can be connected to a three-phase unit. A single phase unit can be connected between the phase and zero. Three-phase units can be connected in two ways, star (Y) or delta (Δ) connection. With the star connection a phase voltage lies between the outlets. With a delta connection a main voltage lies between the outlets.

1.6.4 Power

Active power, P, is the useful power that can be used for work. Reactive power, Q, is the "useless" power and can not be used for work. Apparent power, S, is the power that must be consumed from the mains supply to gain access to active power. The relation between active, reactive and apparent power is usually illustrated by a power triangle.

1:23



This is how the relation between apparent power (S), reactive power (Q) and active power (P) The angle φ between S and P gives the power factor cosφ.

The following relation applies:

Single phase:

$P = U \times I \times \cos\varphi$

$Q = U \times I \times \sin\varphi$

$S = U \times I$

$\cos\varphi = P/S$

Three phase:

$P = \sqrt{3} \times U_h \times I \times \cos\varphi$

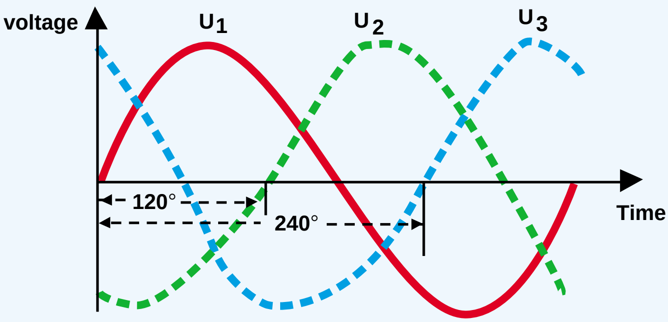
$Q = \sqrt{3} \times U_h \times I \times \sin\varphi$

$S = \sqrt{3} \times U_h \times I$

$\cos\varphi = P/S$

U = voltage (V)
U_h = main voltage, (V)
U_f = phase voltage
I = current (A)
I_h = main current (A)
I_f = phase current (A)
P = active power (W)
Q = reactive power (VAr)
S = apparent power (VA)
φ = phase angle
$\cos\varphi$ = power factor

1:24



The displacement between the generator's windings gives a sinusoidal voltage curve on the system. The maximum value is displaced at the same interval as the generator's windings.

1.6.5 The electric motor

The most common electric motor is a three phase, short circuit induction motor. This type of motor can be found within all industries. Silent and reliable, it is a part of most systems, for example, compressors. The electric motor consists of two main parts, the stationary stator and the rotating rotor. The stator produces a rotating magnetic field and the rotor converts this energy to movement, i.e. mechanical energy.

The stator is connected to the mains supply's three phases. The current in the stator windings give rise to a rotating magnetic force field, which induces currents in the rotor and gives rise to a magnetic field there too. The interaction between the stator's and the rotor's magnetic fields creates turning torque, which makes the rotor shaft rotate.

1.6.5.1 Rotation speed

If the motor shaft should rotate at the same speed as the magnetic field, the induced current in the rotor would at the same time be zero. However, due to losses in, for example the bearings, this is impossible and the speed is always approx. 1-5% lower than the magnetic field's synchronous speed (slip). Applicable for this synchronous speed is:

$$n = 2 \times f \times 60/p$$

n = synchronous speed (r/min)
f = main supply's frequency (Hz)
p = number of poles

1.6.5.2 Efficiency

Energy conversion in a motor does not take place without losses. These are due to, among others, resistive losses, ventilation losses, magnetisation losses and friction losses. Applicable for efficiency is:

$$\eta = \frac{P_2}{P_1}$$

η = efficiency
P_2 = stated power, shaft power (W)
P_1 = applied power (W)

It is always the stated power, P_2, stated on the motor's rating plate.

1.6.5.3 Insulation class

The insulation material in the motor's windings is divided into insulation classes in accordance with IEC 85 (International Electrotechnical Commission). A letter corresponding to the temperature, which is the upper limit for the isolation's calculated application area, designates each class. If the upper limit is exceeded by 10°C the service life of the insulation is shortened by about half.

Insulation class	B=130°C	F=155°C	H=180°C
Ambient temp. °C	40	40	40
Temp. increase °C	80	105	125
Thermal marginal °C	10	10	15
Max. final temp. °C	130	155	180

1.6.5.4 Protection classes

Protection classes state, according to IEC 34-5, how the motor is protected against contact and water. These are stated with the letters IP and two digits. The first states the protection against contact and penetration by a solid object. The other digit states the protection against water. For example IP23 represents: (2) protect against solid objects greater than 12 mm, (3) protect against direct sprays of water up to 60° from the vertical. IP 54: (5) protection against dust, (4) protection against water sprayed from all directions. IP 55: (5) protection against dust, (5) protection against low-pressure jets of water from all directions.

1.6.5.5 Cooling methods

Cooling methods state, according to IEC 34-6, how the motor shall be cooled. This is designated with the letters IC and two digits. For example IC 01 represents: Free circulation, own ventilation and IC 41: Jacket cooling, own ventilation.

1.6.5.6 Installation method

The installation method states, according to IEC 34-7, how the motor should be installed. This is designated by the letters IM and four digits. For example IM 1001 represents: two bearings, shaft with free journalled end, stator body with feet. IM 3001: two bearings, shaft with free journalled end, stator body without feet, large flange with plain securing holes.

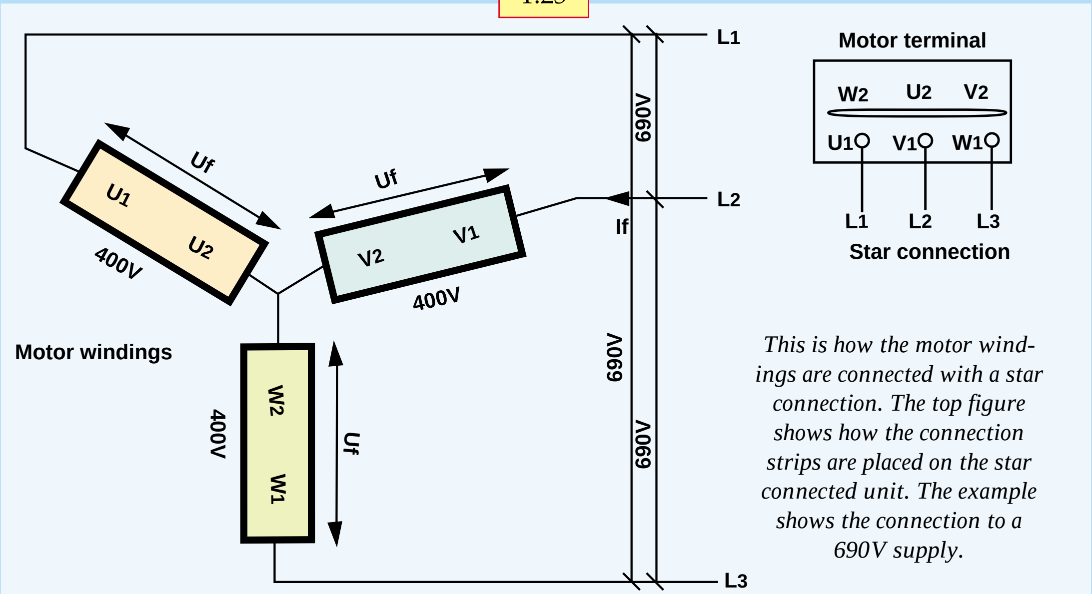
1.6.5.7 Star (Y) and delta (Δ) connections

A three-phase electric motor can be connected in two ways, star (Y) or delta (Δ). The winding phases in a three-phase motor are marked U, V and W (U1-U2; V1-V2; W1-W2). With the star (Y) connection the "ends" of motor winding's phases are joined together to form a zero point, which looks like a star (Y).

A phase voltage (phase voltage = main voltage/$\sqrt{3}$; for example 400V = 690/$\sqrt{3}$) will lie across the windings. The current I_h in towards the zero point becomes a phase current and accordingly a phase current will flow $I_f = I_h$ through the windings.

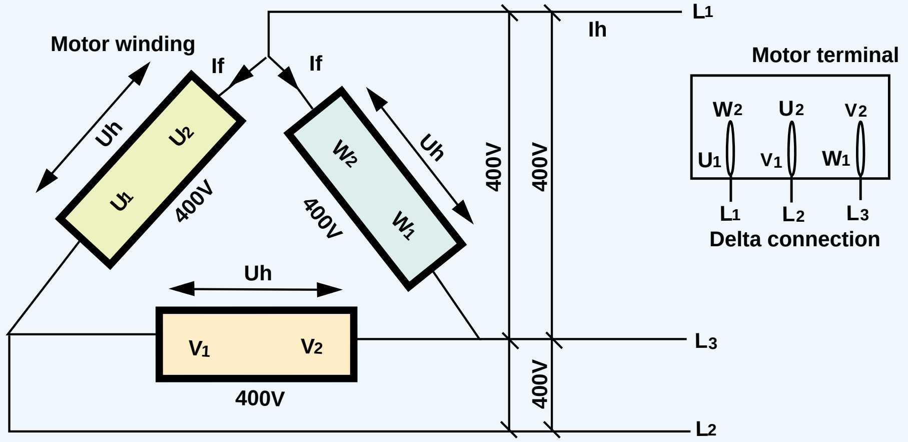
With the delta (Δ) connection you join the beginning and ends between the different phases, which then form a delta (Δ). There will then lie a main voltage across the windings. The current Ih into the motor is the main current and this will be divided between the windings and give a phase current through these, $I_h/\sqrt{3} = I_f$. The same motor can be connected as 690V star connection or 400V delta connection. In both cases the voltage across the windings will be 400V. The current to the motor will be lower with a 690V star connection than with a 400V delta connection. The relation between the current levels is $\sqrt{3}$.

1:25



This is how the motor windings are connected with a star connection. The top figure shows how the connection strips are placed on the star connected unit. The example shows the connection to a 690V supply.

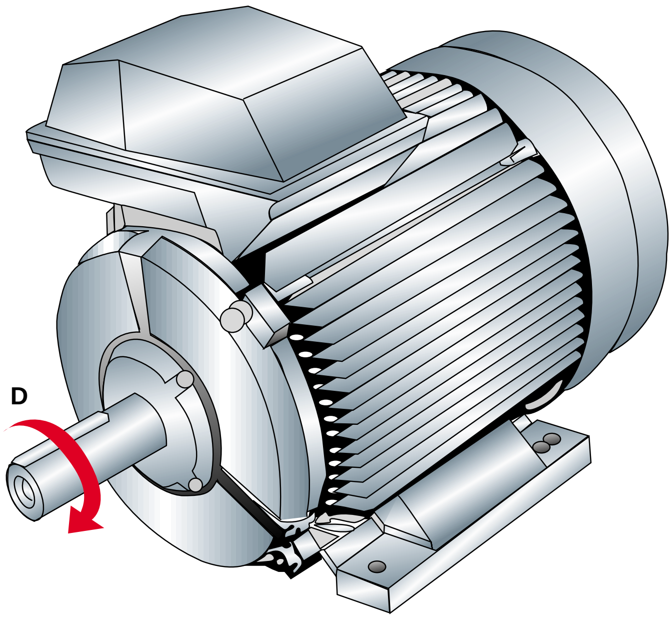
1:26



This is how the motor windings are connected with a delta connection. The top figure shows how the connection strips are placed on the delta connected unit. The example shows the connection to a 400V supply.

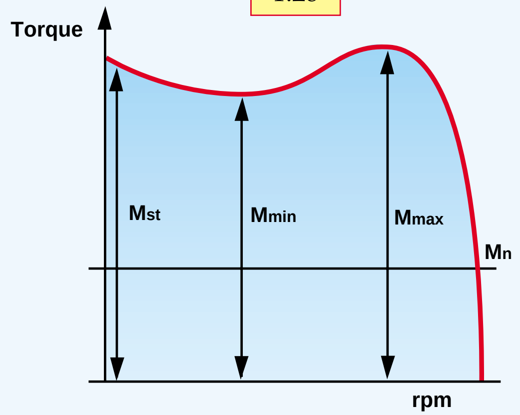
On the motor plate it can, for example, state 690/400 V. This means the star connection is intended for the higher voltage and the delta connection for the lower. The current, which can also be stated on the plate, shows the lower value for the star connected motor and the higher for the delta connected motor.

1:27



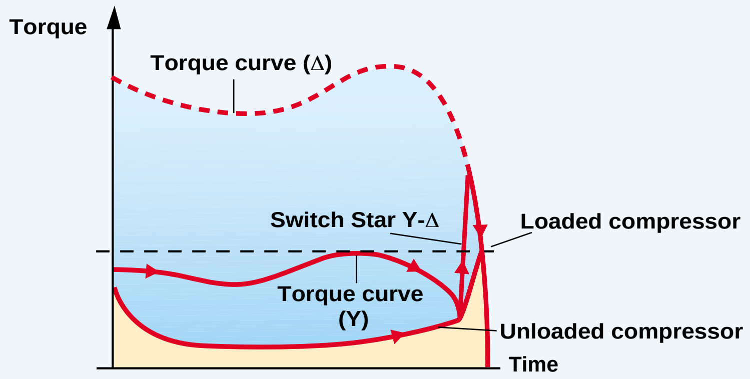
The mains supply is connected to a three-phase motor's terminals marked U, V and W. The phase sequence is L1, L2 and L3. This means the motor will rotate clockwise seen from "D" the drive end. To make the motor rotate anticlockwise two of the three conductors connected to the starter or to the motor are switched. Check the operation of the cooling fan when rotating anticlockwise.

1:28



The torque curve for a short circuited, induction motor. When the motor starts the torque is high, during acceleration the torque first drops a little, to then rise to its max. value before dropping. M = torque, Mst = start torque, Mmax = max torque ("cutting torque"), Mmin = min. torque ("saddle torque"), Mn = rated torque.

1:29



A star/delta started induction motor's torque cure inserted in a torque curve for a screw compressor. The compressor is unloaded (idling) during star operations. When the speed has reached approx. 90-95% of the rated speed the motor is switched to the delta mode, the torque rises, the compressor is loaded and finds its working point.

1.6.5.8 Torque

An electric motor's turning torque is an expression for the rotor's turning capacity. Each motor has a maximum torque. A load above this torque means that the motor does not have the power to rotate. With a normal load the motor works significantly under its maximum torque, however, the start phase involves an extra load. The characteristics of the motor are usually presented in a torque curve.

Chapter 2 Compressors and auxiliary equipment

2.1 Displacement compressors

2.1.1 Displacement compressors in general

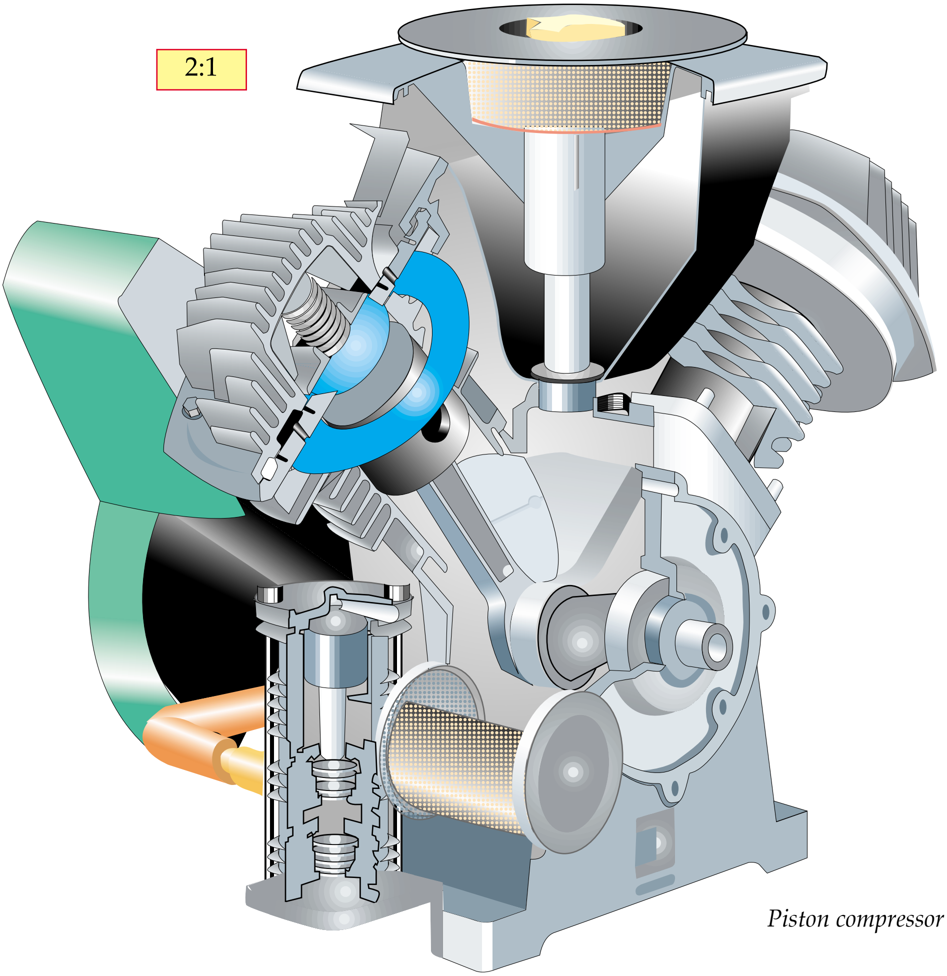
A displacement compressor is characterised by enclosing a volume of gas or air and then increasing the pressure by reducing the area of the enclosed volume.

2.1.2 Piston compressors

The piston compressor is the oldest and most common of all compressors. It is available as single or double acting, oil lubricated or oil–free with a different number of cylinders in different configurations. With the exception of really small compressors with vertical cylinders, the V configuration is the most common for small compressors.

On double acting, large compressors the L-type with vertical low pressure cylinder and horizontal high pressure cylinder, offers immense benefits and is why this is the most common design.

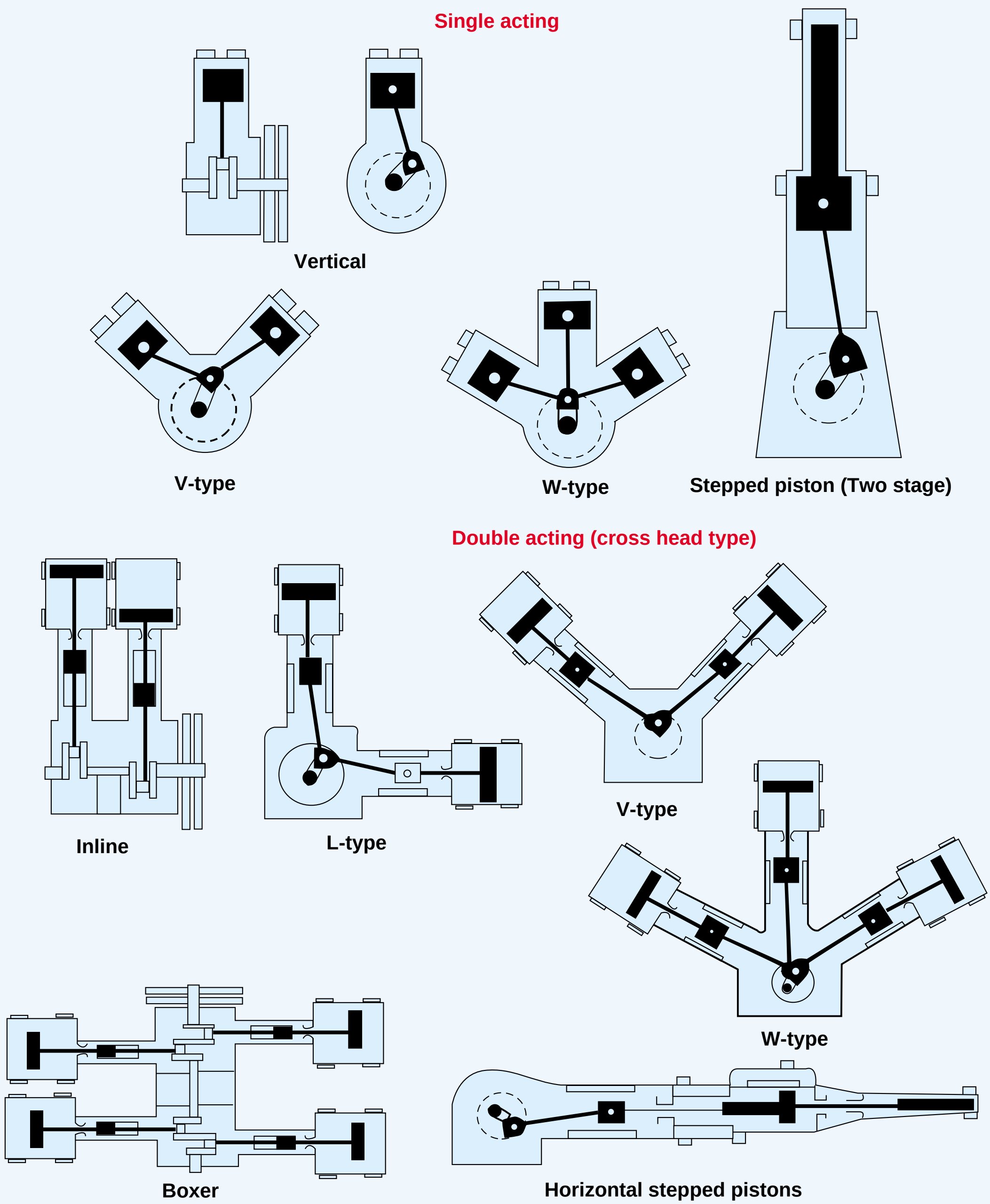
2:1



Piston compressor

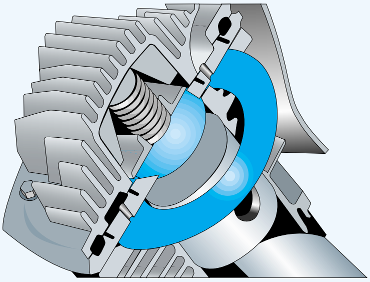
Oil lubricated compressors normally work with splash lubrication or pressure lubrication. Most compressors have self-acting valves. A self-acting valve opens and closes through pressure differences on respective sides of the valve disk.

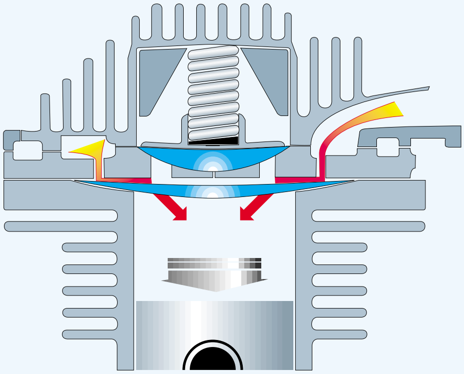
2.2

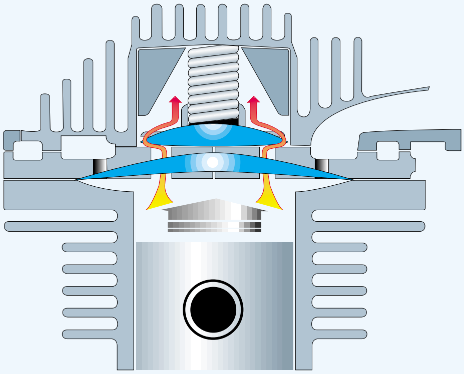


Examples of cylinder placement on piston compressors.

2:3





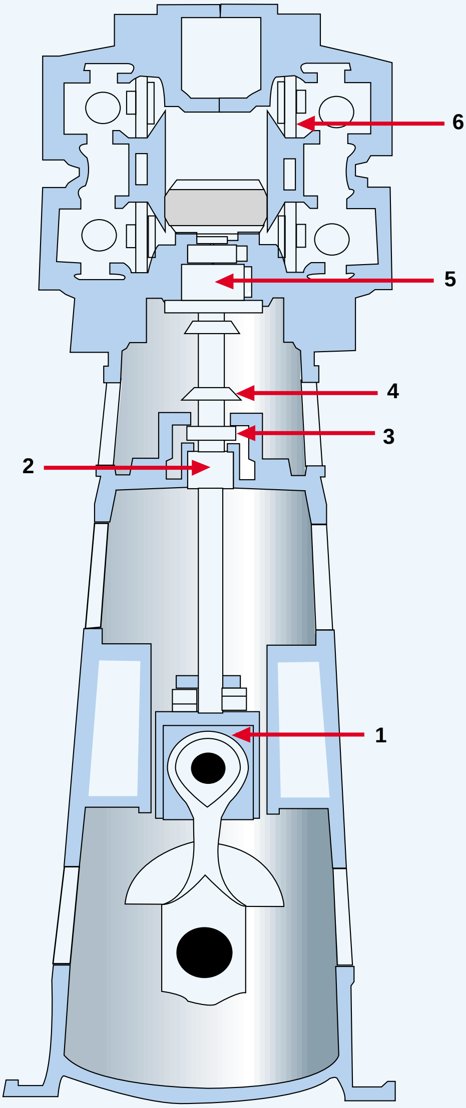


A piston compressor with a valve system consisting of two stainless steel valve discs.

When the piston moves downwards and draws in air into the cylinder the largest disc is sufficiently flexible to fold downwards to allow the air to pass.

When the piston moves upwards, the large disc folds upwards and seals against the seat. The small disc's flexi-function then allows the compressed air to be forced through the hole in the valve seat.

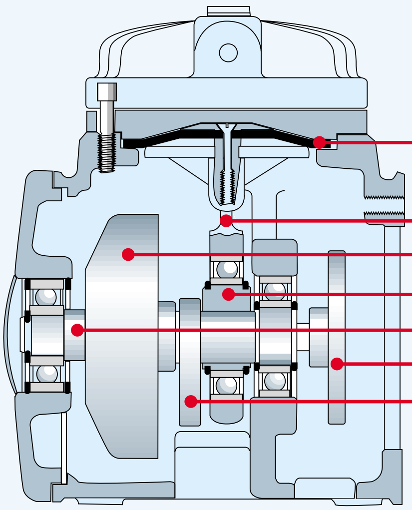
2:4



1. Crosshead
2. Guide bearing
3. Oil scraper
4. Oil thrower ring
5. Stuffing box
6. Valve disc

Labyrinth sealed, double acting oil-free piston compressor with crosshead.

2:5



Mechanical diaphragm compressor, where the movement of the diaphragm is transferred from a conventional crankshaft with connecting rod, which is connected to the diaphragm.

2.1.3 Oil-free piston compressors

Oil-free piston compressors have piston rings of PTFE or carbon, alternatively the piston and cylinder wall can be toothed as on labyrinth compressors. Larger machines are equipped with a crosshead and seals on the gudgeon pins, ventilated intermediate piece to prevent oil from being transferred from the crankcase and into the compression chamber. Smaller compressors often have a crankcase with sealed for life bearings.

2.1.4 Diaphragm compressors

Diaphragm compressors form another group. The diaphragm is actuated mechanically or hydraulically. The mechanical diaphragm compressors are used with a small flow and low pressure or as vacuum pumps. The hydraulic diaphragm compressors are used for high pressure.

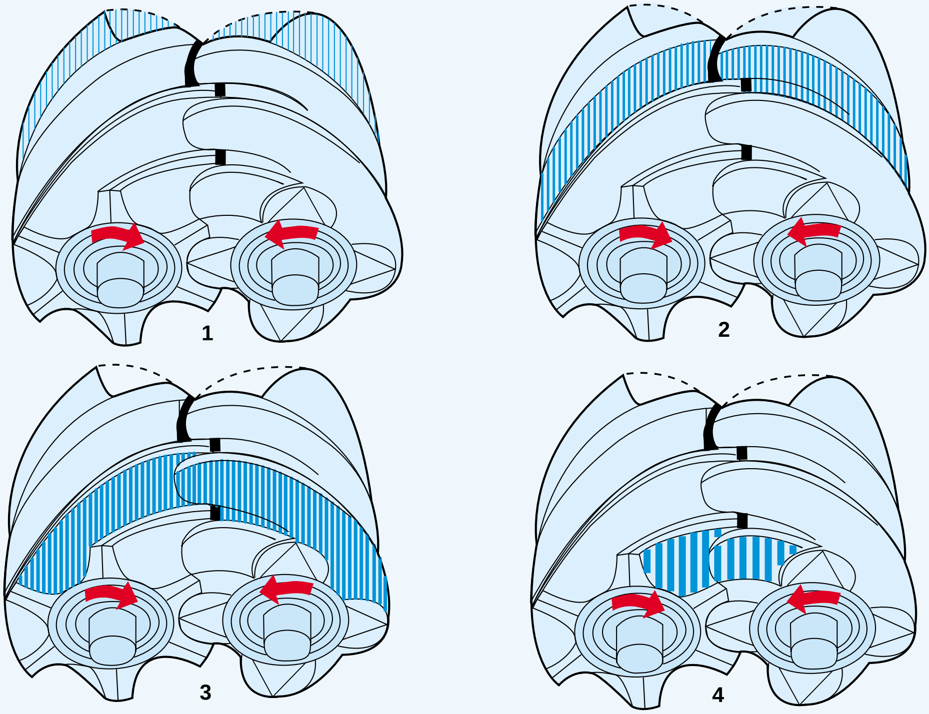
2.1.5 Screw compressors

The principle for a rotating displacement compressor with piston in a screw form was developed during the 1930s, when a rotating compressor with a high capacity and stable flow in varying conditions was required.

The screw element's main parts are the male and female rotors, which move towards each other while the volume between them and the housing decreases. Each screw element has a fixed, integrated pressure ratio that is dependent on its length, the pitch of the screw and the form of the discharge port. To attain the best efficiency the pressure ratio must be adapted to the required working pressure.

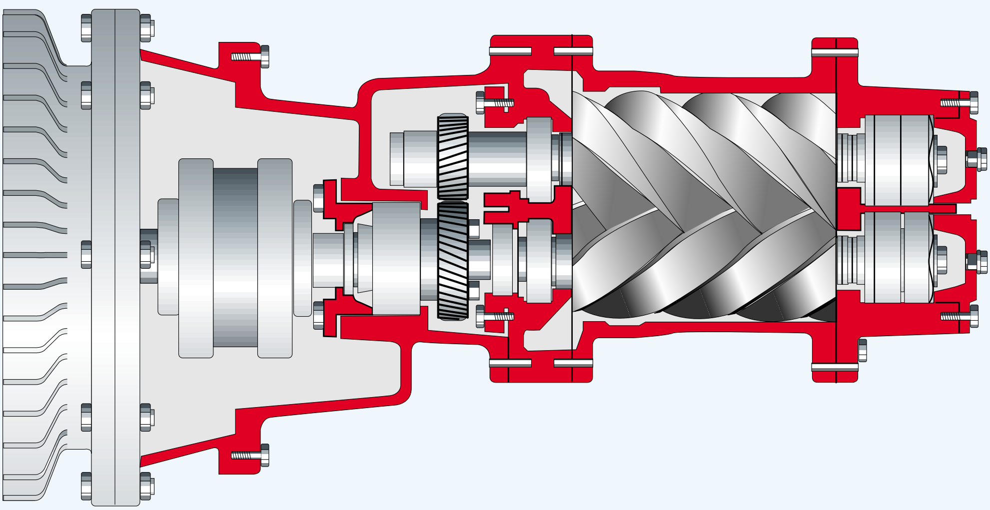
The screw compressor is not equipped with valves and has no mechanical forces that cause unbalance. This means it can work at a high shaft speed and combine a large flow rate with small exterior dimensions. An axial acting force, dependent on the pressure difference between the inlet and outlet, must be taken up by the bearings. The screw, which originally was symmetrical, has now been developed in different asymetrical helical profiles.

2:6

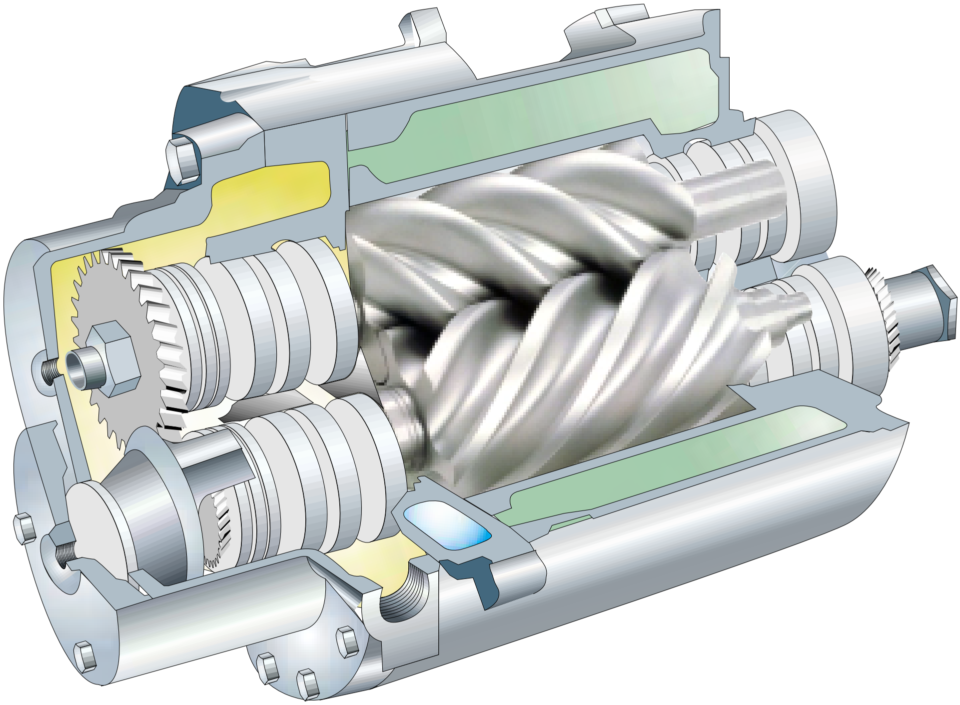


This is how air is compressed in a screw compressor. In figure 1 air fills the space between the rotors, but for each turn the space decreases more and more.

2:7



An oil lubricated screw compressor element.



A stage in an oil-free screw compressor. Male and female rotors are journalled in the rotor housing, which here is water-cooled. The front rotor, with four lobes, is the male, this is connected to the gearbox. The distant rotor, with six lobes, is the female, this is held in place by the synchronising gear to the left.

2.1.5.1 Oil-free screw compressors

The first screw compressors had a symetric profile and did not use liquid in the compression chamber, so-called oil-free or dry screw compressors. At the end of the 1960s a high speed, oil-free screw compressor was introduced with an asymetric screw profile. The new rotor profile resulted in significantly improved efficiency, due to reduced internal leakage.

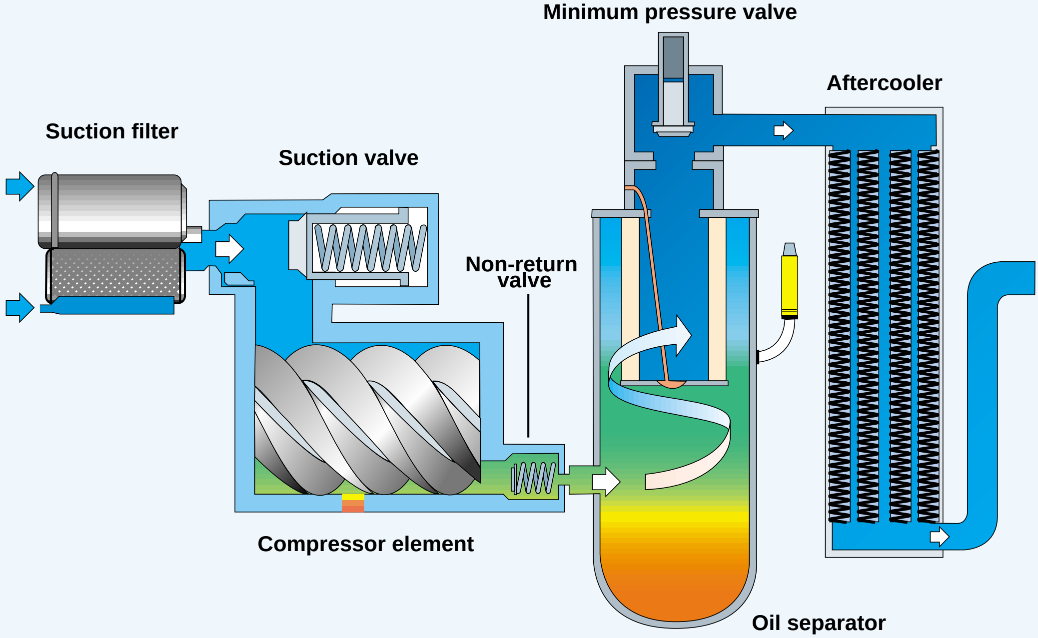
An external gear is used in dry screw compressors to synchronise the counter rotating rotors. As the rotors neither come into contact with each other nor with the compressor housing, no particular lubrication is required in the compression chamber. Consequently the compressed air is completely oil-free. The rotors and housing are manufactured with great precision to minimise leakage from the pressure side to the inlet. The integrated pressure ratio is limited by the temperature difference between the intake and the discharge. This is why oil-free screw compressors are frequently built with several stages.

2.1.5.2 Liquid injected screw compressors

A liquid injected screw compressor is cooled and lubricated by liquid that is injected to the compression chamber and often to the compressor bearings. Its function is to cool and lubricate the compressor element and to reduce the return leakage to the intake.

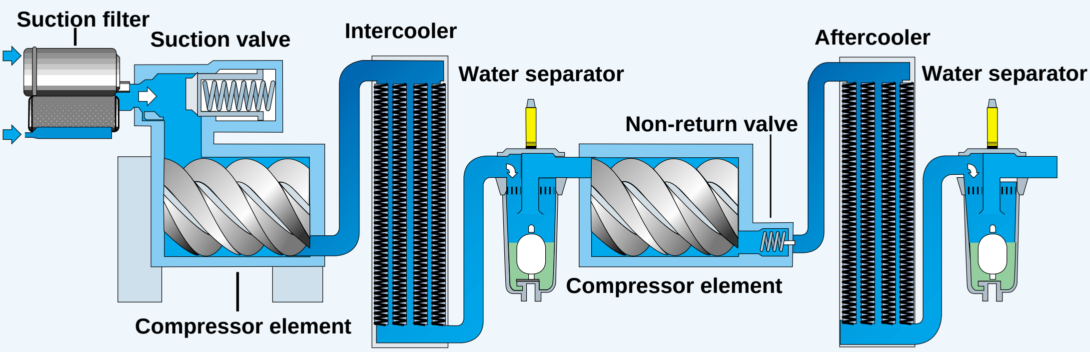
Today oil is the most common liquid due to its good lubricating properties, however, other liquids are also used, for example, water. Liquid injected screw compressor elements can be manufactured for high pressure ratio, which why one compression stage is usually sufficient for pressure up to 13 bar. The element's low return leakage also means that relatively small screw compressors are efficient.

2:9



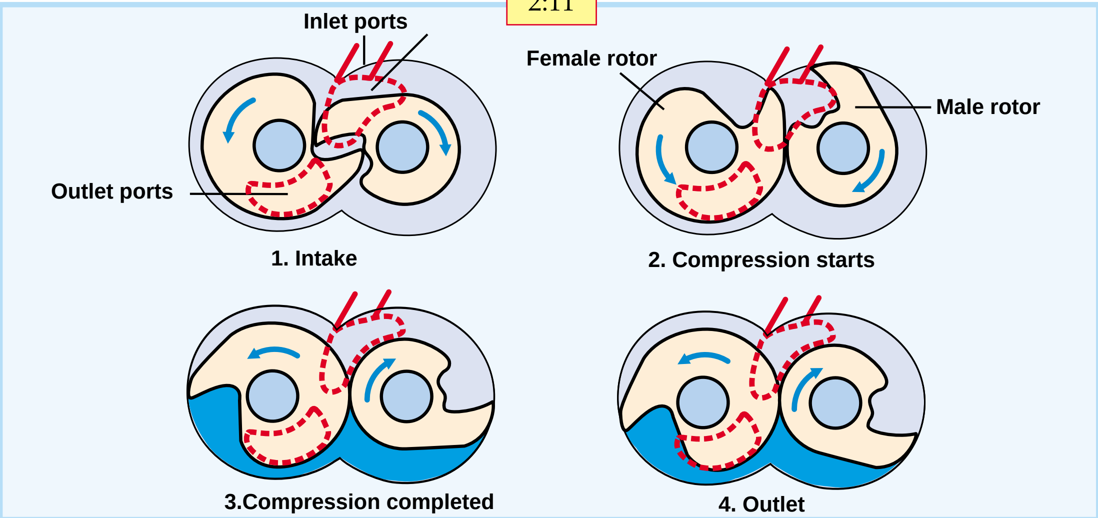
Oil injected screw compressor.

2:10



Oilfree screw compressor.

2:11



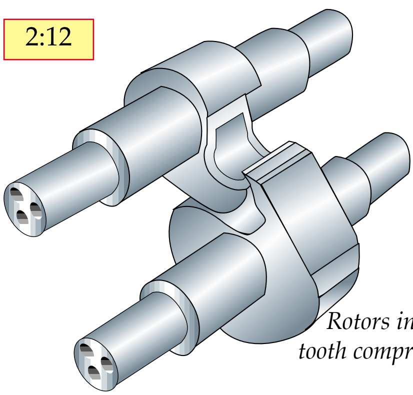
2.1.6 Tooth compressor

The compression element in a tooth compressor consists of two rotors that rotate towards each other in a compression chamber.

The compression process consists of intake, compression and outlet. During the intake phase air is drawn into the compression chamber until the rotors block the inlet. During the compression phase the drawn in air is in the compression chamber, which gets smaller and smaller as the rotors move.

The outlet is blocked during compression by one of the rotors, while the inlet is open to draw in new air into the opposite section of the compression chamber.

2:12



Rotors in a tooth compressor

Discharge takes place when one of the rotors opens the channel and the compressed air is forced out of the compression chamber. Intake and outlet take place radially through the compression chamber, which allows the use of simpler bearing design and improve filling properties.

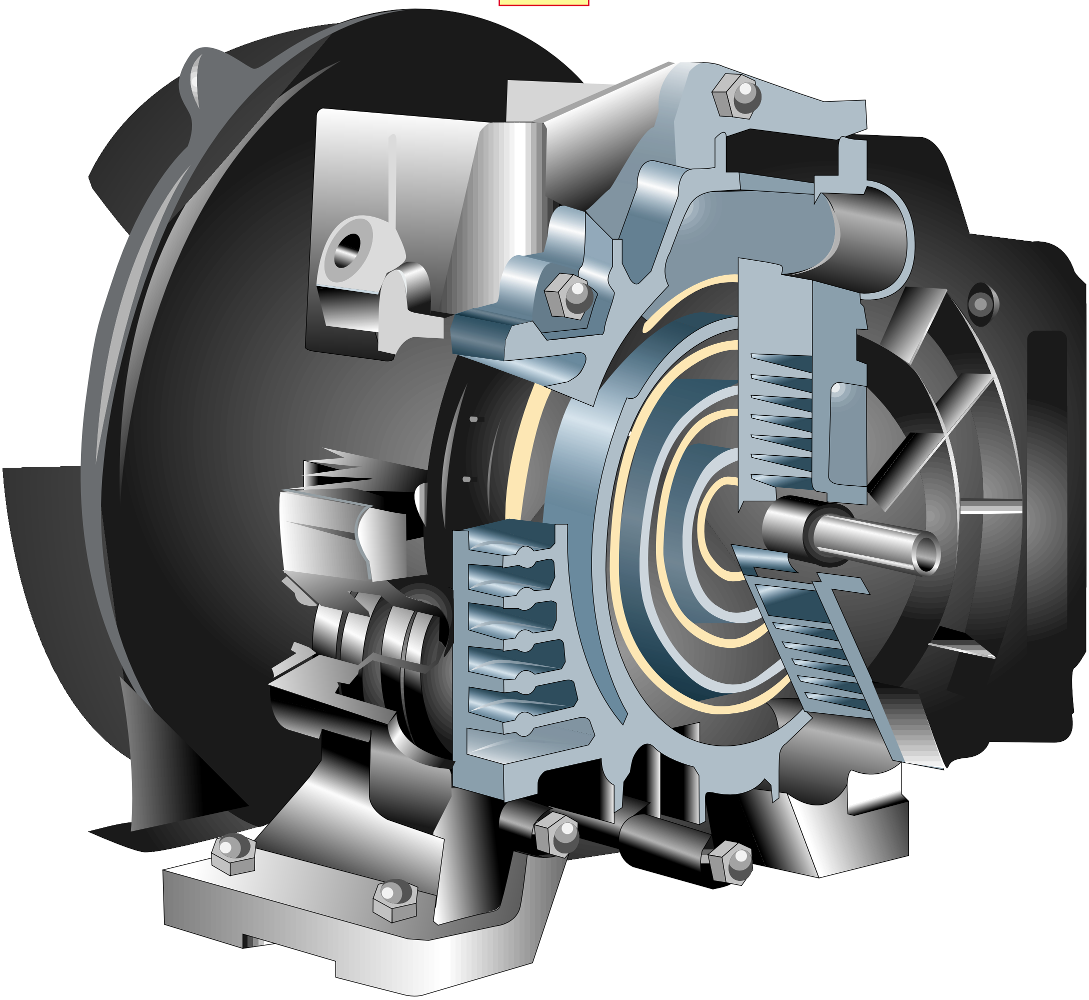
Both rotors are synchronised via a gear wheel. The maximum pressure ratio obtainable with an oil-free tooth compressor is 4.5. Consequently several stages are required for higher pressures.

2.1.7 Scroll compressor

A scroll compressor is a type of oil-free rotating displacement compressor, i.e. it compresses a specific amount of air in an ever decreasing volume. The compressor element consists of a fixed spiral in an element housing and a motor powered eccentric, moveable spiral. The spirals are mounted with 180° phase displacement to form air pockets with a varying volume.

This provides the elements with radial stability. Leakage is minimised as the pressure difference in the air pockets is less that the pressure difference between the inlet and the outlet.

2:13

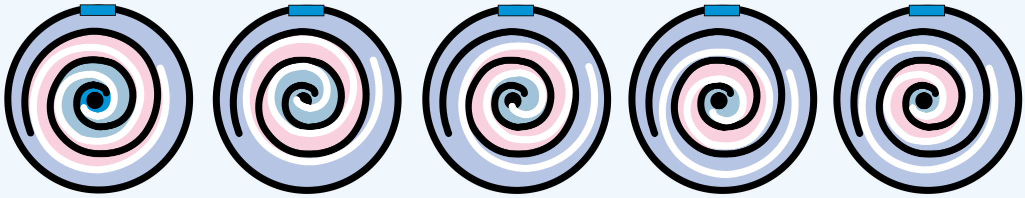


A scroll compressor in cross section.

The moving spiral is driven by a short stroke crankshaft and runs eccentrically around the centre of the fixed spiral. The intake is situated at the top of the element housing.

When the moving spiral runs anticlockwise air is drawn in, and is captured in one of the air pockets and compressed variably in towards the centre where the outlet and a non-return valve are situated. The compression cycle is in progress for 2.5 turns, which virtually gives constant and pulsating free air flow. The process is relatively silent and vibration free, as the element has hardly any torque variation compared to, e.g. a piston compressor.

2:14

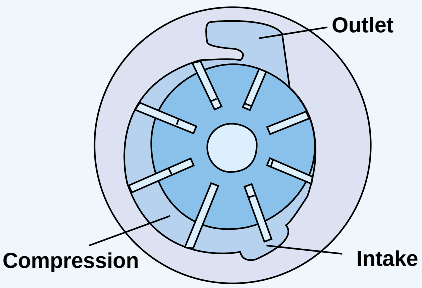


Operating principle for a scroll compressor

2.1.8 Vane compressor

The operating principle for a vane compressor is the same as for many compressed air motors. The vanes are usually manufactured of special cast alloys and most compressors are oil lubricated.

2:15



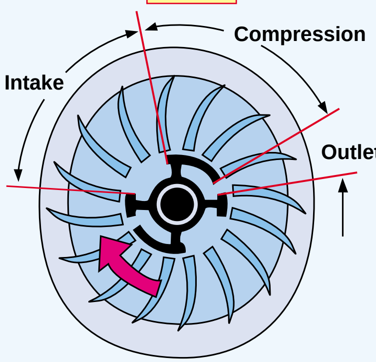
A rotor with radially movable blades is eccentrically mounted in a stator housing. When it rotates the vanes are pressed against the stator walls by centrifugal force. Air is drawn in when the distance between the rotor and stator is increasing. The air is captured in different compressor pockets, which decrease in volume with rotation. The air is discharged when the vanes pass the outlet port.

2.1.9 Liquid-ring compressor

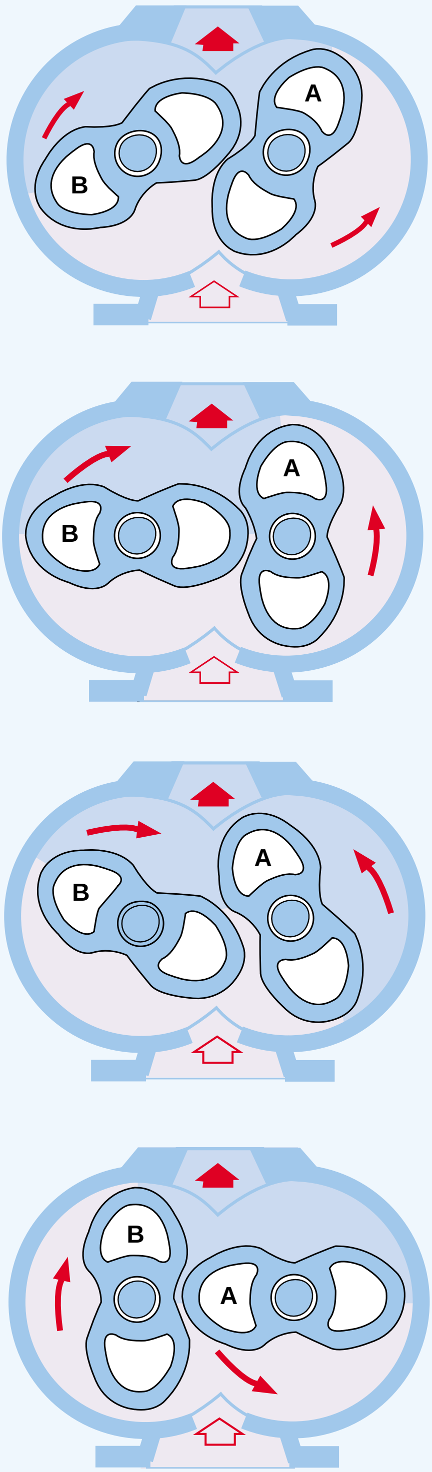
The liquid ring compressor is a displacement compressor with built-in pressure ratio. The rotor has fixed blades and is eccentrically mounted in a housing, which is partly filled with a liquid. The blade wheel transports the liquid around in the compressor housing and a ring of liquid is formed around the compressor housing wall by means of centrifugal force. The liquid ring lies eccentrically to the rotor as the compressor housing has an oval form. The volume between the blade wheel varies cyclically. The compressor is usually designed with two symmetrical, opposite compression chambers to avoid radial thrust on the bearings.

Cooling in a liquid ring compressor is direct, due to the contact between the liquid and the air, and means the temperature increase on the compressed air is very little. However, losses through viscous friction between the housing and the blades are high. The air becomes saturated with compressor liquid, which normally is water. Other liquids can also be used, for example, to absorb a specific constituent part of the gas to be compressed or to protect the compressor against corrosion when aggressive gases are compressed.

2:16



2:17



Operating principle of a blower

2.1.10 Blowers

A blower is not a displacement compressor as it works without internal compression. When the compression chamber comes into contact with the outlet, compressed air floods in from the pressure side. It is first here that compression takes place, when the volume of the compression chamber decreases with continued rotation. Accordingly, compression takes place against full counter-pressure, which results in low efficiency and a high noise level.

Two identical, normally symmetrical, counter-rotating rotors work in a housing with flat ends and a cylindrical casing. The rotors are synchronised by means of a gear wheel. Blowers are usually air cooled and oil-free. The low efficiency limits the blowers to low pressure applications and compression in a single stage, even if two and three stage versions are available. Blowers are frequently used as vacuum pumps and for pneumatic conveyance.

2.2 Dynamic compressors

2.2.1 Dynamic compressors in general

Dynamic compressors are available in axial and radial designs. The latter are frequently called turbo or radial turbo and the former are called centrifugal compressors. A dynamic compressor works with a constant pressure, unlike for example a displacement compressor, which works with a constant flow. The performance of a dynamic compressor is affected by external conditions, for example, a small change in the inlet pressure results in a large change in the capacity.

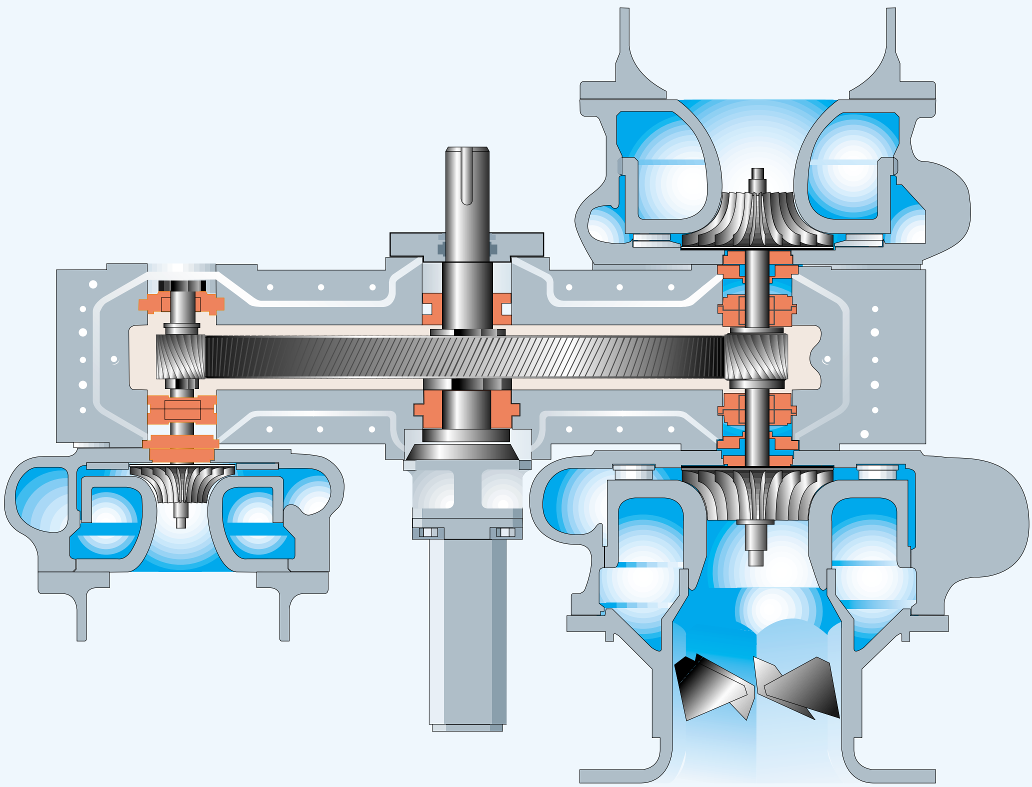
2.2.2 Centrifugal compressors

The centrifugal compressor is characterised by the radial discharge flow. Air is drawn into the centre of a rotating impeller with radial blades and is thrown out towards the periphery of the impeller by centrifugal forces. Before the air is led to the centre of the next impeller, it passes a diffuser and a volute where the kinetic energy is converted to pressure.

The pressure ratio across each stage is determined by the compressor's final pressure. This also gives a suitable velocity increase for the air after each impeller. The air temperature at the inlet of each stage has a decisive significance for the compressor's power requirement, which is why cooling between stages is needed. Centrifugal compressors with up to six stages and pressure up to 25 bar are not uncommon. The impeller can have either an open or closed design. Open is the most common with air applications. The impeller is normally made of special stainless steel alloy or aluminium. The speed is very high compared with other types of compressor, 15,000-100,000 r/min are common.

This means that journalling on the compressor shaft takes place using plain bearings instead of rolling bearings. Rolling bearings are used on single stage compressors with a low pressure ratio.

2:18



Three stage centrifugal compressor

Often multi-stage compressors have two impellers mounted on each end of the same shaft to counteract the axial loads caused by the pressure differences. The lowest volume flow rate through a centrifugal compressor is primarily determined by the flow through the last stage. A practical limit value of 160 l/s in the outlet from a horizontal split machine is often a rule-of-thumb.

Each centrifugal compressor must be sealed in a suitable manner to reduce leakage along the shaft where it passes through the compressor housing. Many types of seal are used and the most advanced can be found on compressors with a high speed intended for high pressures. The four most common types are labyrinth seals, ring seals, (usually graphic seals that work dry, but even sealing liquids are used), mechanical seals and hydrostatic seals.

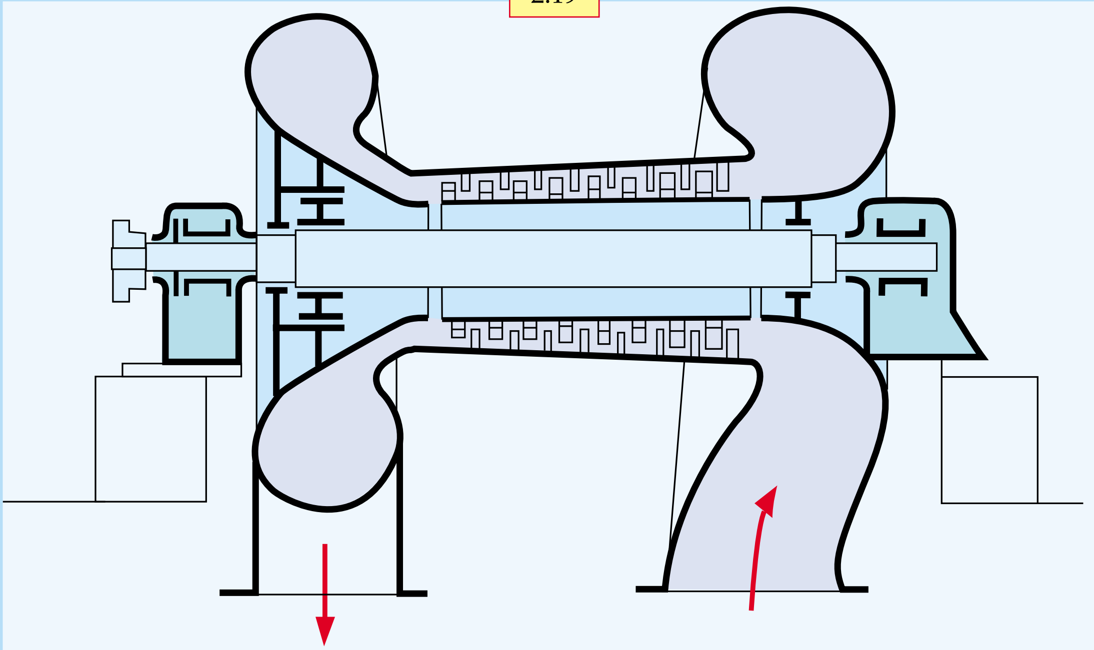
2.2.3 Axial compressors

A axial compressor has axial flow, the air or gas passes along the compressor shaft through rows of rotating and stationary impellers. In this way the velocity of the air is gradually increased at the same time as the stationary blades convert the kinetic energy to pressure.

The lowest volume flow rate through such a compressor is about 15 m^3/s. A balancing drum is usually built into the compressor to counterbalance axial thrust.

Axial compressors are generally smaller than equivalent centrifugal compressors and work ordinarily with about a 25% higher speed. They are used for constant high volume rate of flow at a relatively moderate pressure. With the exception of gas turbine applications the pressure ratio is seldom higher than 6. The normal flow is approx. 65 m^3/s and effective pressure up to approx. 14 bar(e).

2:19



Axial compressor

2.3 Other compressors

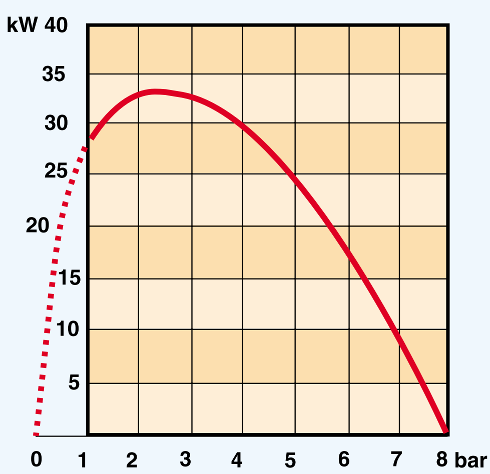
2.3.1 Vacuum pumps

A vacuum means a lower pressure than atmospheric pressure. A vacuum pump is a compressor that works in this pressure range. A typical characteristic of a vacuum pump is that they work with a very high pressure ratio, however despite this, multi-stage machines are common. Multi-stage compressed air compressors can also be used for vacuums within the pressure range 1 bar(a) and 0.1 bar(a).

2.3.2 Booster compressors

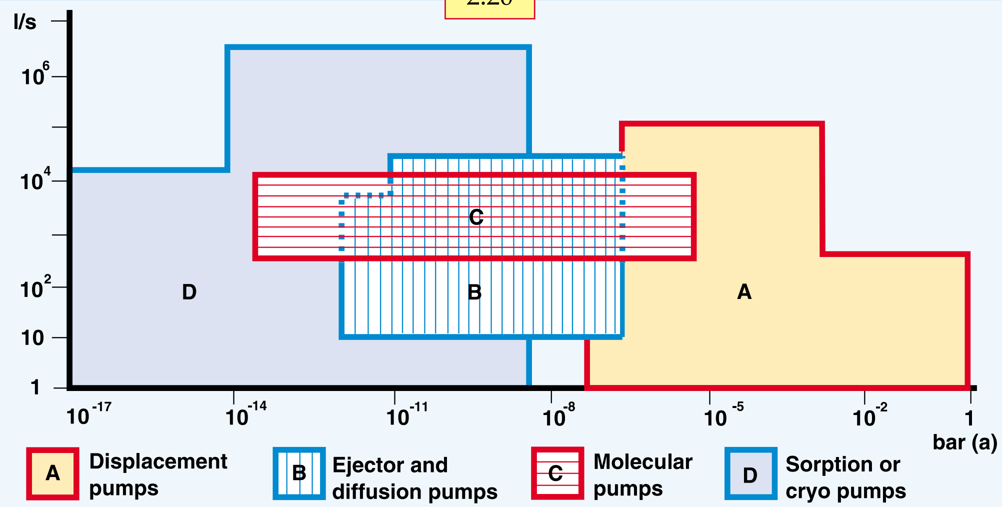
A booster compressor is a compressor that works with air that has been compressed and compresses it to a higher pressure. It is used to compensate the pressure drop in long pipelines or in applications where a higher pressure is required for a sub-process. Compression may be single or multi-staged and the compressor can be of a dynamic or displacement type, but piston compressors are the most common. The power requirement for a booster compressor increases with a rising pressure ratio, while the mass flow drops. The curve for power requirement as a function of the intake pressure has the same general form as the curve for a vacuum pump.

2:21



Adiabatic power requirement for a booster compressor with an absolute final pressure of 8 bar.

2:20

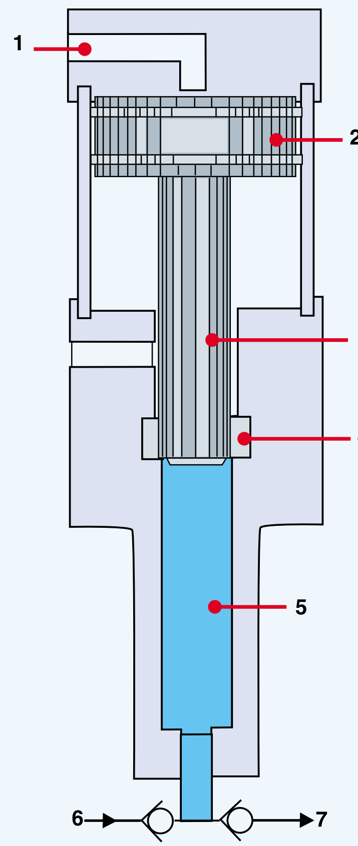


The working range for some types of vacuum pumps

2.3.3 Pressure intensifiers

Pressure intensifiers increase the pressure in a medium, for example, for laboratory tests of valve, pipes and hoses. A pressure of 7 bar can be amplified in a single stage to 200 bar or up to 1700 bar in multi-staged equipment. The pressure intensifier is only available for very small flows.

2:22



1. Propellant
2. Low pressure piston
3. High pressure piston
4. Seal
5. Compression chamber
6. Intake
7. Outlet

A cross section of a single stage pressure intensifier.

When the high pressure chamber is filled, the low pressure piston is lifted. When the propellant flows in, the piston is pressed downwards and forces the medium out under high pressure. The intensifier can work in a cycling process, up until a preset pressure level. All inert gases can be compressed in this way. Air can also be compressed in a pressure intensifier, but must be completely oil-free to avoid selfignition.

2.4 Treatment of compressed air

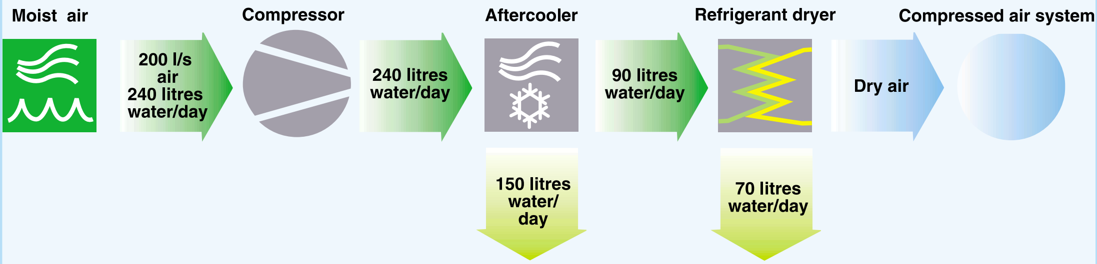
2.4.1 Drying compressed air

All atmospheric air contains water vapour, more at high temperatures and less at lower temperatures. When the air is compressed the water concentration increases. For example, a compressor with a working pressure of 7 bar and a capacity of 200 l/s that draws in air at 20˚C with a relative humidity of 80% will give off 80 litres of water in the compressed air line during an eight hour working day.

The term pressure dew point (PDP) is used to describe the water content in the compressed air. It is the temperature at which water vapour transforms into water at the current working pressure. Low PDP values indicate small amounts of water vapour in the compressed air.

It is important to remember that atmospheric dew point can not be compared with PDP when comparing different dryers. For example, a PDP of +2˚C at 7 bar is equivalent to –23˚C at atmospheric pressure. To use a filter to remove moisture (lower the dew point) does not work. The reason is because further cooling means

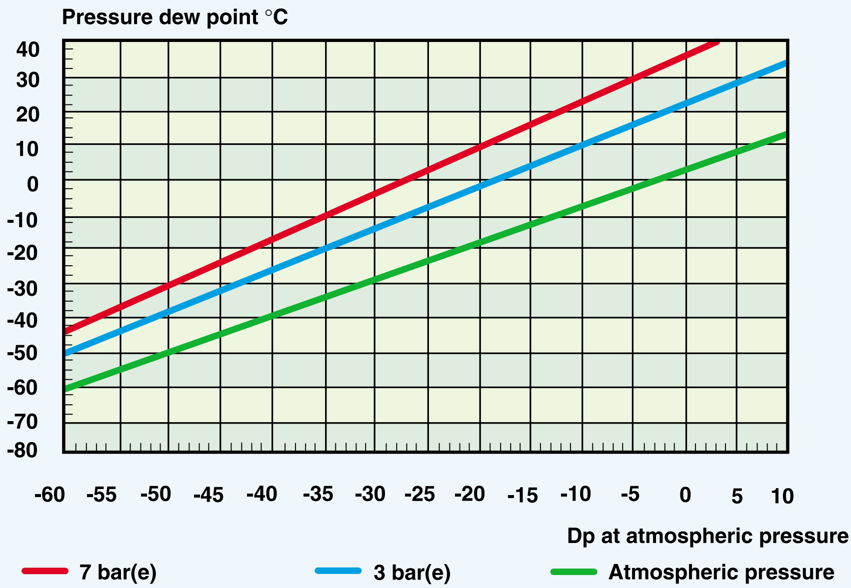
2:23



A compressor that delivers 200 litres of air per second, also supplies approx. 240 litres of water per day if working with air at 20˚C. To avoid problems and disturbances due to water precipitation in the pipes and connected equipment the compressed air must be dried. This takes place in an aftercooler and drying equipment as set out in the figure.

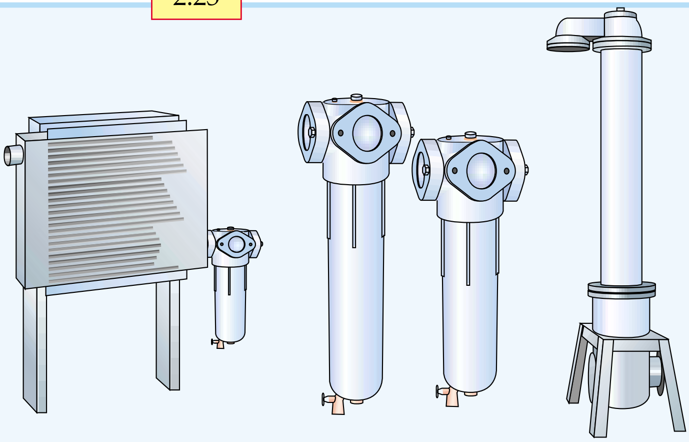
continued precipitation of condensation water. You can select the main type of drying equipment based on the pressure dew point. Seen from a cost point of view, the lower the dew point required the higher the acquisition and operating costs for air drying. In principle, there are four methods to remove the moisture from compressed air: Cooling, over-compression absorption and adsorption. There is equipment available, based on these methods for different types of compressed air systems.

2:24



This is how the relation between dew point and compressed air looks.

2:25



Different aftercoolers and water separators. The water separator can, for example, work with cyclone separating or separation through changes indirection and speed.

2.4.1.1 Aftercooler

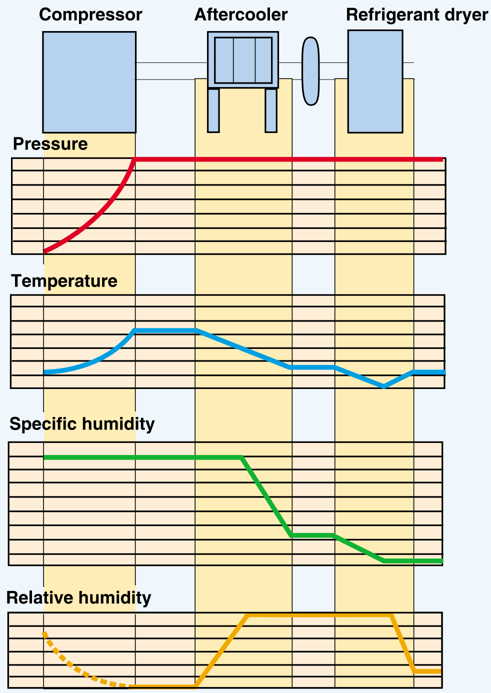
An aftercooler is a heat exchanger, which cools the hot compressed air to precipitate the water that otherwise would condensate in the pipe system. It is water or air cooled, generally equipped with a water separator with automatic drainage and should be placed next to the compressor.

80–90% of the precipitated condensation water is collected in the aftercooler's water separator. A common value for the temperature of the compressed air after the aftercooler is approx. 10°C above the coolant temperature, but can vary depending on the type of cooler. An aftercooler is used in virtually all stationary installations. In most cases an aftercooler is built into modern compressors.

2.4.1.2 Refrigerant dryer

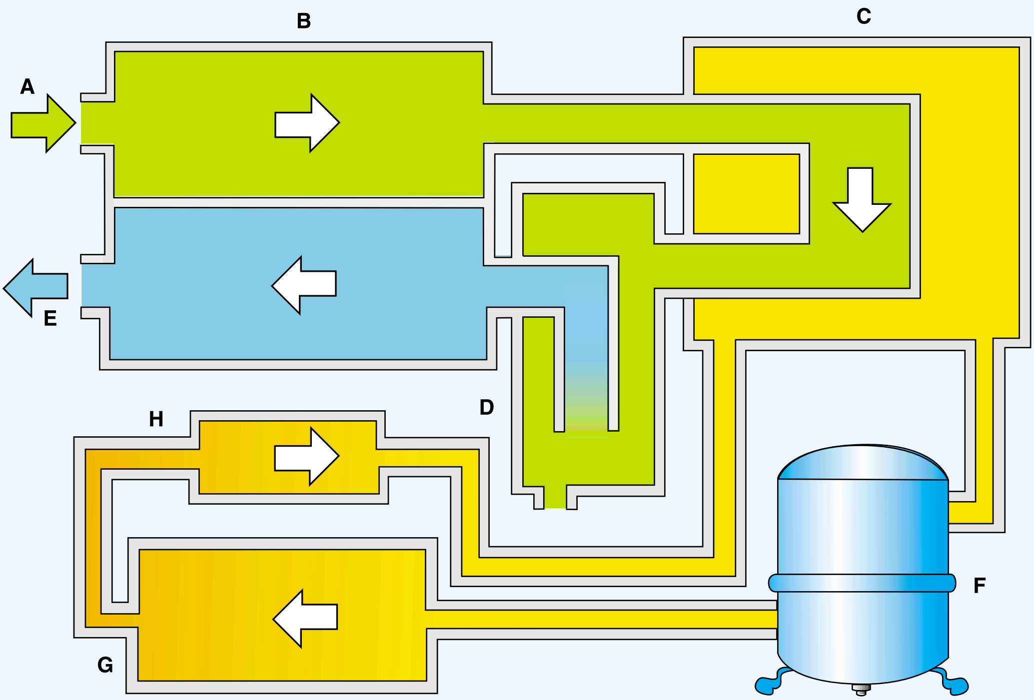
Refrigerant drying means that the compressed air is cooled, whereby a large amount of the water condenses and can be separated. After cooling and condensing the compressed air is reheated to around room temperature so that condensation does not form on the outside of the pipe system. Cooling of the compressed air takes place via a closed coolant system. By cooling the incoming compressed air with the cooled air in the heat exchanger the energy consumption of the refrigerant dryer is reduced. Refrigerant dryers are used with dew points between +2°C to +10°C and are limited downwards by the freezing point of the condensed water.

2:26



Examples of how different parameters change with compression, aftercooling and refrigerant drying.

2:27



- **A** Incoming compressed air
- **B** Air/air heat exchanger
- **C** Air/coolant heat exchanger
- **D** Water separator
- **E** Dry compressed air
- **F** Compressor
- **G** Condenser
- **H** Expansion valve

This is how refrigerant cooling works in principle.

2.4.1.3 Over-compression

Over-compression is perhaps the easiest method to dry compressed air.

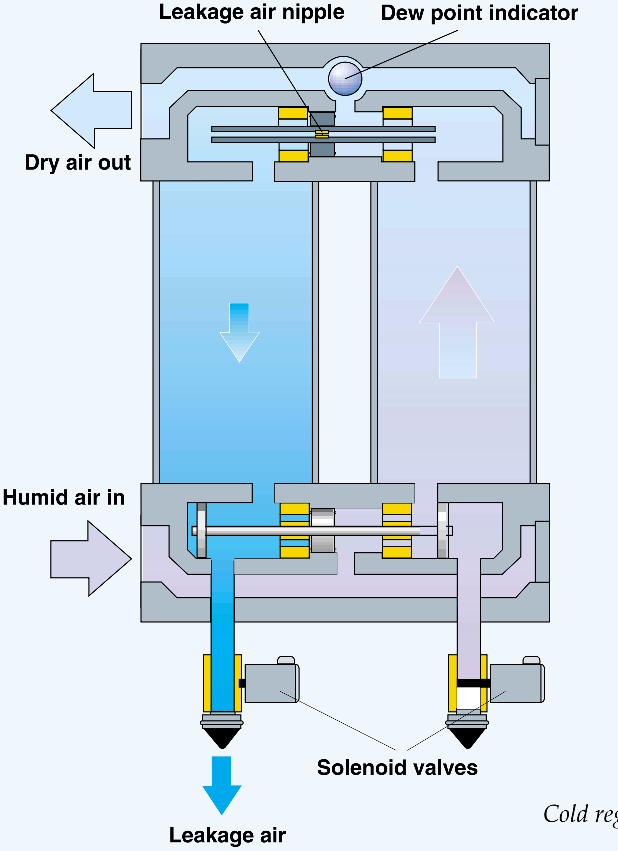
Air is first compressed to a higher pressure than the intended working pressure, which means the concentration of water vapour increases. Thereafter the air is cooled, whereby the water is separated. Finally the air is allowed to expand to the working pressure, whereby a lower PDP is attained. However, this method is only suitable for very small air flow rates, due to the high energy consumption.

2.4.1.4 Absorption drying

Absorption drying is a chemical process, where water vapour is bound to the absorption material. The absorption material can either be a solid or liquid. Sodium chloride and sulphuric acid are frequently used, which means the possibility of corrosion must be taken into consideration.

This method is unusual and has a high consumption of absorption material. The dew point is only lowered to a limited degree.

2:28



Cold regenerative adsorption drying

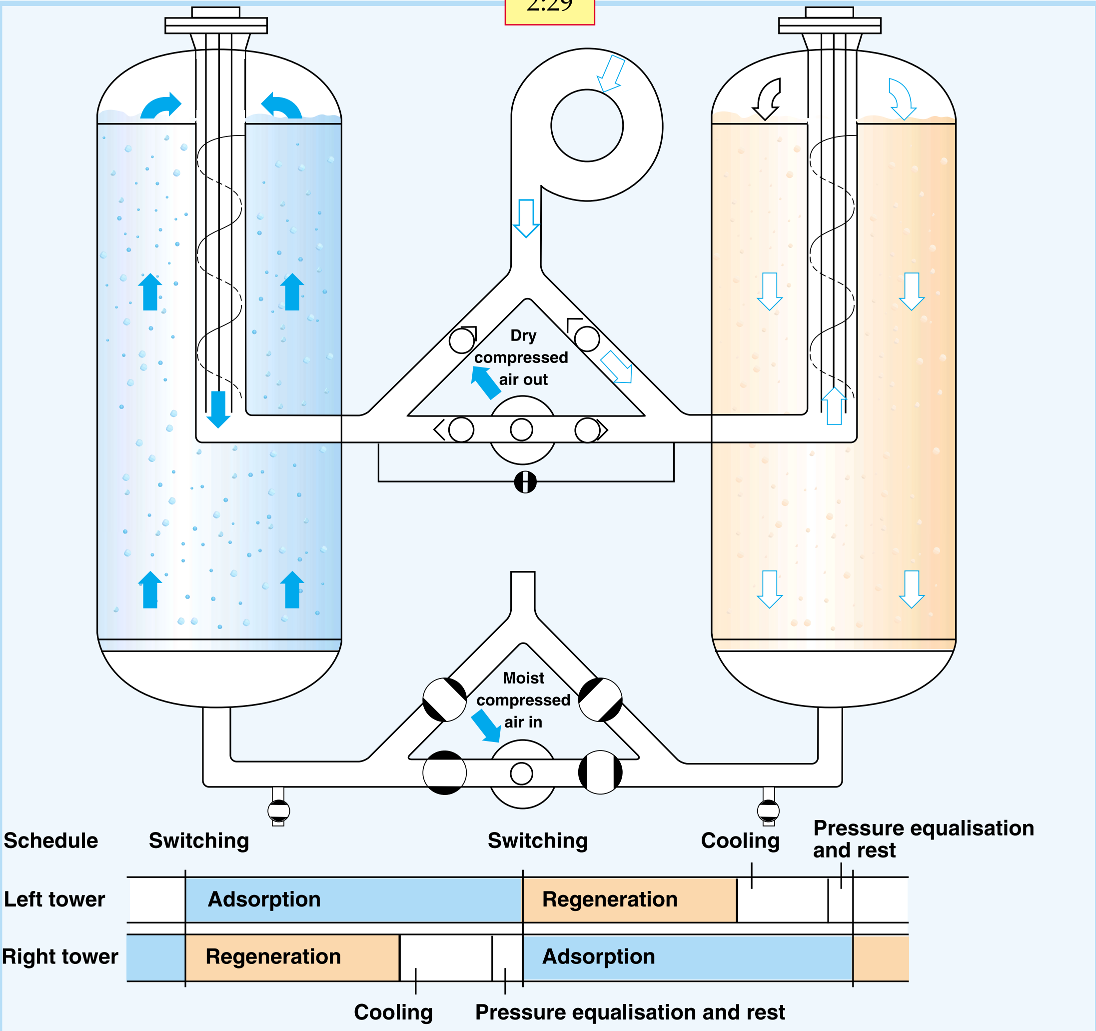
2.4.1.5 Adsorption drying

There are two types of adsorption dryer, cold regenerative and hot regenerative. Cold regenerative dryers are best suited to smaller air flow rates. The regeneration process takes place with the help of compressed air and requires approx. 15–20% of the dryer's nominal capacity at 7 bar(e) working pressure, PDP 20°C. Lower PDP requires a greater leakage air flow. Hot regenerative adsorption drying regenerates the desiccant by means of electrical or compressor heat, which is more economical than cold regeneration. Very low dew points (-30°C or lower) can be obtained.

Guaranteed separation and drainage of the condensation water shall always be arranged before adsorption drying. If the compressed air has been produced using oil lubricated compressors, an oil separating filter should also be fitted before the drying equipment. In most cases a particle filter is required after adsorption drying.

There are adsorption dryers for oil-free screw compressors that use the heat from the compressor to regenerate the desiccant. These types of dryers are generally fitted with a rotating drum with desiccant of which one sector (a quarter) is regenerated by means of a partial flow of hot compressed air (130–200°C) from the compressor stage. Regenerated air is then cooled, the condensation drained and the air returned via the ejector to the main air flow. The rest of the drying drum's surface (three-quarters) is used to dry the com-

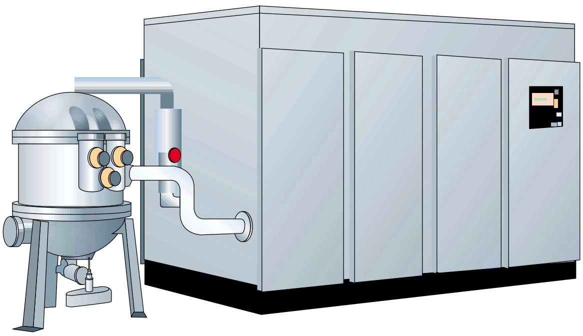
2:29



In the diagram the left tower dries the compressed air while the right tower regenerates. After cooling and pressure equalisation the towers are automatically switched.

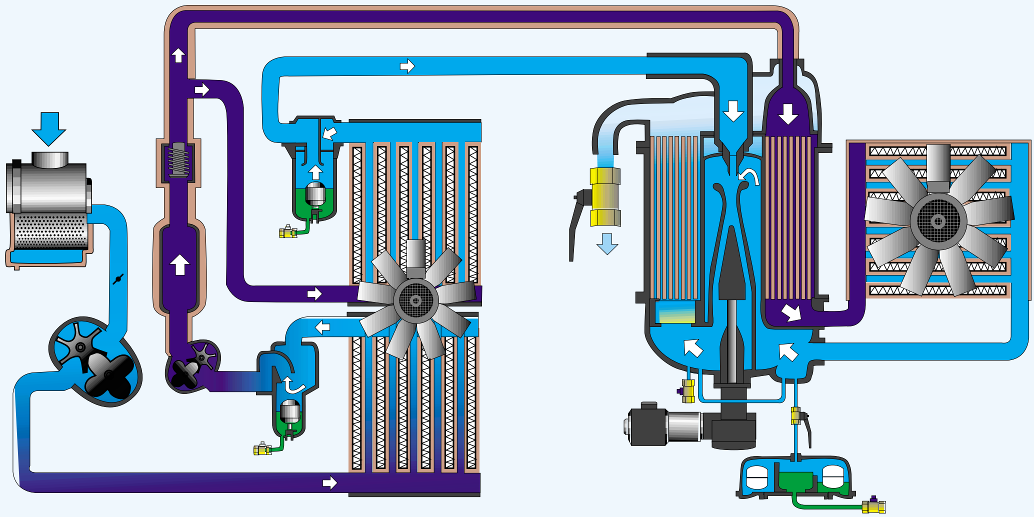
pressed air from the compressor's aftercooler. The system gives no compressed air losses. The power requirement for such a dryer is limited to that required for powering the drum. For example, a dryer with a capacity of 1000 l/s only requires 120 W. In addition, no compressed air is lost and neither oil nor particle filters are required.

2:30



Oil-free screw compressor with a MD type adsorption dryer.

2:31



MD dryer

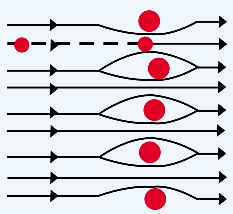
2.4.2 Filters

Particles in an air stream that pass a filter can be removed in several different ways. If the particles are larger than the opening in the filter material they are separated mechanically.

This usually applies for particles greater than 1 µm. The filter's efficiency in this regard increases with a tighter filter material, consisting of finer fibres. Particles between 0.1 μm and 1 μm can be separated by the air stream going around the filter material's fibres, while the particles through their inertia continue straight on. These then hit the filter material's fibres and adhere to the surface. The efficiency of the filter in this regard increases with an increased flow velocity and a tighter filter material consisting of finer fibres.

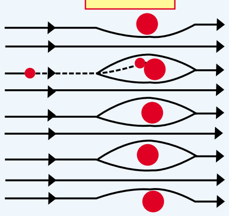
Very small particles (<0.1 μm) move randomly in the air stream influenced by collisions with air molecules. They "hover" in the air flow changing direction the whole time, which is why they easily collide with the filter material's fibres and adhere there. The efficiency of the filter in this regard increases with a reduction in the stream velocity and a tighter filter material consisting of finer fibres.

2:32



This is how filter material with mechanical separation works in theory. Particles that are > 1µm are separated.

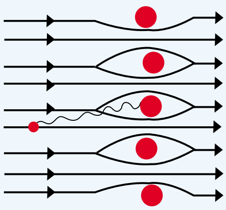
2:33



Particles between 0.1–1 µm move randomly in the air stream and are separated when they collide with the fibres in the filter material.

The separating capacity of a filter is a result of the different sub-capacities as set out above. In reality, each filter is a compromise, as no filter is efficient across the entire particle scale, even the effect of the stream velocity on the separating capacity for different particle sizes is not a decisive factor. For this reason particles between 0.2 μm and 0.4 μm are the most difficult to separate.

2:34

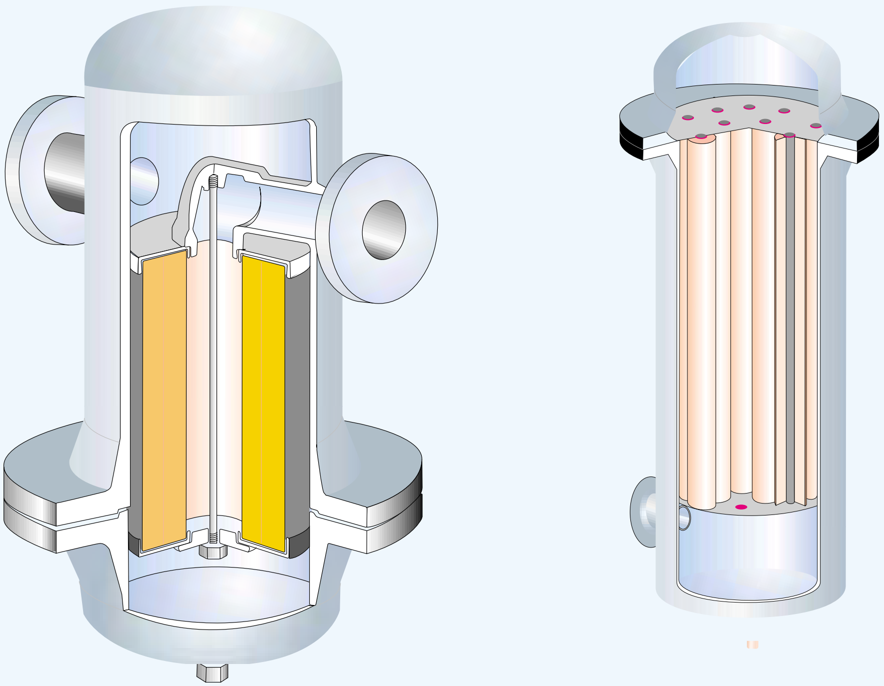


Particles (<0.1 μm) that collide with fibres in the filter are separated by adhering to the surface.

The separating efficiency for a filter is specified for a specific particle size. A separation efficiency of 90–95% is frequently stated, which means that 5–10% of all particles in the air go straight through the filter. Furthermore, a filter with a stated 95% separation efficiency for the particle size 10 μm can let through particles that are 30–100 μm in size. Oil and water in aerosol form behave as other particles and can also be separated using a filter.

Drops that form on the filter material's fibres sink to the bottom of the filter due to gravitational forces. The filter can only separate oil in aerosol form. If oil in vapour form is to be separated the filter must contain a suitable adsorption material, usually active carbon.

2:35



This is how a particle filter can look in reality. A large filter housing and large area mean a low air velocity, less pressure drop and a longer service life.

A filter to remove oil, water and dust particles. The filter element has a small diameter and consists of spun glass fibre

All filtering results in a pressure drop, that is to say, an energy loss in the compressed air system. Finer filters with a tighter structure cause a greater pressure drop and become blocked more quickly, which demands more frequent filter replacement resulting in higher costs.

Accordingly, filters must be dimensioned so that they not only handle the nominal flow, but also have a greater capacity threshold so they can manage a pressure drop due to a degree of blockage.

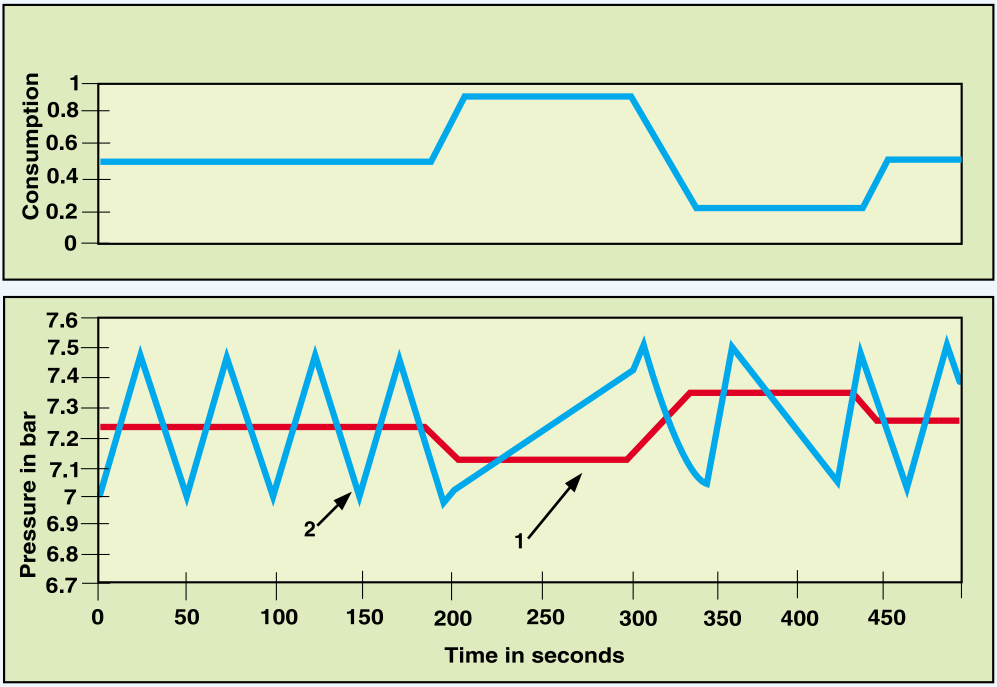
2.5 Control and regulation systems

2.5.1 Regulation, general

Frequently you require a constant pressure in the compressed air system, which makes demands on the ability to be able to control the compressed air flow from the compressor centre. There are a number of methods for this depending on, e.g. the type of compressor, permitted pressure variations, consumption variations and acceptable losses.

Energy consumption represents approx. 80% of the total cost for compressed air, which means that you should carefully consider the choice of regulation system. Primarily this is because the differences in performance broadly overshadow the differences between compressor types and manufacturers. It is ideal when the compressor's full capacity can be exactly adapted to an equal consumption, for example, through carefully choosing the gearbox's transmission ratio, something that is frequently used in process applications. A number of consumers are self-regulating, i.e. increased pressure gives an increased

2:36



1. Continuous capacity regulation 2. Load/unload regulation

flow rate, which is why they form stable systems. Examples can be pneumatic conveyors, ice prevention, chilling, etc. However, normally the flow rate must be controlled, which often takes place using equipment integrated in the compressor. There are two main groups of such regulation systems:

1. Continuous capacity regulation involves the continuous control of the drive motor or valve according to variations in pressure. The result is normally small pressure variations (0.1 to 0.5 bar), depending on the regulation system's amplification and its speed.

2. Load/unload regulation is the most common regulation system and involves the acceptance of variations in pressure between two values. This takes place by completely stopping the flow at the higher pressure (off-loading) and resume the flow rate (loading) when the pressure has dropped to the lowest value. Pressure variations depend on the permitted number of load/unload cycles per time unit, but normally lie within the range 0.3 to 1 bar.

2.5.2 Regulation principles for displacement compressors

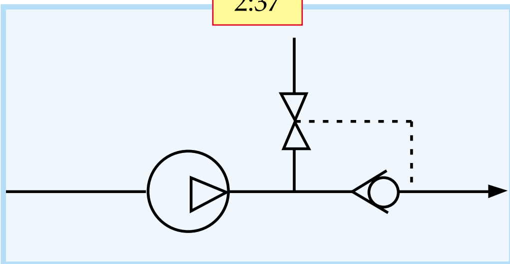
2.5.2.1 Pressure relief

The original method to regulate a compressor is a pressure relief valve, which releases excess pressure into the atmosphere. The valve in its simplest design can be spring loaded, where the spring tension determines the final pressure.

Frequently a servo-valve is used instead, which is controlled by a regulator. The pressure can then be easily controlled and the valve can also act as an off-loading valve when starting a compressor under pressure. Pressure relief makes large energy demands, as the compressor must work continuously against full counter pressure.

A variant, which is used on smaller compressors, is to unload the compressor by fully opening the valve so that the compressor works against atmospheric pressure. Power consumption is significantly more favourable using this method.

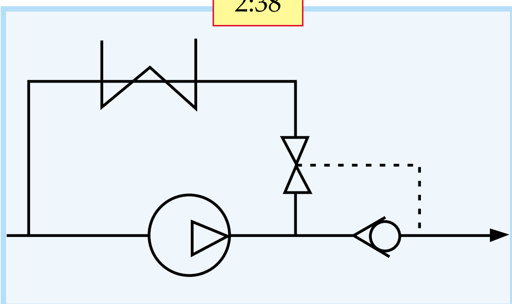
2:37



2.5.2.2 Bypass

Bypass regulation has in principle the same function as pressure relief. The difference is that the pressure relieved air is cooled and returned to the compressor's intake. The method is often used on process compressors where the gas is unsuitable or too valuable to release into the atmosphere.

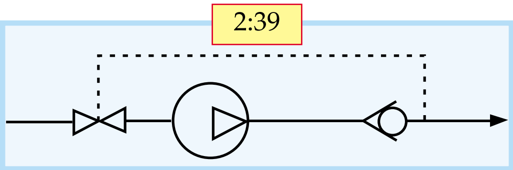
2:38



2.5.2.3 Throttling the intake

Throttling is an easy method to reduce the flow. By increasing the pressure ratio across the compressor, depending on the induced underpressure in the intake, the method is however limited to a small regulation range. Liquid injected compressors, which have a large permitted pressure ratio, can however be regulated down to 10% of the maximum capacity. This method makes relatively high energy demands, due to the high pressure ratio.

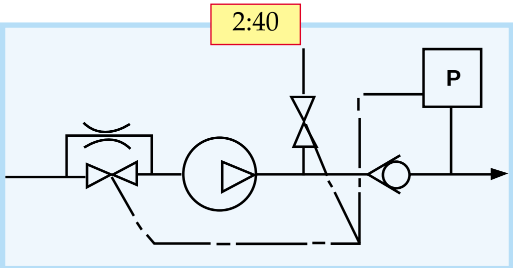
2:39



2.5.2.4 Pressure relief with throttled intake

The most common regulation method currently used that unites a maximum regulation range (0-100%) with low energy consumption, only 15–20% of full load power with an off-loaded compressor (zero flow). The intake valve is closed, but with a small opening remaining, at the same time as a blow off valve opens and relieves the outgoing air from the compressor.

2:40

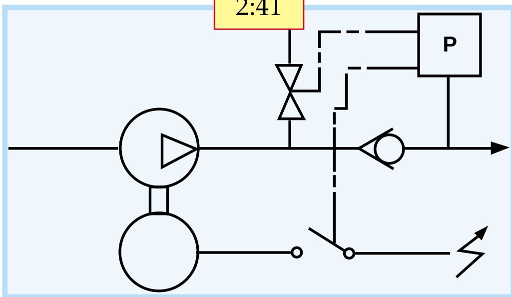


The compressor element then works with a vacuum in the intake and low counter pressure. It is important the pressure relief is carried out quickly and that the relieved volume is small to avoid unnecessary losses during the transition from loaded to unloaded. The system demands a system volume (air receiver), the size of which is determined by the acceptable difference between loading and off-loading pressure and by the permitted number of unloading cycles per hour.

2.5.2.5 Start/stop

Compressors less than 5–10 kW are often controlled by completely stopping the electric motor when the pressure reaches an upper limit value and restarting it when the pressure passes the lower limit value. The method demands a large system volume or large pressure difference between the start and stop pressure, to minimise the load on the electric motor. This is an effective regulation method under the condition that the number of starts per time unit is kept low.

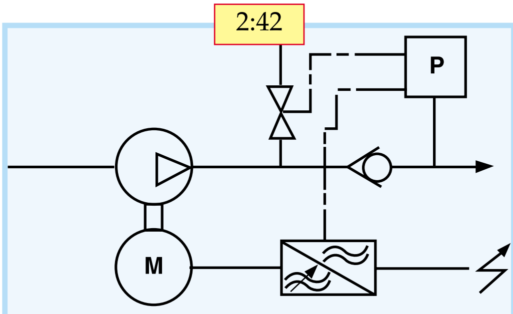
2:41



2.5.2.6 Speed regulation

A combustion engine, turbine or frequency controlled electric motor controls the compressor's speed and thereby the flow. It is an efficient method to attain an equal outgoing pressure and a low energy consumption.

2:42



The regulation range varies with the type of compressor, but is greatest for liquid injected compressors. Frequently speed regulation and pressure relief are combined, with or without a throttled intake, at low degrees of loading.

2.5.2.7 Variable discharge port

The capacity of screw compressors can be regulated by moving the position of the discharge port in the housing, in the screw's lengthways direction, towards the intake. However, the method demands high power consumption with sub-loads and is relative unusual.

2.5.2.8 Suction valve unloading

Piston compressors can be effectively relieved by mechanically forcing the intake valves to the open position. Air is then pumped out and in under the position of the piston, with minimal energy losses as a result, often lower than 10% of the loaded shaft power. On double acting compressors there is generally multi-stage off-loading, where one cylinder at a time is balanced to better adapt the capacity to the demand. An odd method used on process compressors is to allow the valve to be open during a part of the piston stroke and thereby receive a continuous flow control.

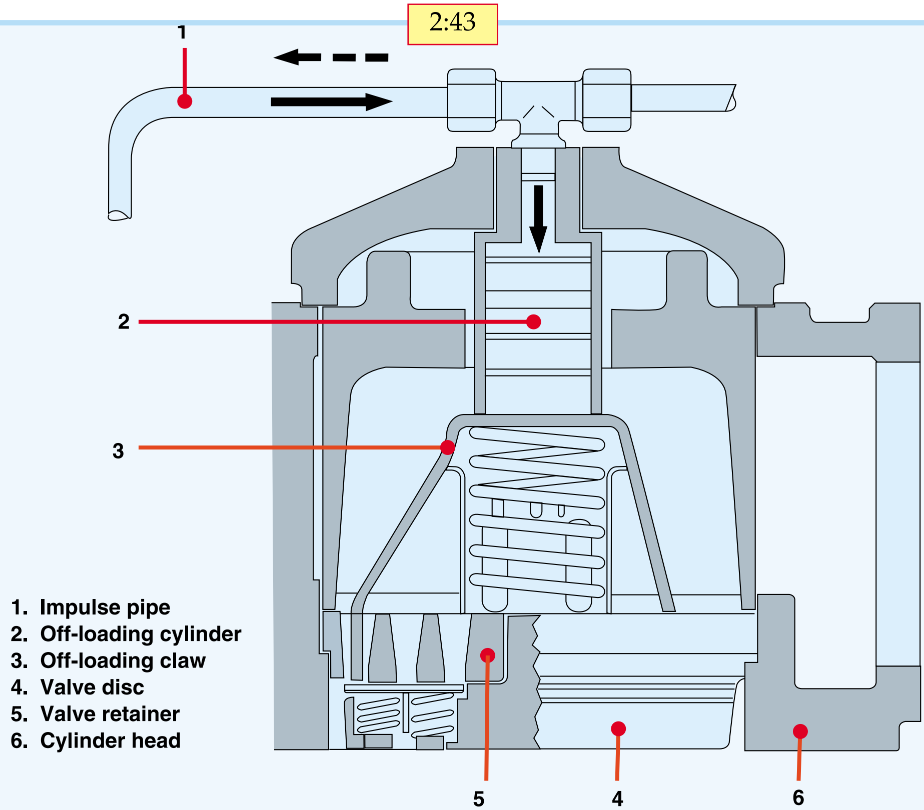
2.5.2.9 Clearance volume

By varying the clearance volume on a piston compressor the degree of filling decreases and thereby the capacity. The clearance volume is varied by means of an externally connected volumes.

2.5.2.10 Load–unload–stop

The most common regulation method used for compressors greater than 5 kW that combines a large regulation range with low losses. In practice a combination of the start/stop and different off-loading systems. See further under 2.5.4.2.

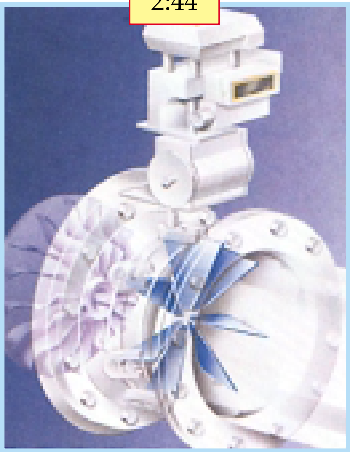
2:43



Off-loading device for a piston compressor

2.5.3 Regulation principles for dynamic compressors

2:44



2.5.3.1 Throttling the intake

The intake can be throttled on a dynamic compressor to continuously reduce the capacity of the compressor. The minimum flow is determined when the pressure ratio reaches the pump limit and the machine becomes unstable (surge).

The regulation range is determined by the design of machine, for example, the number of stages and the impeller design, but also to a large degree by external factors such as counter pressure, suction temperature, and the coolant temperature. The minimum flow often varies between 60% and 85% of the maximum flow.

2.5.3.2 Inlet guide vanes

Vanes arranged as radial blades in the intake cause the in-drawn gas to rotate, at the same time as the flow is throttled. The method acts as throttling, but with a greater regulation range and with improved energy utilisation. Regulation down to 50–60% of the design flow is a typical value. There is also the possibility of increasing the capacity and pressure of the compressor to a certain degree, by turning the vanes in the opposite direction, however, this does impair performance a little.

2.5.3.3 Outlet guide vanes (diffuser)

To further improve the regulation range you can also control the flow in the compressor stage's diffuser. Regulation down to 30% with maintained pressure is common. Normally usage is limited to single stage compressors, due to the complexity and increased costs.

2.5.3.4 Pressure relief

The original method of regulating a dynamic compressor was also to use a pressure relief valve, which releases excess compressed air into the atmosphere. The method works in principle as pressure relief on a displacement compressor.

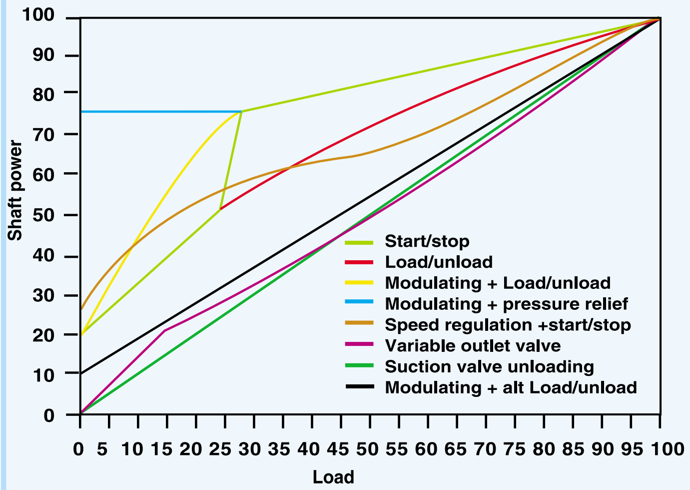
2.5.3.5 Load–unload–stop

While throttling the compressor's inlet is limited by the pump limit, this can be solved in one of two ways:

1. Modulating. The excess flow is released into the atmosphere (or intake), however, with unchanged energy consumption.
2. Auto Dual. The regulation system virtually fully closes the intake valve at the same time as the compressor's outlet is opened to the atmosphere (compare the displacement compressor). However, the off-loading power is relatively high, over 20% of the full load power, depending on the design of the impeller, etc.

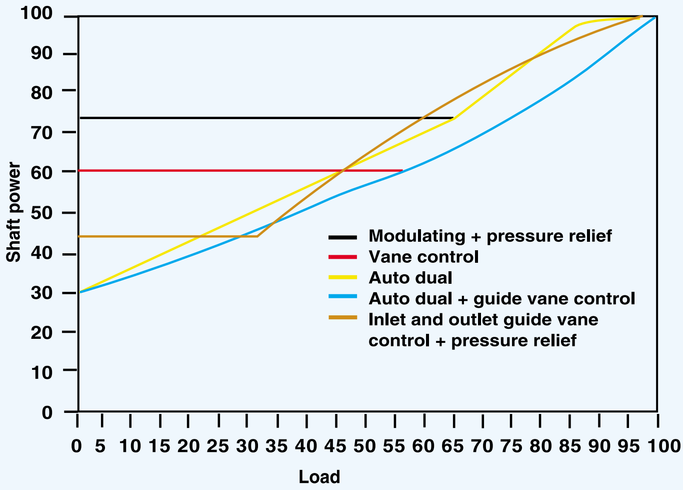
2.5.3.6 Speed regulation

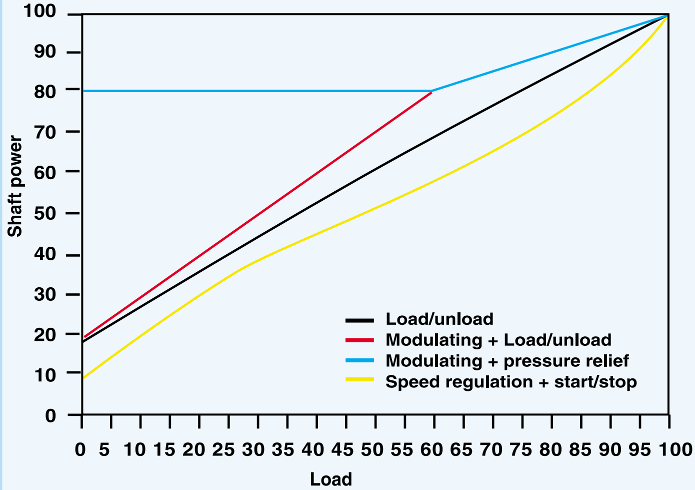
Speed regulation is commonly used on compressors where the flow shall be regulated and the pressure is permitted to vary. With constant pressure regulation, speed regulation gives no benefits when compared with other regulation systems.



The relation between the shaft power and flow for oil lubricated compressors with different regulation systems and combinations of regulation systems.

The relation between the shaft power and flow for dynamic compressors with different regulation systems and combinations of regulation systems.





The relation between the shaft power and flow for oil-free screw compressors with different regulation systems and combinations of regulation systems.

2.5.4 Control and monitoring

2.5.4.1 General

Regulation principles for different compressors are taken up in chapters 2.5.2 and 2.5.3. To control compressors according to these principles requires a regulation system that can either be intended for an individual compressor or an entire compressor installation.

Regulation systems are becoming more advanced and development goes quickly. Relay systems have been replaced by programmable equipment (PLC), which in turn are being replaced by product adapted systems based on microcomputers. The designs are often an attempt to optimise operations and cost.

This section deals with a few of the control and monitoring systems for the most common types of compressor.

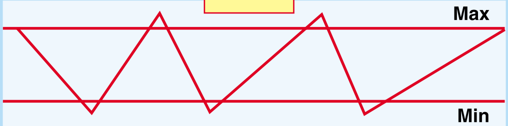
2.5.4.2 Load–unload–stop

The most common regulation principles for displacement compressors are "produce air"/"don't produce air" (loaded/unloaded), see 2.5.2.4 and 2.5.2.5.

When air is required a signal is sent to a solenoid valve, which in turn guides the compressor's intake damper to the fully open position. The damper is either fully opened (loaded) or fully closed (unloaded), there is no intermediate position.

The traditional control, now common on smaller compressors, has a pressure switch placed in the compressed air system that has two settable values, one for the minimum pressure (= loaded) and one for maximum pressure (unloaded). The compressor will then work within the limits of the set values, for example, 0.5 bar. If the air requirement is small or nothing the compressor runs off-loaded (idling). The length of the idling period is limited by a timer (set e.g. to 20 minutes). When the time elapses, the compressor stops and does not start again until the pressure has dropped to the minimum value. This is the traditional tried and trusted control method. The disadvantage is that it gives slow regulation.

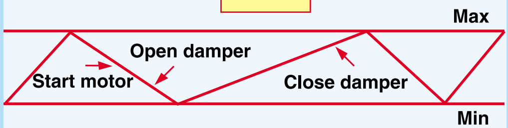
2:46



Pressure band, min–max, within which the compressor works. "Min" = load, "Max" = off-load.

A further development of this traditional system is to replace the pressure switch with an analogue pressure transducer and a fast electronic regulation system. The analogue transducer can, together with the regulation system, sense how quickly the pressure in the system changes. The system starts the motor and controls the opening and closing of the damper at the right time. This gives quick and good regulation within ± 0.2 bar.

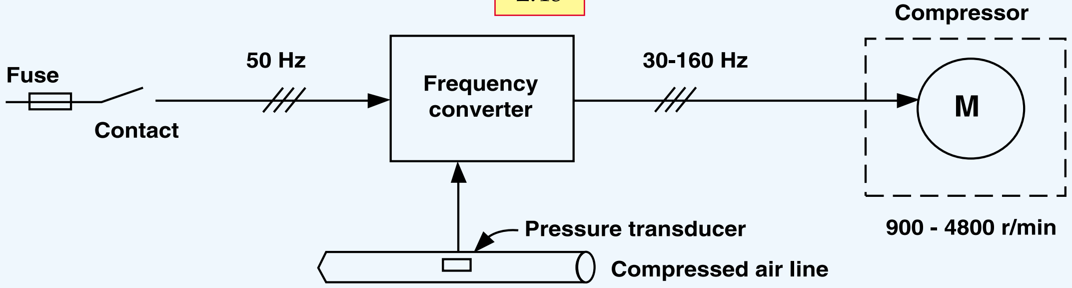
2:47



An advanced regulation system can send signals to the motor, starter and regulator at "the right time".

If no air is used the pressure will remain constant and the compressor will run off-loaded (idling). The length of the idling period is controlled by how many starts and stops the electric motor can withstand without becoming too hot and by the overall operating economy. The latter is possible as the system can analyse trends in air consumption and thereby decide whether to stop the motor or continue to idle.

2:48



A system with a speed controlled compressor

2.5.4.3.Speed control

Compressors with a power source whose speed is controlled electronically provide a great opportunity to keep the compressed air constant within a very tight pressure range.

A frequency converter, which regulates the speed on a conventional induction motor, is an example of such a solution. The compressor's capacity can be adapted exactly to the air requirement by continuously and accurately measuring the system pressure and then allow the pressure signals to control the motor's frequency converter and thereby the motor's speed. The pressure within the system can be kept within ± 0.1 bar.

2.5.5 Control and monitoring

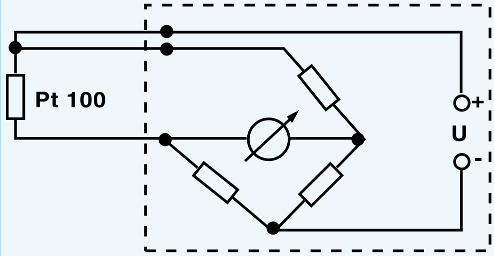
All compressors are equipped with some form of monitoring equipment to protect the compressor and prevent production downtime. The transducer is used to sense the current condition of the installation. Information from the transducers is processed by the monitoring system, which gives a signal to, e.g. an actuator.

A transducer for measuring the pressure or temperature often consists of a sensor and a measurement converter. The sensor senses the quantity to be measured. The measurement converter converts the sensor's output signal to an appropriate electrical signal that can be processed by the control system.

2.5.5.1 Temperature measurement

A resistance thermometer is normally used to measure the temperature. This has a metal resistor as a transducer, whose resistance increases with the temperature. The change in resistance is measured and converted to a signal of 4–20 mA. Pt 100 is the most common resistance thermometer. The nominal resistance at 0°C is 100Ω.

2:49



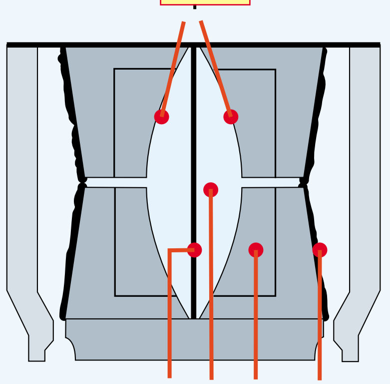
An example of a 3 wire connection using a resistance thermometer of 100Ω. The resistance thermometer and connectors are connected on a bridge.

The thermistor is a semiconductor, whose resistance changes with the temperature. It can be used as a temperature controller, for example, on an electric motor. PTC, Positive Temperature Coefficient, is the most common type. The PTC has an insignificant change in resistance with increased temperature up to a reference point, where the resistance increases with a jump. The PTC is connected to a controller, which senses the "resistance jump" and gives a signal, for example, stop the motor.

2.5.5.2 Pressure measurement

A pressure sensing body, e.g. a diaphragm is used to measure the pressure. The mechanical signal from the diaphragm is then converted to an electrical signal, 4–20 mA or 0–5 V.

2:50

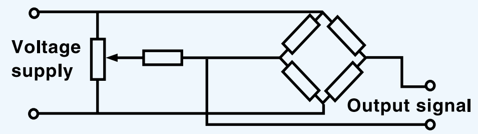


Example of a capacitive system for pressure measurement

The conversion from a mechanical to an electrical signal can take place in different measurement systems. In a capacitive system, pressure is transferred to a diaphragm. The position of the measurement diaphragm is sensed by a capacitor plate and is converted in a measurement converter to a direct voltage or direct current, proportional to the pressure.

The resistive measurement system consists of a strain gauge, connected in a bridge connection and attached to the diaphragm. When the diaphragm is exposed to

2:51



Bridge connection with strain gauge.

pressure a low voltage (mV) is received. This is then amplified to a suitable level. The piezo electric system is based on specific crystals (e.g. quartz) generating electrical charges on the surface of the crystal. The charges are proportional to the force (pressure) on the surface.

2.5.5.3 Monitoring

Monitoring equipment is adapted according to the type of compressor, which entails a large range when concerning the scope of equipment. A small piston compressor is only equipped with a conventional overload cut-out for the motor, while a large screw compressor can have a number of cut-outs/transducers for overloading, temperature and pressure, etc.

On smaller, more basic machines the control equipment switches off the compressor and the machine is blocked for restarting when a cut-out gives an alarm value. A warning lamp, can in some cases, indicate the cause of the alarm.

Compressor operations can be followed on a control panel on more advanced compressors, for example, by directly reading off the pressure, temperature and status, etc. If a transducer value approaches an alarm limit the monitoring equipment gives off a warning. Measures can then be taken before the compressor is switched off. If the compressor is stilled stopped by an alarm, the restart of the compressor is blocked until the fault has been rectified or is reset by hand.

Trouble shooting is significantly facilitated on compressors equipped with a memory where data on, e.g. temperatures, pressure and operating status are logged. The capacity of the memory covers, for example, the last 24 hours. By using this it's possible to produce trends over the last day and then, using logical trouble shooting, quickly track down the reason for the downtime.

2.5.6 Comprehensive control system

Compressors that are a part of a system of several machines should have co-ordinated compressor operations. There are many factors that speak for a comprehensive control system. The division of operating times between machines reduces the risk of unexpected stoppages. Servicing compressors is also easier to plan. Standby machines can be connected if something should occur during operations.

2.5.6.1 Starting sequence selector

The simplest and most common form of master control system is the well tried and tested start sequence selector. This has the task of equally dividing the operating times and starts between the connected compressors. The start sequence can be switched manually or automatically according to a time schedule. This basic selector utilises an on/off pressure transducer, with one transducer per compressor, which is a simple and practical solution.

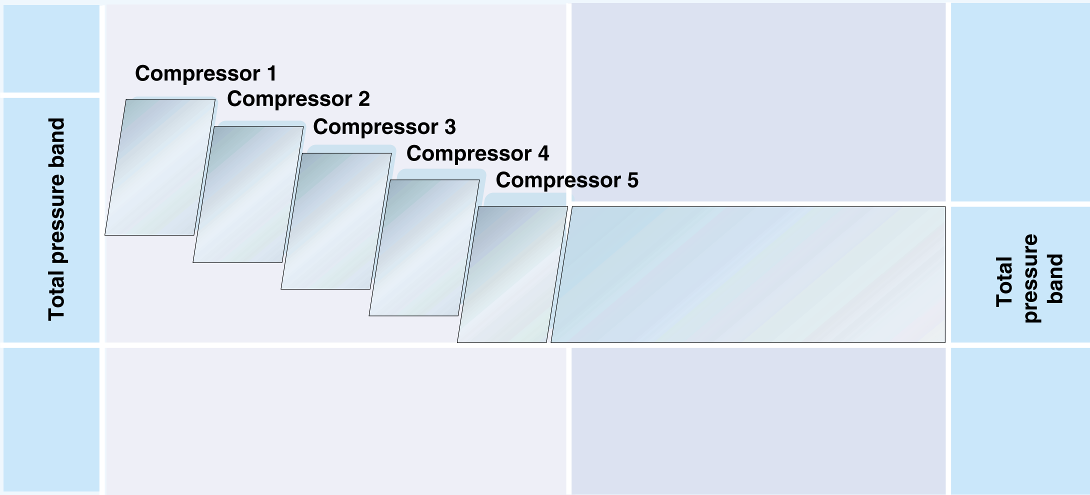
The disadvantage is that there are relatively large steps between the different compressor's loading and off-loading levels, which in turn gives relatively broad pressure bands (the span between maximum and minimum levels) for the installation. Therefore this type of selector should not be used to control more than 2–3 compressors.

A more advanced type of start sequence selector has the same type of sequence control, but with only one, centrally positioned, analogue pressure transducer. This manages to keep the installation's total pressure band within a few tenths of

2:52

A user-friendly monitoring panel shows all the operating parameters for the compressor, for example, pressure and temperatures, with data logically grouped for direct read-off.

2:53



The difference in the pressure band for five compressors controlled by conventional pressure switches (left-hand field) and the same machines controlled by a regulation system with an analogue transducer (right-hand field).

a bar and can control 2–7 machines. A start sequence selector of this type, which selects the machines in fixed sequences, does not take the capacity of the compressors into consideration. It is therefore appropriate that the connected compressors are of approximately the same size.

2.5.7 Central control

Central control in association with compressors usually means relatively intelligent control systems. The basic demand is to be able to maintain a predetermined pressure within tight limits and that the installation's operation shall be as economic as possible. To achieve this, the system must be capable of predicting what will happen in the system and at the same time sense the load on the compressor.

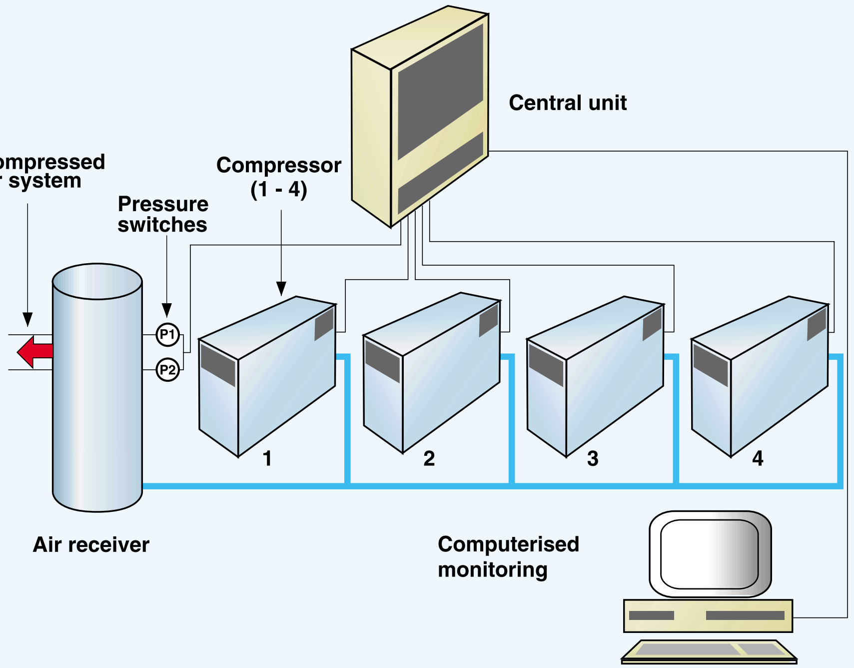
The system senses how quickly the pressure changes, upwards or downwards (i.e. the time derived pressure). Using these values the system can perform calculations that make it possible to predict the air requirement and, for example, off load/load or start/stop the machines. In a correctly dimensioned installation the pressure will be within ± 0.2 bar.

It is extremely important for the operating economy that the central control system selects a compressor or compressor combinations, if compressors of different capacity are included in the system.

The compressors shall run virtually continuously loaded and thereby minimise idling to give the best economy.

An another advantage of a comprehensive control system is that it is generally possible to connect older machines to these systems and thereby, in a relatively easy manner, modernise the entire compressor installation. Operations become more economic and availability increases.

2:54



A centrally controlled compressor installation

2.5.8 Remote monitoring

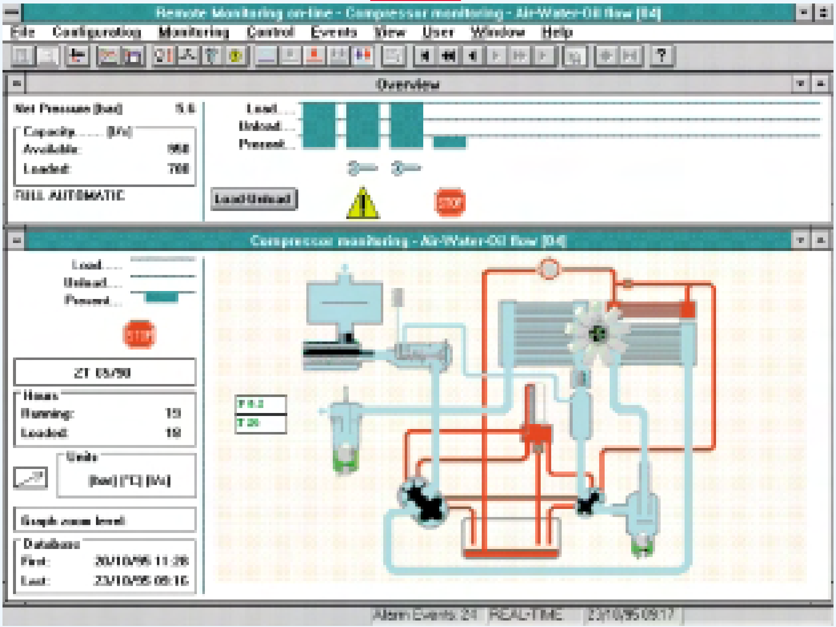
In a number of compressor installations there may be a need of monitoring and controlling compressor operations from a remote location. On smaller installations it is fairly easy to connect an alarm, operating indicator, etc. from the compressor. Normally it is also possible for remote starting and stopping.

On larger installations, where immense value is at stake, central monitoring can be motivated. It should consist of equipment that gives a continuous overview of the system, but where it is also possible to access individual machines to control details such as the intercooler pressure, oil temperature/ etc.

The monitoring system should also have a memory, so that it is possible to produce a log of what has happened during the past 24 hours. The log makes up the basis for trend curves, where it is possible to read off whether any values have a tendency of deviating from the default. The curves can form the basis for continued operations or a planned stop.

The system frequently presents compressor installation status reports in the form of different levels, from a total overview to detail status for individual machines.

2:55



An overview display with remote monitoring. The upper section shows the installation status. Three machines in operation, one stopped. In the lower section details for compressor 4 are shown, among others, flow chart for the compressed air, cooling water and oil as well as prevailing compressor data.

Chapter 3

Dimensioning compressor installations

3.1 Dimensioning compressor installations

3.1.1 General

A number of decisions must be made when dimensioning a compressed air installation for it to suit the user's needs, give the best operating economy and be prepared for future expansion.

The foundation is the applications or process that will use the compressed air. Therefore, you must start by mapping out these to gain the correct basis for continued dimensioning.

The areas to be looked at are calculation or assessment of the air requirement and the reserve capacity and the space for future expansion. The working pressure is a critical factor, as this significantly affects the energy consumption. Sometimes it can be economical to use different compressors for different pressure ranges.

The quality of the compressed air is not just a question of the water content, but has also become increasingly directed towards environmental issues. Odour and the microorganism content are important factors that can affect the product quality, rejections, the working environment and the outdoor environment. The issue of whether the compressor installation should be centralised or decentralised affects the space requirement and perhaps future expansion plans. From economic and environmental standpoints it is becoming more important to investigate the possibilities of recovering energy at an early stage, this often gives very quick return on the investment.

It is important to analyse these types of issues with regard to current as well as future requirements. It is then possible, and only then, to design an installation offering sufficiently flexibility.

3.1.1.1 Calculating the working pressure

The compressed air equipment in an installation determines the requisite working pressure. The right working pressure does not just depend on the compressor but also on the design of the compressed air system with piping, valves, compressed air dryers, filters, etc.

Different types of equipment can demand a different pressure in the same system. Normally the highest pressure determines the requisite installation pres-

3:1

Connected equipment	Nominal air requirement	Utilisation factor max/min	Total air requirement max/min
Tools, total			
Production lines, total			
Process lines, total			

The air requirement for connected equipment is obtained from, e.g. tool catalogues and descriptions of production equipment. By analysing and assessing the utilisation factor you can easily attain upper and lower limits for the overall air requirement.

sure and other equipment is fitted with reducing valves at the point of consumption. In more extreme cases the method can be uneconomical and a separate compressor for special needs can be a solution.

Also bear in mind that the pressure drop increases quickly with an increasing flow. If a change in consumption can be expected, it is makes economic sense to adapt the installation to these conditions.

Filters, special dust filters, have a low initial pressure drop, but in time become blocked and are replaced at the recommended pressure drop, which here will be a factor in the calculation. The compressor's flow regulation also brings about pressure variations and shall be included in the assessment. It may be appropriate to systematise the calculations according to the following example:

Description	Pressure drop bar(e)
End user	6
Final filter	0.1–0.5
Pipe system	0.2
Dust filter	0.1–0.5
Dryer	0.1
Compressor's regulation range	0.5
Compressor's max working pressure	7.0–7.8

Primarily it is the end user together with the pressure drop between the compressor and the consumer that determines the pressure that the compressor needs to produce. By, as in this example, adding the pressure drop in the system the working pressure can be determined.

3.1.1.2 Calculating the air requirement

The nominal compressed air requirement is determined by the air consumers. This is calculated as a sum of the air consumption for all tools, machines and processes to be connected, bearing in mind the utilisation factor that experience tells us will be pertinent. Additions for leakage, wear and future changes in the air requirement must also be considered.

A simple method to estimate the present and future air requirement is to compile the air requirement for connected equipment and the utilisation factor.

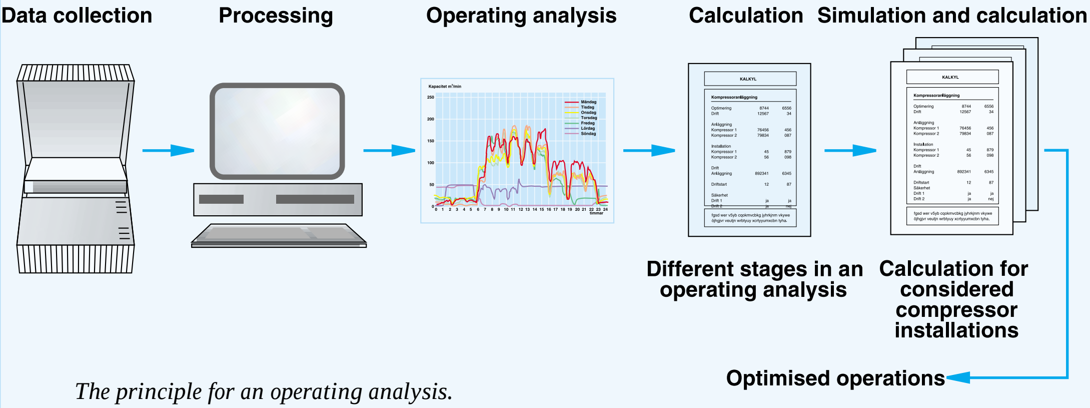
This type of calculation requires a list of machines, supplemented with respective machine's air consumption and expected utilisation factor. If you do not have data for the air consumption or utilisation factor, standard values can be used. The utilisation factor for tools can be difficult to estimate, therefore calculation values should be compared with measured consumption in similar applications.

For example, when large power consumers such as grinders and sand-blasting machines are used, it is frequently for long periods (3–10 min) under continuous operation, despite the low overall utilisation factor. This does not really correspond to intermittent operation, which is why it is necessary to estimated how many machines will be used simultaneously to judge the total maximum air consumption.

The compressor capacity is essentially determined by the total nominal compressed air requirement. The compressor's free output flow rate should cover this rate of air consumption. The calculated reserve capacity is primarily determined by the cost of lost production with a possible compressed air failure.

The number of compressors and the mutual size is determined principally by the required degree of flexibility, control sys-

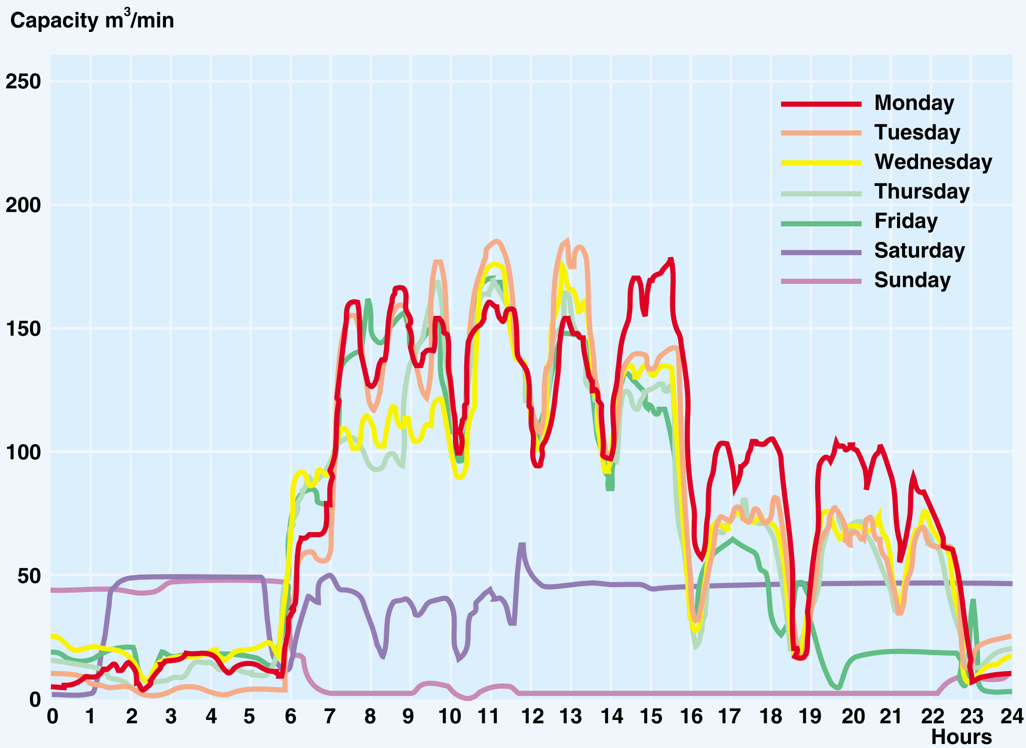
3:2



The principle for an operating analysis.

tem and energy efficiency. In an installation where, due to reasons of cost, only one compressor shall answer for the compressed air supply, the system can be prepared for the quick connection of a portable compressor in connection with servicing. An older compressor, used as a reserve source, can be used for inexpensive reserve power.

3:3



During an operating analysis the compressed air production is continuously measured, for example, during a week. This is the results in diagram form for a compressor with the capacity 260 m³/min where the different curves represent the days of the week.

3.1.1.3 Measuring the air requirement

An operating analysis provides key factors about the compressed air requirement and forms the basis for assessing how much compressed air it is best to produce. Most industrial companies develop continuously, which means that the compressed air requirement also changes. It is therefore important that the compressed air supply is based on the current prevailing conditions, at the same time as an appropriate margin for expansion is built into the installation.

Operating analysis involves the measurement of operating data, possibly supplemented with the inspection of an existing compressed air installation during a suitable period of time. The analysis should comprise at least one week of operations and the measurement period should be selected with care in order to serve as a typical case to ensure that a relevant picture is obtained. The stored data also provides an opportunity to simulate different measures and changes in compressor operations and to analyse the significance for the installation's overall economy.

Factors such as the loading times and off-loading times also have a bearing on the total assessment of compressor operations. These provide the basis to assess the loading factor and the compressed air requirement, spread over a day or working week. Accordingly, the loading factor can not just be read off on the compressor's running hour meter.

An operating analysis also gives a basis for the potential energy recovery. Frequently more than 90% of the supplied energy can be recovered. Furthermore, the analysis can provide answers relating to dimensioning as well as the operating method for the installation. For example, the working pressure can often be reduced at certain times and the control system can be modified in order to improve compressor usage with changes in production. Another important factor is to check whether there is any leakage.

For the production of small quantities of air during the night and weekends, you must consider whether it is worthwhile installing a smaller compressor to cover this requirement.

3:4

Hole diameter: mm	**1**	**3**	**5**	**10**
Leakage, (l/s) at 6 bar	**1**	**10**	**27**	**105**
Power loss, kW at the compressor	**0,3**	**3,1**	**8,3**	**33**

Even small leakage can result in large costs and downtime.

3.1.2 Centralisation or decentralisation

3.1.2.1 General

There are several factors that affect the choice between one large or several smaller compressors to meet the same compressed air requirement. For example, the cost of a production stoppage, the availability of electricity, loading variations, costs for the compressed air system and the available floor space.

3.1.2.2 Centralised compressor installations

A centralised compressor installation is in most cases the solution of choice, as it is less expensive than several, locally situated compressors. Compressor plant can be efficiently interconnected, which result in lower energy consumption. A central installation also involves lower monitoring and maintenance costs as well as better conditions for recovering energy. The overall, requisite floor area for the compressor installation is less. Filters, coolers and other auxiliary equipment and the air intake can be optimally dimensioned and installed. Noise insulation will also be easier to fit.

A system comprising several, different sized compressors in a central installation can be sequence controlled to improve efficiency. One large compressor has difficulty in meeting large variations in the compressed air requirement, without efficiency dropping.

For example, systems with one large compressor are often supplemented with a smaller compressor, for use during periods such as a night shift or at weekends. Another factor worth considering is the effect the start of a large electric motor has on the mains supply.

3.1.2.3 Decentralised compressor installations

A system with several decentralised compressors involves a smaller, simpler compressed air system. The disadvantages of decentralised compressors is the difficulty in inter-regulating the compressed air supply, the expense and that maintenance work is more demanding as well as it is hard to maintain a reserve capacity. Decentralised compressors can be utilised to maintain the pressure in a system with a large pressure drop if the intermediate processes temporarily draw too much air. Otherwise an alternative with extremely short peaks is to solve the problem by positioning a buffer (air receivers) at strategic places.

A unit or building normally supplied from a compressed air central and which is the sole consumer of compressed air at specific periods can be sectioned off and supplied with its own compressor. The advantage of this is you avoid "feeding" any leakage in the remaining part of the system and that the localised compressor can be adapted to the smaller requirement.

3.1.3 Dimensioning at high altitude

3.1.3.1 General

The ambient pressure and temperature diminish at heights above sea level. This affects the pressure ratio, for compressors as well as the connected equipment, which in practice means an influence on the power and air consumption. At the same time the changes also affect the available rated power from electric motors and combustion engines.

You should also be aware of how the ambient conditions influence the end user. Is it a specific mass flow rate, e.g. in a process or is it volume flow rate you require? Is it the pressure ratio, absolute pressure or over pressure that was used for dimensioning? Is the compressed air temperature significant?

All these create different conditions for dimensioning a compressed air installation installed at a high altitude and can be fairly complex to calculate. If you feel unsure you should always contact the manufacturer of the equipment.

3:5

Atmospheric pressure

Height below/ above sea level	Pressure bar	Temperature °C
-1000	1.138	21.5
-800	1.109	20.2
-600	1.080	18.9
-400	1.062	17.6
-200	1.038	16.3
0	1.013	15.0
200	0.989	13.7
400	0.966	12.4
600	0.943	11.1
800	0.921	9.8
1000	0.899	8.5
1200	0.877	7.2
1400	0.856	5.9
1600	0.835	4.6
1800	0.815	3.3
2000	0.795	2.0
2200	0.775	0.7
2400	0.756	-0.6
2600	0.737	-1.9
2800	0.719	-3.2
3000	0.701	-4.5
3200	0.683	-5.8
3400	0.666	-7.1
3600	0.649	-8.4
3800	0.633	-9.7
4000	0.616	-11.0
5000	0.540	-17.5
6000	0.472	-24.0
7000	0.411	-30.5
8000	0.356	-37.0

The table shows the standardised pressure and temperature variations at different heights. The pressure is dependent on the weather and varies approx. ± 5%, while the local season dependent temperature variations can be considerable.

3.1.3.2 The effect on a compressor

To choose the right compressor where the ambient conditions differ from those stated on the data sheet, you should take the following factors into consideration:

- **Height above sea level or ambient ressure**
- **Ambient temperature**
- **Humidity**
- **Coolant temperature**
- **Type of compressor**
- **Power source**

These factors primarily affect the following:

- **Max. working pressure**
- **Capacity**
- **Power consumption**
- **Cooling requirement**

The most important factor is the intake pressure variations at altitude. For example, this means a compressor, with a pressure ratio of 8.0 at sea level, will have a pressure ratio of 11.1 at an altitude of 3000 metres (under the condition that the working pressure is constant). This affects the efficiency and thereby the power requirement. To what degree is dependent on the type of compressor and the design as set out in figure 3:6.

The ambient temperature, humidity and coolant temperature interact and

3:6

Compressor type	Reduction for each 1000 metres above sea level: Free output flow rate %	Reduction for each 1000 metres above sea level: Mass flow or Normal flow %
Single stage oil-free screw compressor	0.3	11
Two stage oil-free screw compressor	0.2	11
Single stage oil injected screw compressor	0.5	12
Single stage piston compressor	5	17
Two stage piston compressor	2	13
Multi-stage centrifugal compressor	0.4	12

A rule of the thumb for the altitude's effect on the compressor at 7 bar(e) working pressure and constant ambient temperature. Bear in mind that each compressor has a top pressure ratio that can not be exceeded.

3:7

Height above sea level, metres	Ambient temperature, °C					
	<30	30-40	45	50	55	60
1000	107	100	96	92	87	82
1500	104	97	93	89	84	79
2000	100	94	90	86	82	77
2500	96	90	86	83	78	74
3000	92	86	82	79	75	70
3500	88	82	79	75	71	67
4000	82	77	74	71	67	63

The table shows the permitted load in % of the electric motor's rated power.

affect the compressor's performance to different degrees on single or multi-stage compressors, dynamic compressors and displacement compressors.

3.1.3.3 Power source

3.1.3.3.1 Electric motors

Cooling becomes impaired on electric motors by the thinner air at high altitude. It should be possible for standard motors to work up to 1000 m and with an ambient temperature of 40°C without the rated data deteriorating. With greater heights table 3:7 can be used as a guideline for standard motors. Notice that for some types of compressor the motor performance is impaired more than the compressor's requisite shaft power at high altitude.

3.1.3.3.2 Combustion engines

A reduction in the ambient pressure, a temperature increase or a reduction in humidity reduce the oxygen content in the intake air and thereby the extractable power from the engine. The degree of shaft power degradation depends on the type of engine and its breathing method as set out in figure 3:8. The humidity plays a lesser part (<1% / 1000 m) when the temperature falls below 30°C.

Notice that the engine power falls more rapidly than the compressor's requisite shaft power, which means that for each compressor/engine combination there is a maximum working height. Generally, you should let respective suppliers calculate and state the specific data that applies to the compressor, engine and air consumption equipment in question.

3:8

Engine type	Power reduction in % per 1000 m	Power reduction in % per 10°C temperature increase
Suction engine	12	3.6
Compressor fed	8	5.4

The table shows how combustion engines are affected by altitude and temperature.

3.2 Air treatment

3.2.1 General

It is a matter of vital importance to the user that the compressed air is of the right quality. If air that contains contamination comes into contact with the final product, rejection costs can quickly become unacceptably high and the cheapest solution can quickly become the most expensive. It is important that you select the compressed air quality in line with the company's quality policy and even attempt to judge future requirements.

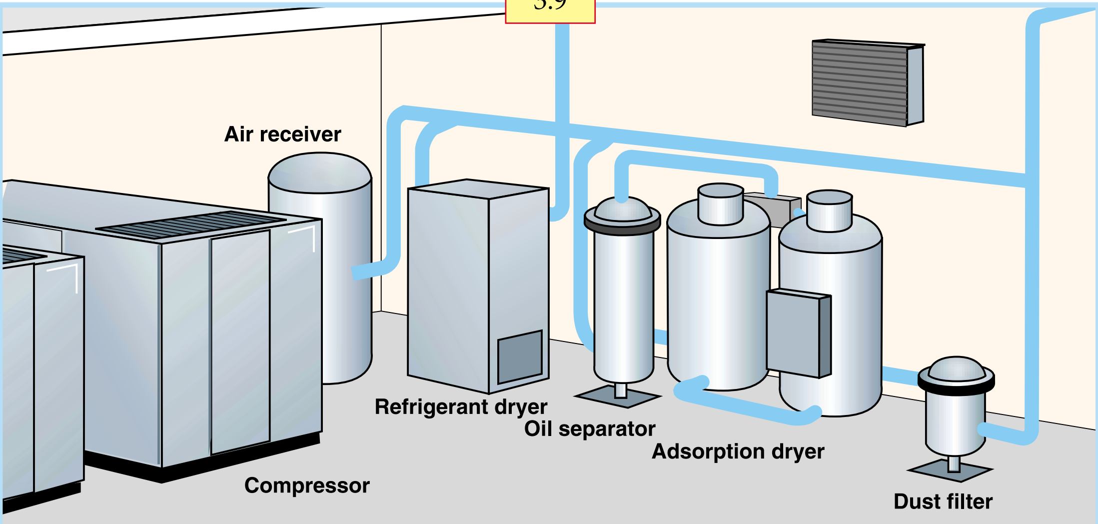
The compressed air can contain unwanted substances, for example, water in drop or vapour form, oil in drop or aerosol form as well as dust. Depending on the compressed air's application area, these substances can impair the production result and even increase costs. The purpose of air treatment is to produce the compressed air quality specified by the consumer.

When the compressed air's part in the process is clearly defined, the answer to which system is the most profitable and efficient will be found. It is a question, among others, of establishing whether the compressed air will come into direct contact with the product or whether, e.g. oil mist can be accepted in the working environment. A systematic method is necessary to select the right equipment.

3.2.2 Water vapour in the compressed air

Air in the atmosphere always contains moisture in the form of water vapour. Some follows with the compressed air and can cause problems. Examples of which are: High maintenance costs, shortened service life and impaired tool performance, high rate of rejection with spray painting and plastic injection, increased leakage, disturbances in the control system and instruments, shorter service life for the

3:9



The main components in a compressed air system when set up and installed. The equipment for air treatment determines the quality of the compressed air, which has a great effect on the economy of the installation.

3:10

ISO 8573-1 quality classes				
Quality class	Contamination Particle size (µm)	Contamination Max. concentration (mg/m^3)	Water Max pressure dew point (°C)	Oil Max. concentration (mg/m^3)
1	0.1	0.1	-70	0.01
2	1	1	-40	0.1
3	5	5	-20	1.0
4	15	8	+3	5.0
5	40	10	+7	25
6	-	-	+10	-

For example: compressed air of quality class 2.2.2. (Contamination: Particles 1 µm and 1 mg/m^3, water: -40°C pdp (pressure dew point), oil: 0,1 mg/m^3)

ISO has quality classified compressed air with regard to the degree of contamination.

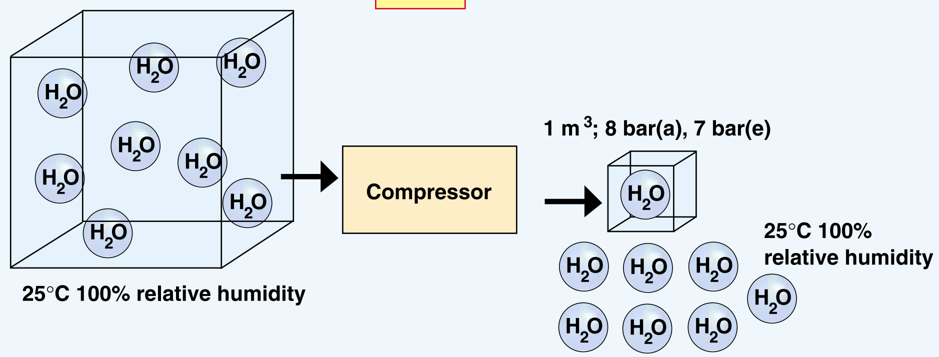
pipe system due to corrosion and more expensive installation. The water can be separated by using accessories, e.g. after-coolers, condensation sepators, refrigerant dryers and adsorption dryers.

A compressor that works with 7 bar(e) overpressure, compresses air to 7/8 the volume. This also reduces the air's ability to hold water vapour by 7/8. The quantity of water that is released is considerable. For example, a 100 kW compressor that draws in air at 20°C and with 60% relative humidity will give off approx. 85 litres of water during an 8 hour shift. Consequently, the amount of water to be separated depends on the compressed air's application area. This in turn is decides which combination of coolers and dryers are suitable.

3.2.3 Oil in the compressed air

The quantity of oil in the compressed air is dependent on several factors, among others, the type of machine, design, age, condition, etc. There are two main types of compressor design in this respect, those working with lubricant in the compression chamber and those working without lubricant. In lubricated compressors the oil takes part in the compression process and also follows with the compressed air fully or partly. However, on modern, lubricated

3:11



A compressor that works with 7 bar(e) overpressure, compresses air to 1/8 of the volume.

piston and screw compressors the oil quantity is only small. For example, in an oil injected, screw compressor the oil content in the air is less than 3 mg/m³ at 20°C. The oil content can be reduced by means of multi-stage filters. If such a solution is chosen it is important to consider the quality limitations, risks and energy costs this brings about.

3.2.4 Microorganisms in the compressed air

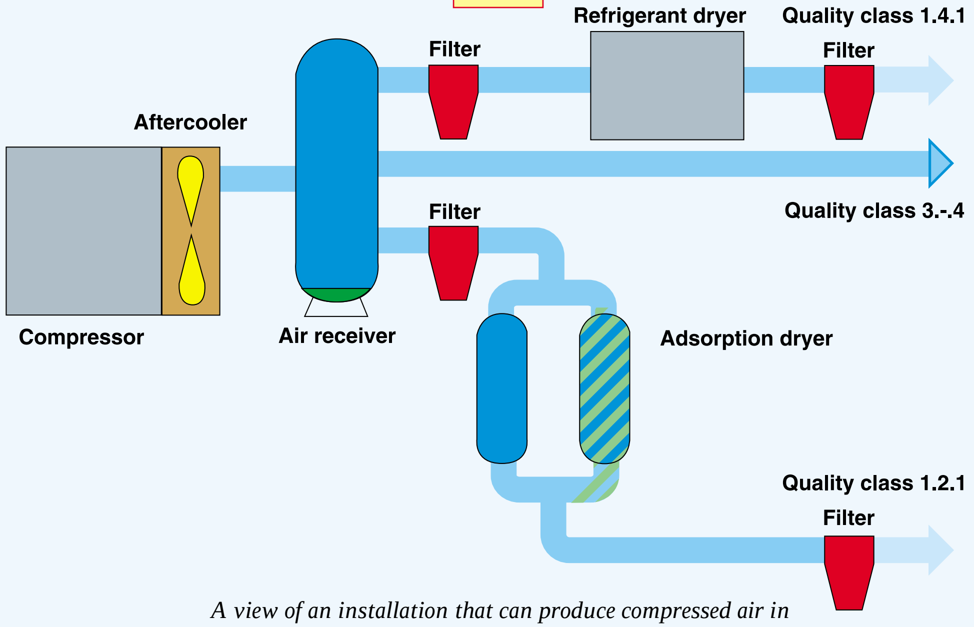
More than 80% of the particles that contaminate the compressed air are less than 2 µm and thereby easily pass through the compressor's intake filter. Thereafter the particles are spread in the pipe system and mix, among others, with the water and oil residue and pipe deposits. This can result in the growth of microorganisms. A filter directly after the compressor can eliminate these risks. Nevertheless, to have pure or sterile compressed air you must have full control over any bacteria growth after the filter.

The picture becomes even more complicated as gases and aerosol can be concentrated into drops (through concentration or electric charging) even after passing several filters. Microorganisms germinate through the filter walls and therefore exist in the same concentrations on the inlet and outlet sides of the filter.

During investigations it has been established that microorganisms thrive in compressed air systems with undried air and thereby high humidity (100%). Contamination smaller than 1 µm and thereby microorganisms, can pass unimpeded through the compressor's intake filter.

Oil and other contamination act as nutrients for growth. The most decisive factor is to dry the air to a humidity of <40%, which is achieved by using an adsorption dryer and at room temperature also with a refrigerant dryer.

3:12



A view of an installation that can produce compressed air in different quality classes in accordance with ISO 8573-1.

3.2.5 Filters

Modern fibre filters are very efficient at removing oil. However, it is difficult to exactly control the quantity of oil remaining in the air after filtration as the temperature, among others, has an important effect on the separation process. Efficiency is also affected by the oil concentration in the compressed air and the amount of free water.

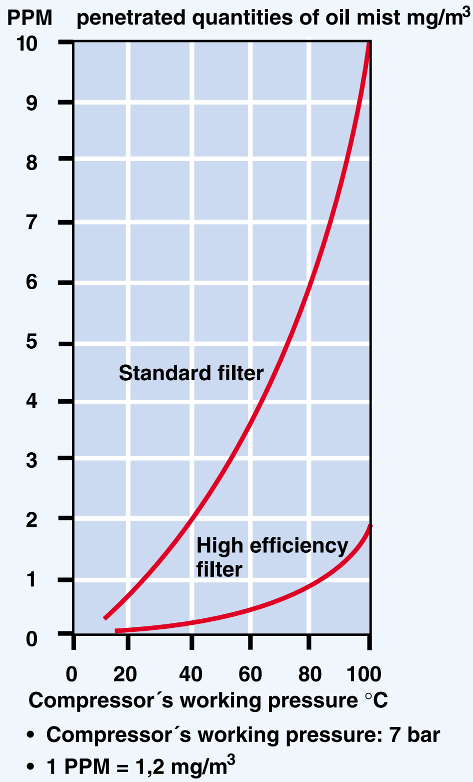
To achieve the best results the air should be as dry as possible. Oil, carbon and sterile filters all give bad results if there is free water in the air (the filter specifications do not apply in such conditions). Fibre filters can only remove oil in the form of droplets or as aerosols. Oil vapour must be removed using a filter with active carbon. A correctly installed fibre filter, together with a suitable prefilter, can reduce the quantity of oil in the compressed air to approximately 0.01 mg/m³ at 21°C. A filter with active carbon can reduce the quantity of oil to 0.003 mg/m³ at 21°C.

Carbon filters should have the correct quality of carbon and dimensioned to give the lowest possible pressure drop. To have the best effect the filters should also be placed as close to the application in question as possible. In addition, they must be checked carefully and replaced relatively frequently. Filters with active carbon only remove air contamination in the form of vapour, for example, oil. Sterile filters shall withstand sterilisation on site in the pipe system with the help of, e.g., steam or be removed for treatment. A filter's capacity to separate oil from compressed air varies at different operating temperatures.

Data stated in the filter specification always applies at a specific air temperature, normally 21°C. This corresponds approximately with the temperature after an air cooled compressor working in an ambient temperature of 10°C. However, climate and seasonal changes give temperature variations, which in turn affect the filter's separation capacity.

An oil free compressor eliminates the need of an oil filter. This means the compressor can work at a low pressure, which reduces energy consumption. It has been shown in many cases that oil-free compressors are the best solution, both economically and for quality.

3:13



Temperature variations in compressed air result in quality variations with the use of filters.
Note that standard and high effect filters do not have an effect on oil in vapour form.

3.2.6 Aftercooler

The compressed air from the compressor is hot after compression, often 70–200°C. An aftercooler is used to lower the temperature, which also reduces the water content and now is frequently included as standard equipment in a compressor installation. The aftercooler should always be placed directly after the compressor. It is the heat exchanger that cools the hot air, to then precipitate the main part of the condensation water as quickly as possible that would otherwise follow out into the system. The aftercooler can either be water or air cooled and is generally fitted with a water separator with automatic drainage.

3.2.7 Water separator

Most compressor installations are fitted with an aftercooler as well as a water separator, in order to separate as much condensation water as possible from the compressed air. With the right choice and sizing of the water separator an efficiency of 80-90% can be achieved. The remainder follows with the compressed air as water mist into the air receiver.

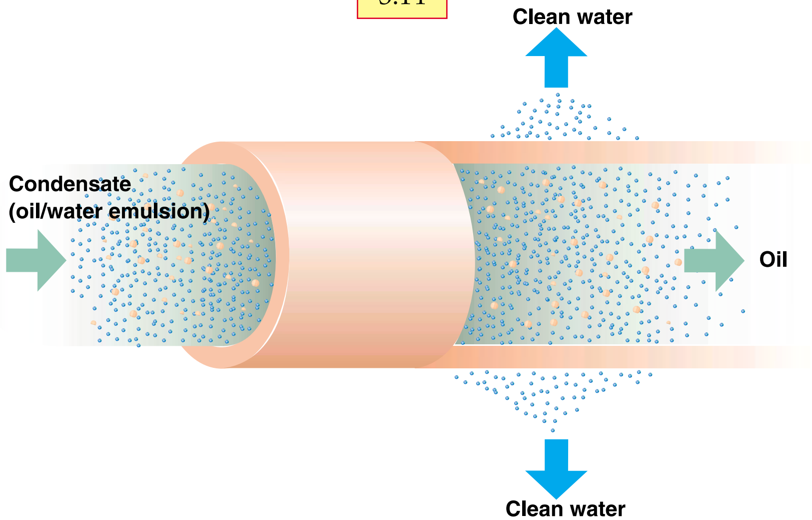
3.2.8 Oil as droplets

Oil in the form of droplets is separated partly in, e.g. an aftercooler, condensation separator or a condensation tap and follows with the condensation water. This oil/water emulsion is classed from an environmental point of view as waste oil and must not be lead off in the sewage system or directly into nature.

New and more stringent laws are continuously being introduced with regard to the handling of environmentally hazardous waste. The drainage of condensation, as well as the collection and drainage, is involved and expensive.

An easy and cost effective solution to the problem is to install an oil/water separator, for example, with a diaphragm filter, which gives clean drainage water and leads the oil off into a special receiver.

3:14



This is how a diaphragm filter for oil separation works. The diaphragm lets through small molecules (clean water), while larger molecules (oils) are kept in the system and can be collected in a container.

3.3 Cooling system

3.3.1 Water cooled compressors

3.3.1.1 General

A water cooled installation makes little demand on the ventilation of the compressor room, as the larger part of the heat produced is led off by the cooling water. The cooling water from a water cooled compressor contains, in the form of heat, approx. 90% of the energy taken up by the electric motor.

A compressor's cooling water system can be designed based on one of three main principles. As an open system without circulating water, as an open system with circulating water, and as a closed circulating system.

3.3.1.2 Open system, without circulating water

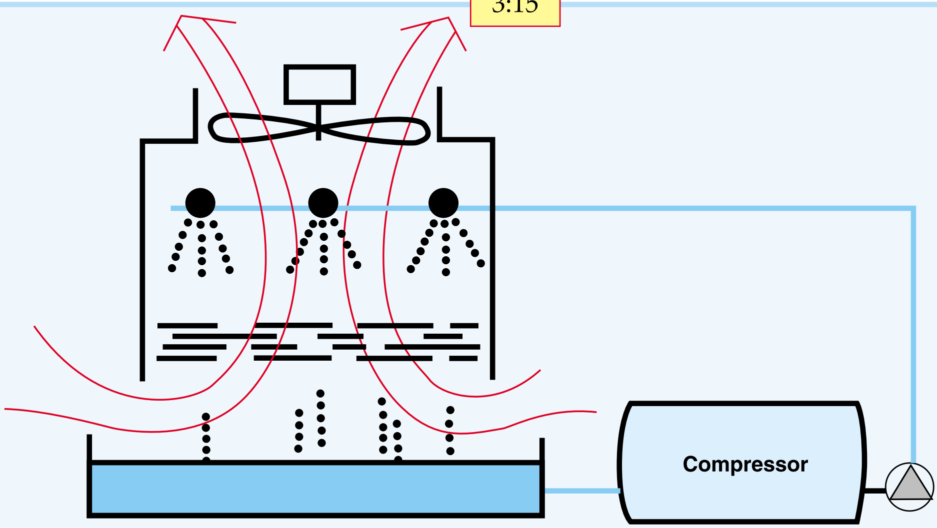
An open system without circulating water means that the water comes from the municipal water mains, a lake, a stream, or a well and is used to cool the compressor and is then discharged as waste water. The system should be controlled by a thermostat, to maintain the desired temperature as well as to govern water consumption. The cooling water pressure should be lower than the pressure the components parts are designed for.

Generally an open system is easy and inexpensive to install, but expensive to run, especially if the cooling water is taken from the municipal water mains. Water from a lake or stream is normally free, but must be filtered and purified to be used without the risk of blocking the cooling system. Furthermore, water rich in lime can result in boiler scale forming in the coolers, bringing about impaired cooling. The same applies to salt water, which however is possible to use if the system is designed and dimensioned accordingly.

3.3.1.3 Open system, circulating water

An open system with circulating water means that cooling water from the com-

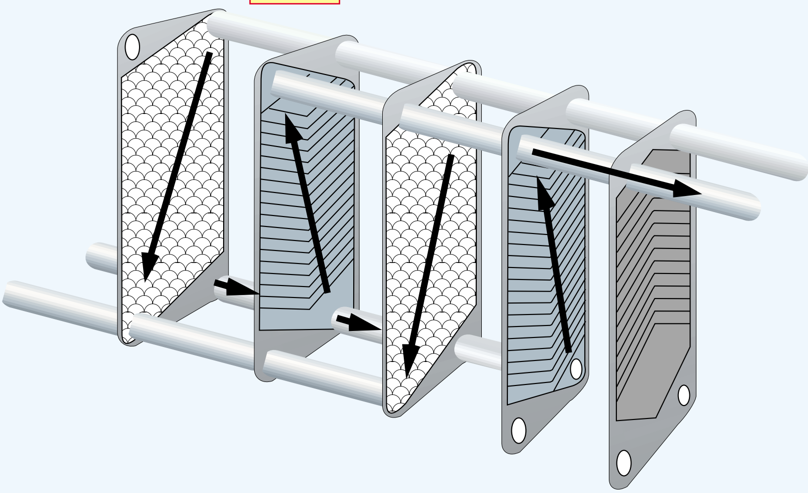
3:15



An open cooling system with circulating cooling water.

3:16

This is how a flat heat exchanger can appear. The advantage of a flat heat exchanger is that it is easy to clean, which makes it possible to indirectly cool the compressor using lake or stream water.



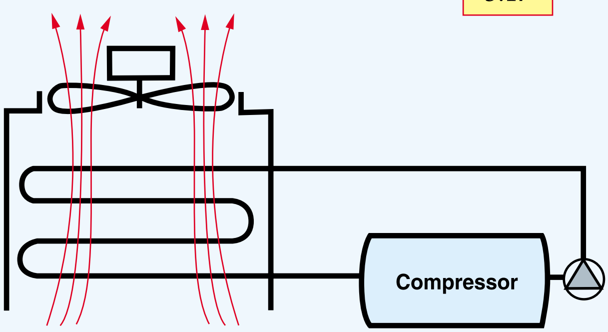
pressor is recooled in a cooling tower. Water is cooled in the cooling tower by allowing it to sprinkle down into a chamber at the same time as surrounding air is blown through, whereby a part of the water vaporises and the remaining water is cooled to 2˚C under the ambient temperature (can vary depending on the temperature and relative humidity). Open systems with circulating water are primarily used when the availability of water is limited. The disadvantage here is that the water becomes contaminated by the surrounding air. The system must be continuously diluted using fresh water due to evaporation.

Dissolvable salts are deposited on the hot metal surfaces, this reduces the thermal dissipation capacity of the cooling tower. The water must be regularly analysed and treated with chemicals to prevent algae from growing in the water. During the winter, when the compressor is not operating, the cooling tower must either be emptied or the water heated to prevent freezing.

3.3.1.4 Closed system

In a compressor cooled with a closed system the same water circulates between the compressor and some form of cooler. This cooler is in turn cooled either by means of another water circuit or the surrounding air. Generally when the water is cooled against another water circuit a flat heat exchanger is used.

3:17



A basic air heater (air cooled heat exchanger) can be found in closed cooling systems for cooling liquids such as water/glycol, oil, etc. In aggressive environments or with aggressive liquids, materials such as stainless steel or titan are used.

When the water is cooled against the surrounding air a cooling battery is used consisting of pipes and cooling flanges. The surrounding air is forces to circulate around the battery by means of one or more fans. The air is normally filtered first to prevent blockage. This method is suitable if the availability of water is limited. The cooling capacity of open or closed circuits is about the same, i.e. the compressor water is cooled by 5°C above the coolant temperature.

If the water is cooled by the surrounding air, the addition of an antifreeze, e.g. glycol is required. The closed cooling water system is filled with pure, softened water.

When glycol is added the compressor system's water flow must be recalculated, as the type and concentration of glycol affects the thermal capacity and the viscosity. The addition of chemical agents also affects the cooling water's capacity to crawl through the connection points.

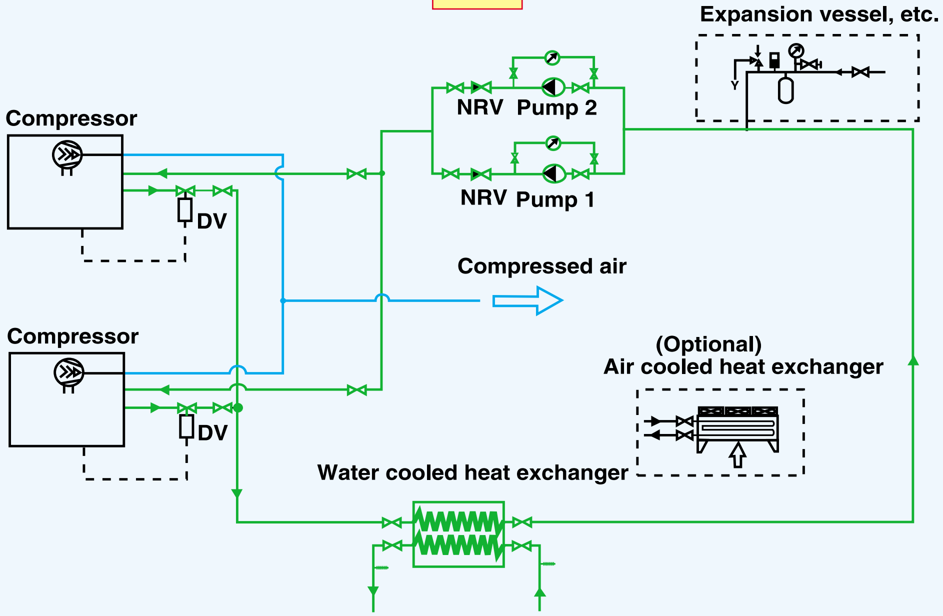
3:18

Freezing point °C	Glycol mixture %	Heat capacity kJ/°C. kg
-10	23	3.850
-15	30	3.650
-20	37	3.450
-25	43	3.350
± 0	0	4.190

The water must be protected from freezing at low temperatures. It is important to remember that the size of the cooler may need to be increased as, for example, a water/glycol mixture has a lower thermal capacity than pure water.

It is also important that the entire system is well cleaned before being filled for the first time. A correctly implemented closed water system requires very little supervision and has low maintenance costs. For installations where the available cooling water is aggressive, it is appropriate to use a cooler designed using a corrosion retardant material such as incoloy.

3:19



This is how a closed cooling system can be made up. The heat exchanger can be water or air cooled.

3.4 Energy recovery

3.4.1 General

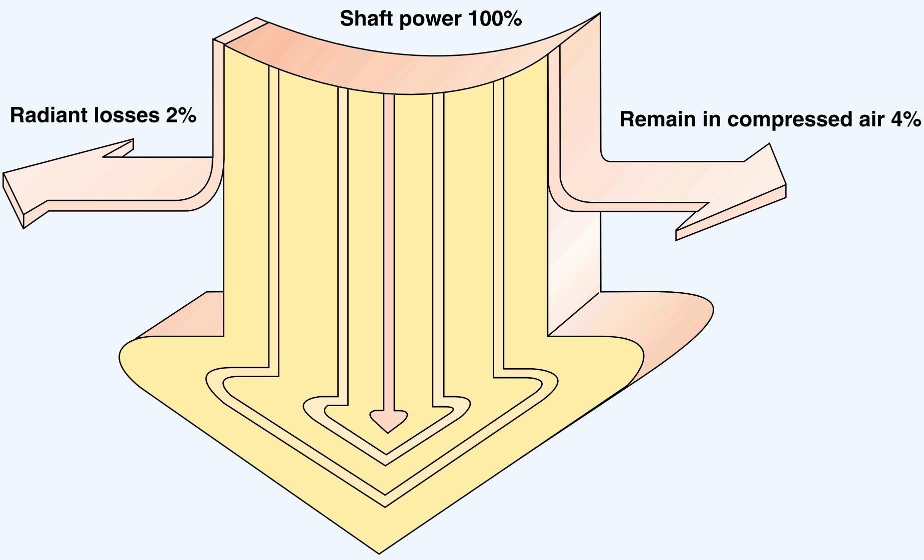
When air is compressed heat is formed. The heat energy is concentrated in the decreasing volume and the excess is led off before the air goes out into the pipe system. For each compressed air installation you must assure yourself that there is sufficient and reliable cooling capacity for the installation. This can take place either by means of the outdoor air or a water system such as municipal water, stream water or process water in an open or closed system.

On many installations producing compressed air there are significant and frequently unutilised energy saving possibilities in the form of energy recovery from the compressors. Energy costs can, in larger industries, amount to 80% of the total cost for the production of compressed air. As much as 94% of the compressor's supplied energy can, for example, be recovered as 90° hot water from large, oil-free screw compressors. This means that each saving measure quickly gives noticeable dividends.

Presuppose that a compressor central in a large industry consumes 500 kW during 8,000 operating hours per annum. This corresponds to no less than approximately 4 million kWh/year. The possibilities to recover this waste heat via hot air or hot water are good.

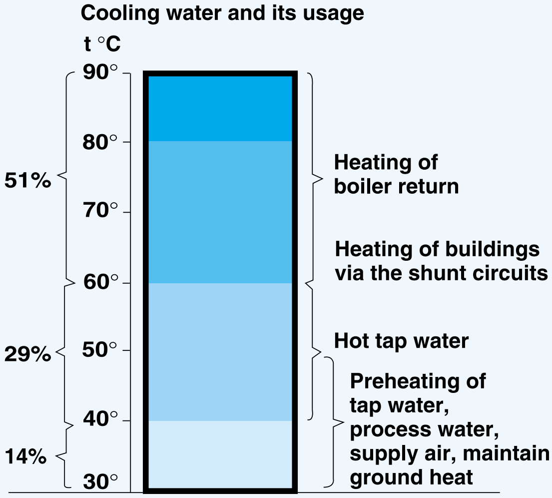
The return on the investment for energy recovery is usually as short as 1–3 years. In addition, energy recovered by means of a closed cooling system is advantageous to the compressor's operating conditions, reliability and service life due to

3:20



As heat is the natural by-product of compression, energy can be recovered in the form of hot water from the compressor's cooling system.

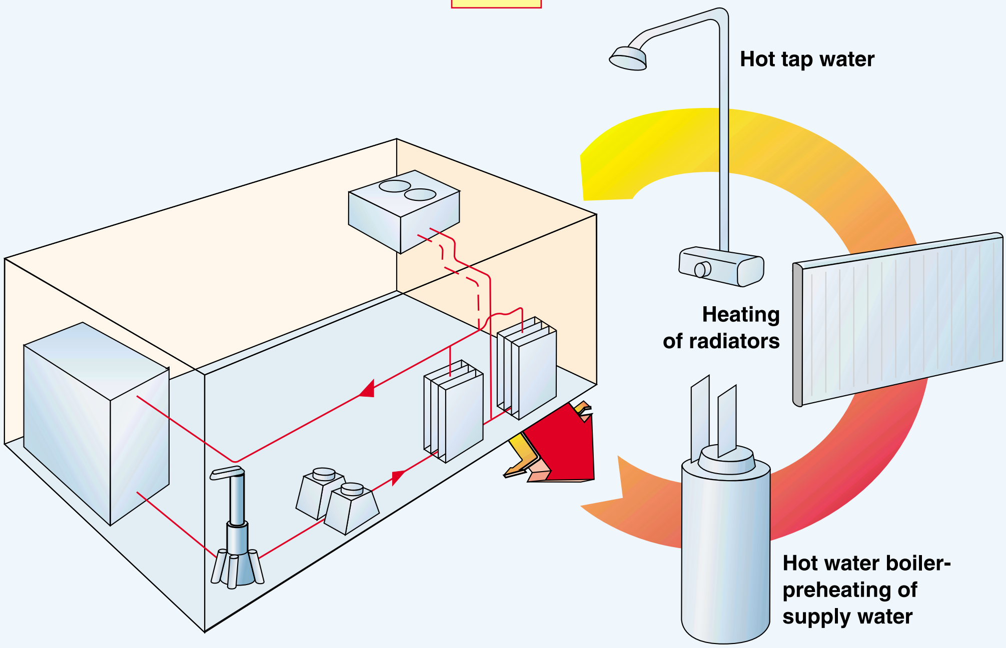
3:21



The diagram illustrates some of the typical application areas for energy recovery from the compressor's cooling water in different temperature ranges. In the highest temperature levels the degree of recovery is the greatest.

an equal temperature level and high cooling water quality to name but a few. The Nordic countries are somewhat of a forerunner and energy recovery has for long time been praxis when it comes to compressors.

3:22



Each compressor installation represents large possibilities for energy recovery. A whole 95% of the compressor's supplied energy can, for example, be recovered from large, oil-free screw compressors.

Most compressors from the major suppliers are today adapted to be supplemented with standard equipment for recovery.

3.4.2 Calculation of the recovery potential

Virtually all energy supplied to a compressor installation is converted to heat. The more energy you can recover and use in other processes, the higher the system's efficiency. The quantity that can be recovered can easily be calculated by the relation:

Recovered energy kWh/year:

$$W = [(K_1 \times Q_1) + (K_2 \times Q_2)] \times T_R$$

Saving/year: $W \times e_p / \eta$

Saved oil m³/year: $W / 68000 \times \eta$

W =Recovered energy Wh/year)

T_R =Time per year when there is a need of recovered power hours/year)

K_1 =Part of T_R with loaded compressor

K_2 =Part of T_R with off-loaded compressor

Q_1 =Available power in coolant with load compressor (kW)

Q_2 =Available power in coolant with off-loaded compressor (kW)

e_p =Energy price

η =Normal heat source´s efficiency

In many cases the degree of recovery can exceed 90%, if the energy you gain through cooling the compressor installation can be taken care of efficiently. The function of the cooling system, distance to the point of consumption, and the degree and continuity of the demand are all decisive factors.

When it is a question of a large thermal flow it can be of interest to look at the possibility of selling the recovered heat energy. A purchaser can be the energy supplier and you can seek agreement forms for investment, suborder and delivery. There is also the possibility of co-ordinating energy recovery from several processes.

3:23

Energy recoverable power

FAD m³/min	Heat flow kW	Saving at 2000 oper.hours/year kWh/year	Oil EO1 m³/year
6.4	34	68 000	10.0
7.4	40	80 000	11.8
11.4	51	102 000	15.0
14.0	61	122 000	17.9
18.7	92	184 000	27.1
21.6	109	218 000	32.1
23.2	118	236 000	34.7
27.9	137	274 000	40.3
34.8	176	352 000	51.8
43.1	215	430 000	63.2
46.9	235	470 000	68.1
46.5	229	458 000	67.4
51.3	253	506 000	74.7
56.9	284	568 000	83.5
62.9	319	638 000	93.8
69.7	366	732 000	108
75.4	359	718 000	106
83.2	392	784 000	115
103.6	490	980 000	144
124.5	602	1 200 000	177

Example of recovery potential from compressors.

3.4.3 Recovery methods

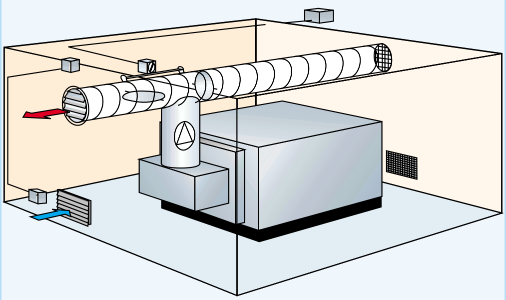
3.4.3.1 General

Energy recovery from compressed air installations does not always give heat when it is required and perhaps not in sufficient quantities. The quantity of recovered energy will vary if the compressor has a variable load. In order for recovery to be possible a corresponding energy requirement is needed, which is normally met through an ordinary system supply. Recovered energy is best utilised as additional energy to the ordinary system, so that the available energy is always utilised when the compressor is running.

3.4.3.2 Air cooled systems

Options for air cooled compressors, which give off a large hot air flow rate at a relatively low temperature, are direct building heating or heat exchanging to a preheating battery. The heated cooling air is then distributed using a fan.

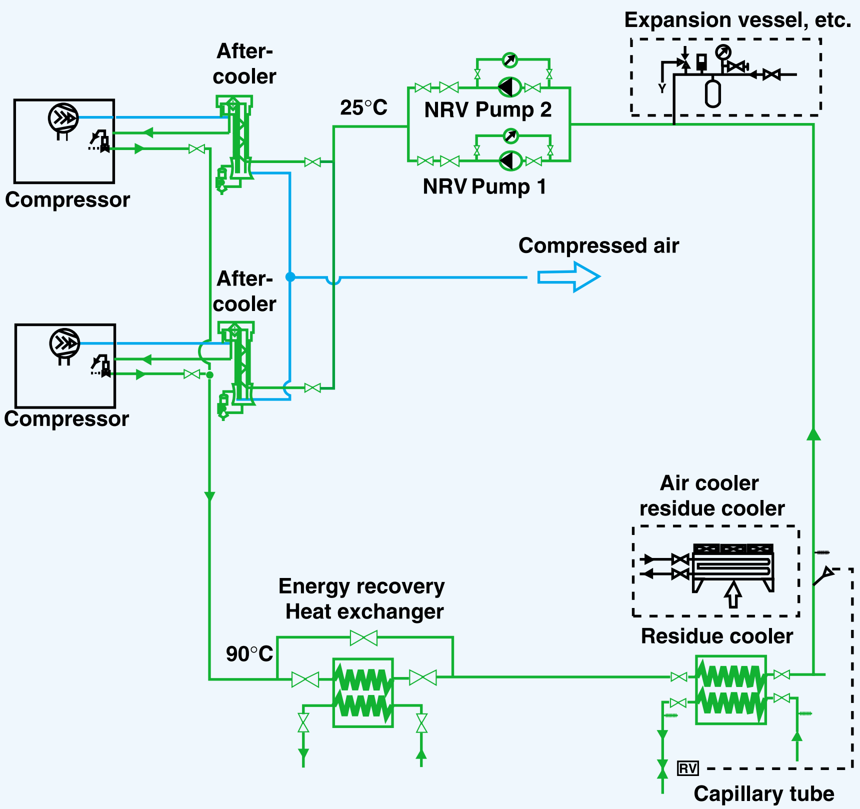
3:24



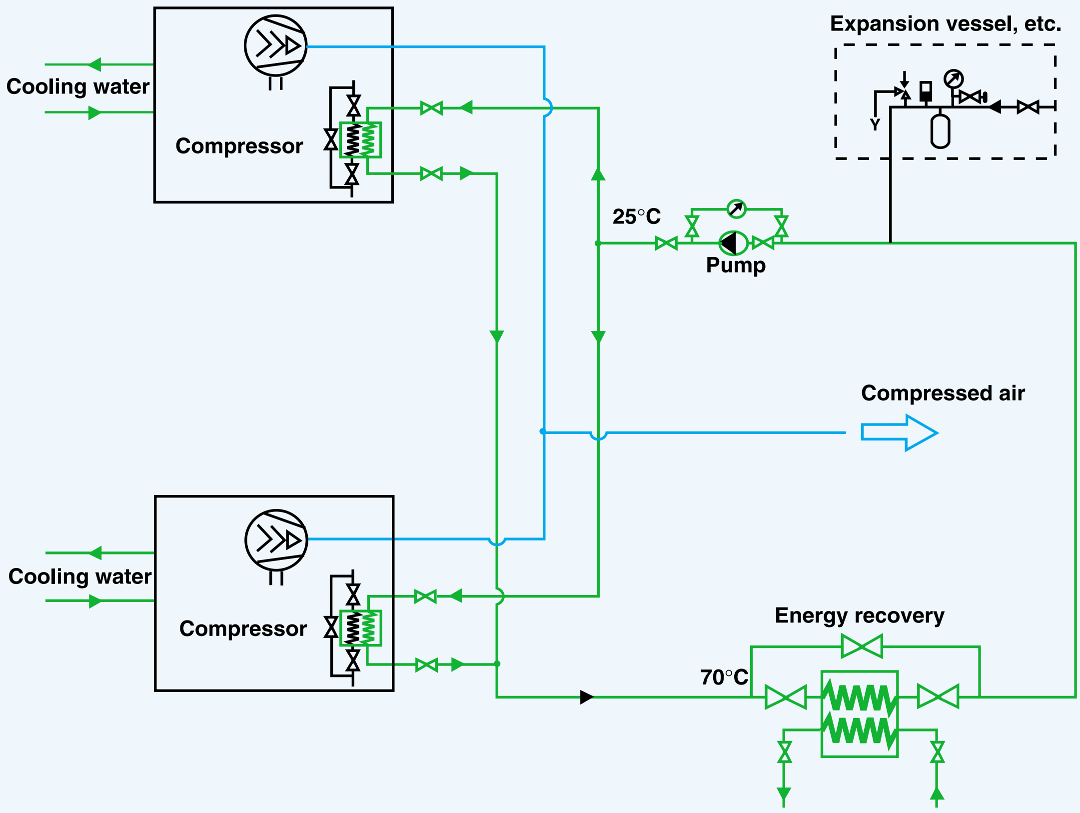
Energy recovery from an air cooled compressor.

When the buildings do not require additional heat, the hot air is led off into the atmosphere either automatically using

3:25



An example of water-borne energy recovery for an oil-free screw compressor.



Example of water-borne energy recovery for oil lubricated screw compressor. The residue cooler with regulation system is integrated in the compressor.

thermostat control or manually by controlling the air damper. A limiting factor is that the distance between the compressors and the building to be heated should be short, preferably it should be a question of an adjoining building. Furthermore, the possibility of recovery is limited to the colder parts of the year. Air-borne energy recovery is more common on small and medium sized compressors. Recovery results in small losses and requires little investment.

3.4.3.3 Water cooled system

On a water cooled compressor, the cooling water from the compressor with a temperature up to 90° can supplement a hot water flow. If hot water is used for washing, cleaning or showering a normal hot water boiler is still required. The energy recovered from the compressed air system provides an addition that reduces the load on the boiler, saves fuel and possibly can result in the use of a smaller boiler.

Prerequisites for energy recovery from compressed air compressors differs partly depending on the type of compressor. Oil-free compressors even in the standard design are easy to modify for energy recovery. This type of compressor in a hot water design reaches the water temperatures (90°C) required for efficient energy recovery. On oil lubricated compressors the oil, which takes part in compression, is the factor that limits the possibilities to reach higher cooling water temperatures.

In centrifugal compressors the temperature levels are lower an thereby, so is the degree of recovery. In addition the performance of the compressor is negatively affected by the increased water temperatures.

Water-borne energy recovery is best suited to compressors with an motor power over 10 kW. Water-borne recovery of energy brings about a more complex installation than air-borne energy recovery. The basic equipment consists of pumps, heat exchanger and regulation valves.

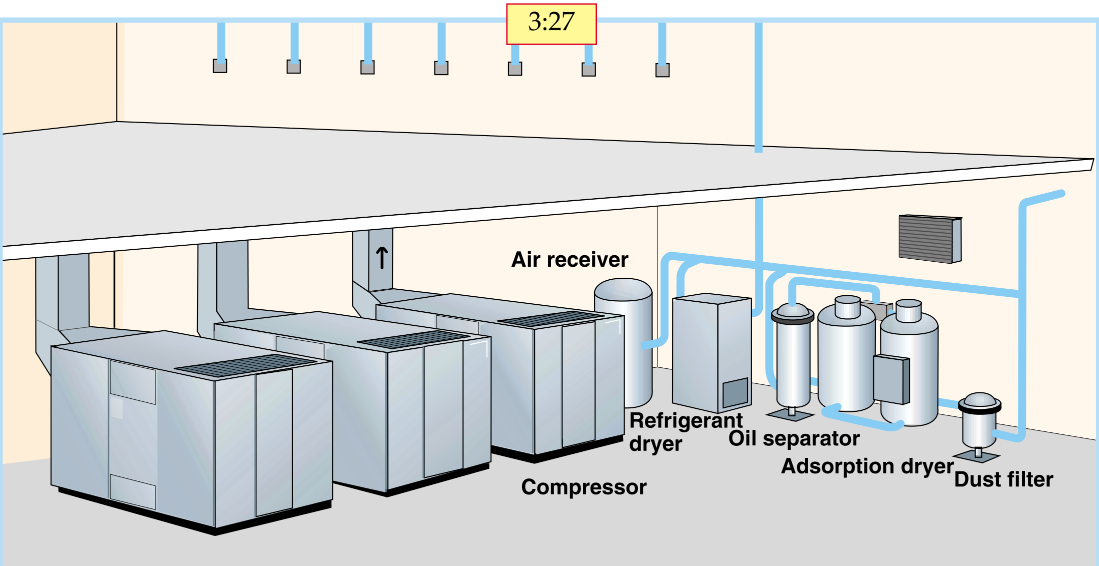
Heat can also be distributed to remote buildings using relatively small pipe dimensions (40-80 mm) without significant heat losses using water-borne energy recovery. The high initial temperature means that energy can be used to increase the temperature of the return water from a hot water boiler. Thereby the normal heating source can be periodically switched off and replaced by the compressor's waste heat. Waste heat from compressors in the process industry can also be used to increase the temperature of the process. It is fully possible even with the use of air cooled, oil lubricated screw compressors to arrange water-borne energy recovery. This requires a heat exchanger in the oil circuit, but the system gives lower temperature levels than with an oil-free compressor.

3.5 The compressor room

3.5.1 General

Not so long a go the acquisition of a compressor meant that you needed to buy the electric motor, starter equipment, aftercooler, intake filters, etc. You then needed to go through capacity and quality demands

3:27



Installation in a compressor room is easier today than before. The compressor plant is now a turnkey solution ready to install and connect to requisite auxiliary equipment.

with each supplier of the different components This was necessary to ensure that they would work well together with the compressor. Nowadays, the compressor and accessories are purchased in as a turnkey solution. A compressor package consists of a box frame, on which the compressor and accessories are mounted. All internal connections between the different parts are already made. The complete compressor package is enclosed in a sound reducing hood to reduce noise levels.

This has resulted in a significant simplification of the installation and you can be completely assured from the outset that the system will work. Irrespective of this, it is important to remember that the installation method and technology still have a significant influence on the compressor system's performance and reliability.

The main rule for an installation is first and foremost to arrange a separate compressor central. Experience says that centralisation is preferable, irrespective of the type of industry. This gives, among others, improved operating economy, a better designed compressed air system, service and user friendliness, protection against authorised access, good noise control and simpler possibilities for controlled ventilation.

Secondly, a demarcated area in a building used for other purposes can be used for the compressor installation. The risk for other problems should be observed with such an installation, for example, disturbances from noise, the compressor's ventilation requirements, physical risks and/or the risk of overheating, drainage for condensation, hazardous surroundings e.g. dust or inflammable substances, aggressive substances in the air, space requirements for future expansion and accessibility for service. However, installation in, e.g. a workshop or warehouse can facilitate the installations for energy recovery. If there are no facilities to install the compressor indoors it can also be placed outdoors under a roof. You must however bear in mind the risk of freezing in condensation pockets and discharges, rain and snow protection on the air intake, suction inlet and ventilation, demands of a solid and flat foundation, e.g. asphalt, concrete slab or a flattened bed of shingle, the risk of dust, inflammable or aggressive substances and unauthorised access protection.

3.5.2 Placement and design

The compressed air central should be placed to facilitate routing of the distribution system in large installations with long piping. It can be advantageous for service and maintenance to place the compressed air central close to auxiliary equipment such as pumps and fans; even a location close to the boiler room can be beneficial.

The building should have access to lifting equipment dimensioned to handle the heaviest components in the compressor installation, (usually the electric motor) and/or the possibility of using a fork lift truck. It should also have floor space for the installation of an extra compressor with future expansion.

In addition, the clearance height must be sufficient to allow the lift of an electric motor or the like if the need arises. The compressed air central should have a floor drain or other facilities to handle condensation from the compressor, aftercooler, air receiver, dryers, etc. The floor drain shall be implemented in accordance with municipal directives.

3.5.3 Foundation

Normally only a flat floor of sufficient bearing capacity is required to set-up the compressor plant. In most cases vibration dampening is integrated in the plant. It is usual with new installations to cast a plinth for each compressor package to allow the floor to be cleaned.

Large piston and centrifugal compressors can require a concrete slab foundation, which is anchored to the bedrock or on a solid soil base. The effects of externally produced vibration has been reduced to a minimum on advanced, complete compressor plants. In systems with centrifugal compressors it may be necessary to vibration dampen the compressor room's foundation.

3.5.4 Intake air

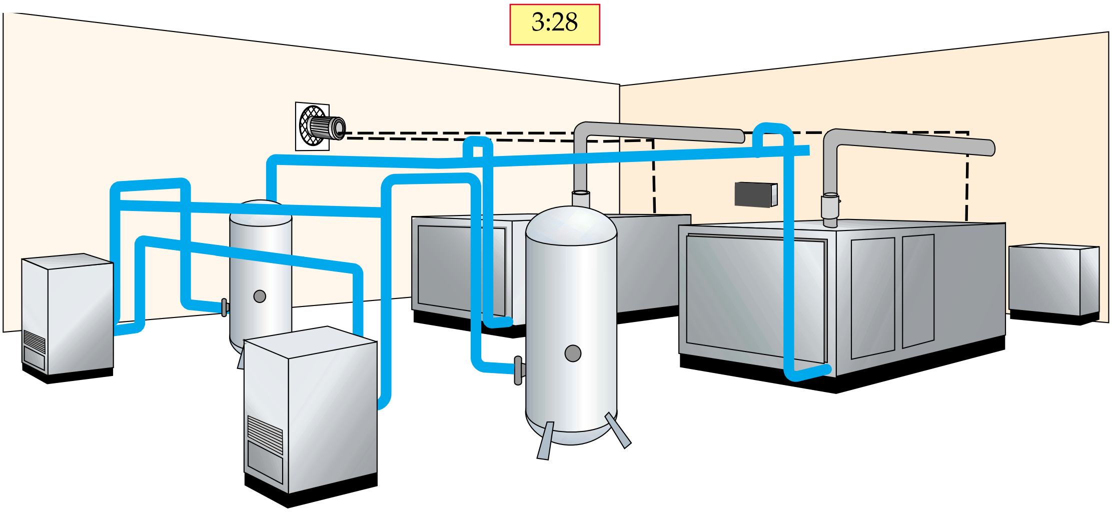
The compressor's intake air must be clean and free from solid and gaseous contamination. Particles of dirt that cause wear and corrosive gases can be particularly damaging.

The compressor's air intake normally takes place via the sound reducing hood, but can also be placed where the air is as clean as possible. Gas contamination such as vehicle exhaust fumes, can be fatal if mixed in air to be breathed. For example, hospital applications usually make special demands on the placement of the air intake. A prefilter (cyclone, panel or rotary band filter) should be used on installations where the surrounding air has a high dust concentration. In such cases the pressure drop caused by the prefilter must be observed, so that it does not exceed the maximum limits prescribed by the manufacturer.

It is also beneficial for the intake air to be cold. It can therefore be appropriate to route this via a separate pipe from the outside of the building to the compressor.

It is important that corrosion resistant pipes, fitted with mesh over the inlet

3:28



It is important that the compressor installation has a design that is service friendly and flexible to accommodate future expansion. The minimum area at service points in front of the machine's electrical cabinets should be 1200 mm.

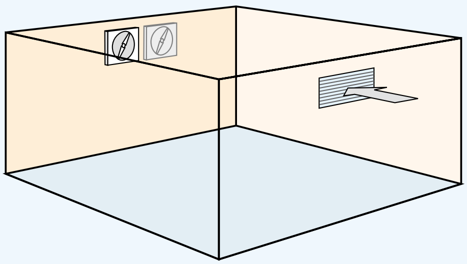
and designed so that there is no risk of drawing in snow or rain into the compressor, are used for this purpose. It is also important to use pipes of a sufficient large dimension to gain as low a pressure drop as possible.

The design of the inlet pipes on piston compressors is particularly critical. Pipe resonance caused by the compressor's cyclic pulsating frequency, can damage the compressor, cause vibration and affect the surroundings through low frequency noise.

3.5.5 Compressor room ventilation

Heat in the compressor room is generated from all compressors. This heat is led off through ventilating the compressor room. The quantity of ventilation air is determined by the size of the compressor and whether it is air or water cooled.

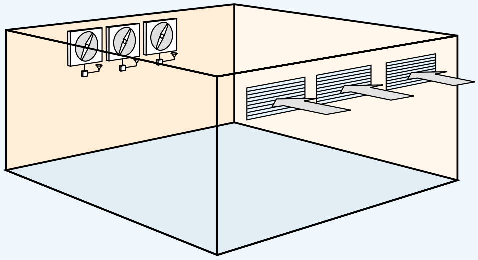
3:29



This is how a basic ventilation solution can be designed. The disadvantage is that ventilation is constant irrespective of the outer temperature. In addition difficulties can occur if two compressors are installed. The fans will be over specified if only one of the compressors is used. The problem can be solved by fitting the fans with speed controlled motors, which start via a multi-stage thermostat.

The ventilation air with air cooled compressors contains close to 100% of the energy consumed by the electric motor in the form of heat. The ventilation air with water cooled compressors contains down

3:30



This is how a system with several thermostat controlled fans, which together can handle the total ventilation requirement, can be designed. The thermostats on the individual fans are set for different ranges, which means the quantity of ventilation air can vary depending on the outer temperature and/or the number of compressors in use (as the thermostats will switch on the fans one after another depending on the temperature in the compressor room). Alternatively, the fans can be started via a multi-stage thermostat.

to 10% of the energy consumed by the electric motor. The heat must be removed to maintain the temperature in the compressor room at an acceptable level. The compressor manufacturer should provide detailed information regarding the requisite ventilation, but it can also be calculated according to the following:

$$P_V = \frac{Q_V}{1{,}25 \times \Delta T}$$

P_V = requisite quantity of ventilation air (m^3/s)

Q_V = heat flow (kW)

ΔT = permitted temperature rise (°C)

A better way to deal with the problem is to recover the energy and use it in the enterprise.

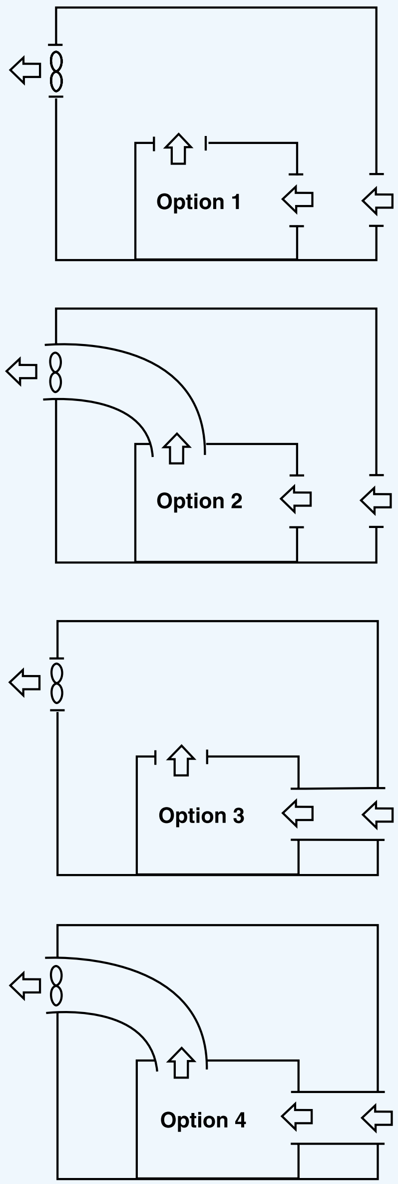
Ventilation air should be taken from outdoors, preferably without long ducting. The inlet for the ventilation air should be placed on a north facing wall if possible, or in another place in the shade so that the intake air is as cool as possible during the summer. A grille should be fitted to the outside of the intake and an air stream operated damper on the inside to prevent foreign objects from entering and cold draughts.

Furthermore, the intake should be placed as low as possible, yet avoiding the risk of being covered with snow during the winter. Even the possible risk of dust and explosive or corrosive substances entering the compressor room must be taken into consideration.

The ventilation fan/fans should be placed high up on one of the compressor room's end walls, while the air intake is placed on the opposite wall. The air velocity at the intake should not exceed 4 m/s.

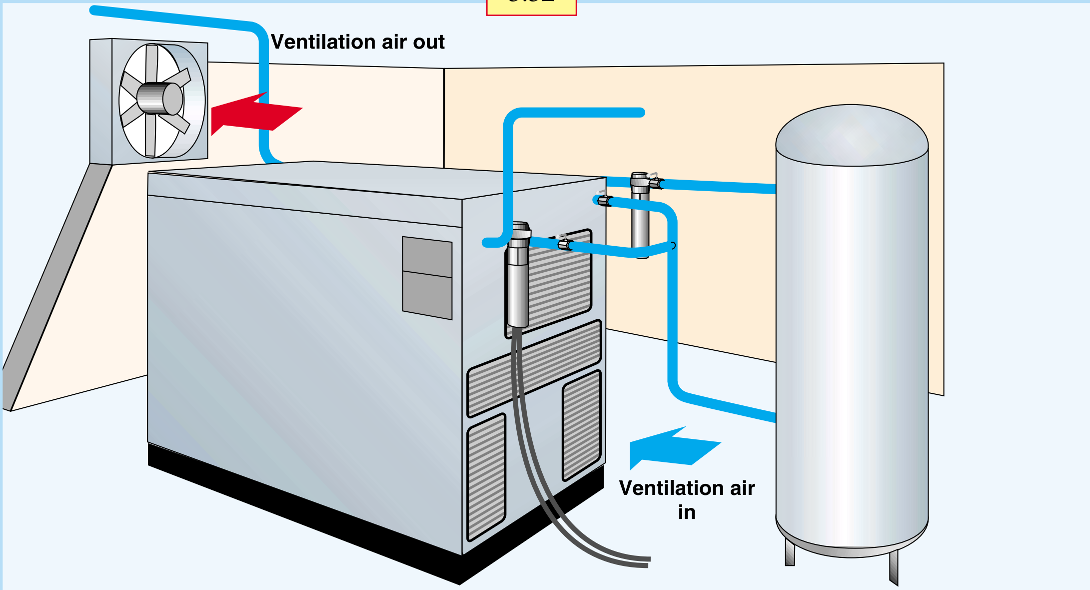
Thermostat controlled fans are the most appropriate. These must be dimensioned to handle the pressure drop in the ducting, outer wall grille, air stream operated damper, etc. The quantity of ventilation air must be sufficient to limit the temperature increase in the room to 7–10°C. The possibility in using water cooled compressors should be considered, if there is a problem in arranging sufficient ventilation in the room.

3.31



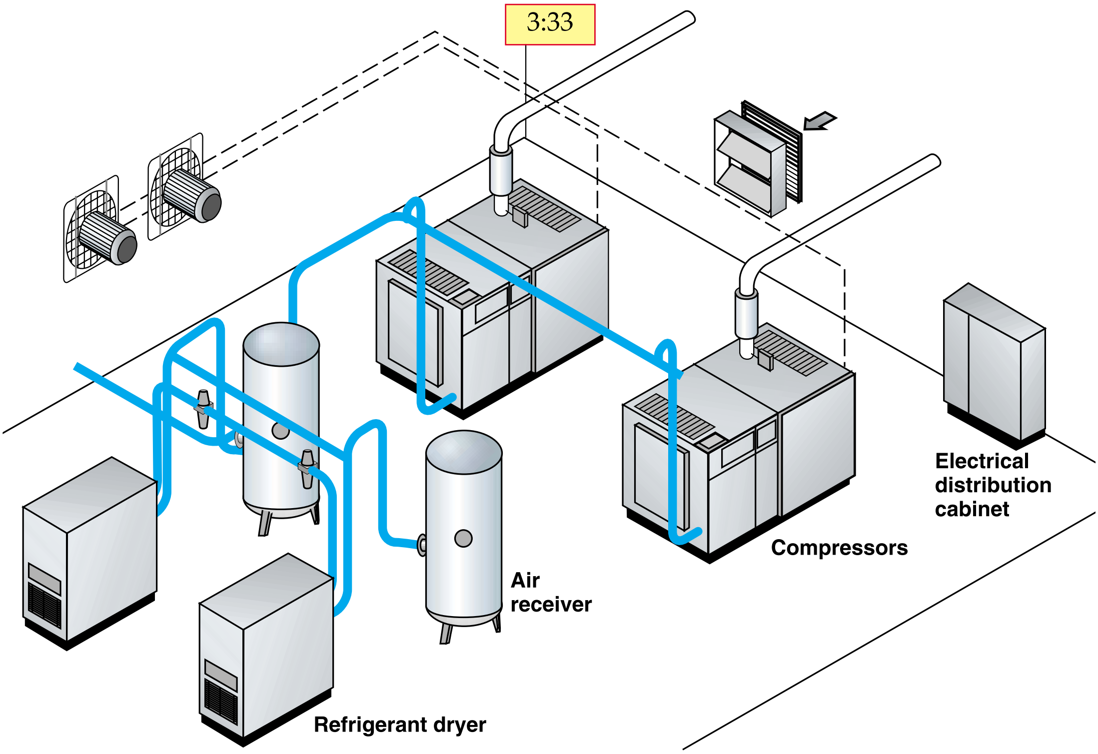
Examples of different ventilation solutions

3:32



A small compressor installation in practice. The pipe for draining condensation must not open out under the water surface in the floor drain. An oil separator must be installed, if there is a risk that the waste water contains oil.

3:33



Example of a hospital installation with closed supply on the suction side and 100% reserve system (double independent system).

3.6 The compressed air network's structure

3.6.1 General

Three demands are placed on a distribution system to provide reliable operations and good economy: a low pressure drop between the compressor and point of consumption, a minimum of leakage, and the best possible condensation separation in the system if a compressed air dryer is not installed.

This primarily applies to the main pipes. The cost of installing larger pipe dimensions as well as fittings than that initially demanded is low compared with the cost of rebuilding the system at a later date. The air line network's routing, design and dimensioning are important for the efficiency of the installation, reliability and cost. Sometimes a large pressure drop in the pipeline is compensated by increasing the working pressure of the compressor from, e.g. 7 bar(e) to 8 bar(e). This gives inferior compressed air economy. When the compressed air consumption falls, the pressure drop also falls and the pressure at the point of consumption rises above the permitted level.

Fixed compressed air installations should be dimensioned so that the pressure drop in the pipes does not exceed 0.1 bar between the compressor and the furthest point of consumption. Added to this is the pressure drop in hoses, hose couplings and other fittings. It is particularly important how these components are dimensioned, as the greatest pressure drop frequently occurs at such connections.

The longest permitted length in the pipe network for a specific pressure drop can be calculated from the following empirical relation:

$$l = \frac{\Delta p \times d^5 \cdot p}{450 \times Q_c^{1,85}}$$

l = overall pipe length (m)
Δp = largest permitted pressure drop in the network (bar)
p = absolute inlet pressure (bar)
Q_c = flow, FAD (l/s)
d = internal pipe dimension (mm)

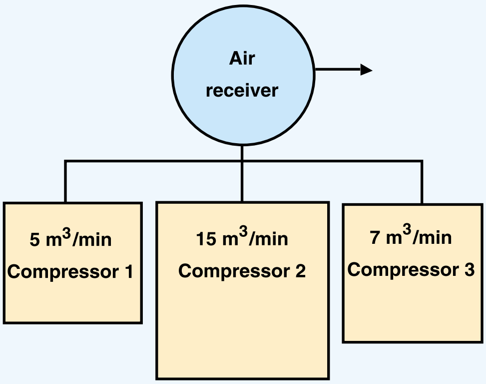
In general the best solution is to design a pipe system as a ring line around the area where air consumption will take place. Branch pipes are then taken off of the main pipeline to the points of consumption. This gives an even compressed air supply, despite heavy intermittent usage as the air is led to the actual point of consumption from two directions.

This system should be used for all installations, even if some points of consumption are at a great distance from the compressor installation. A separate main pipe is then routed to these areas.

3.6.1.1 Air receiver

One or more air receivers are included in each compressor installation. The size is adapted, e.g. according to the compressor capacity, regulation system and the consumer's air requirement. The air receiver forms a storage area for the compressed air, balances pulsation from the compressor and cools the air and collects condensation. Accordingly, the air receiver must be fitted with a drainage device.

3:34



The air receiver is always dimensioned according to the capacity of the largest compressor when a system comprises of several compressors.

The following relation applies when dimensioning the receiver's volume. Note that the relation only applies for compressors with offloading/loading regulation.

$$V = \frac{0{,}25 \times Q_c \times p_1 \times T_0}{f_{max} \times (p_U - p_L) \times T_1}$$

V = air receiver's volume (l)
Q_c = Compressor's capacity (l/s) FAD
p_1 = Compressor's intake pressure (bar(a))
T_1 = Compressor's maximum intake temperature (K)
T_0 = Compressed air temperature in receiver (K)
(pu-pL) = set pressure difference between the loaded and off-loaded
f_{max} = maximum frequency = 1 cycle/30 seconds (applies to Atlas Copco compressors)

Simplified formula that applies with an ambient relation 1 bar(a) and approx. 20°C and 30 s cycle time.

$$V = \frac{Q}{8 \times \Delta p}$$

V = air receiver's volume (m³)
Q = capacity of the largest compressor (m³/min)
Δp = desired pressure difference (bar)

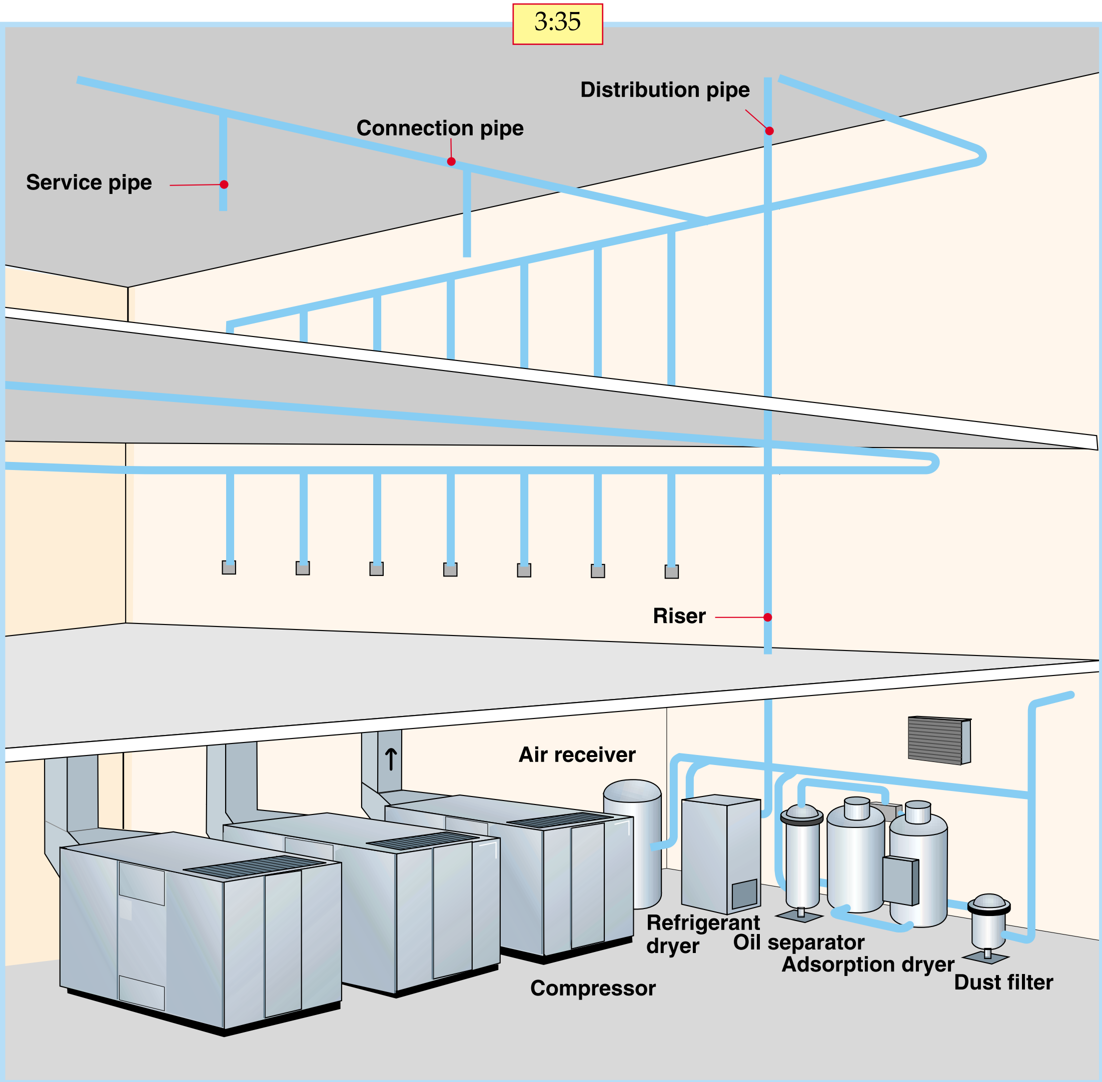
When air is required in large quantities during short periods it is not economic to dimension the compressor or pipe network according to this. A separate air receiver is then placed close to the consumer and is dimensioned according to the maximum air output.

In more extreme cases, a smaller high pressure compressor is used together with a large receiver to meet short-term, large air requirements between long intervals. The compressor will then be dimensioned for the mean consumption. The following relation applies for such a receiver:

$$V = \frac{Q \times t}{p_1 - p_2} = \frac{L}{p_1 - p_2}$$

V = air receiver's volume (l)
Q = air flow during the emptying phase (l/s)
t = length of the emptying phase (s)
P_1 = normal working pressure in the network (bar)
P_2 = minimum pressure for the consumer's function (bar)
L = filling phase's air requirement (1/work cycle)

The formula does not take into consideration the fact that the compressor can supply air during the emptying phase. A common application is the start of large ship engines, where the receiver's filling pressure is 30 bar.



3.6.2 Design of the compressed air network

In smaller installations the same pipe can serve as the riser and distribution pipe. The starting point when designing and dimensioning a compressed air network is an equipment list with all the compressed air consumers, and a drawing indicating their placement. The consumers are grouped in logical units and are supplied via the same distribution pipe. The distribution pipe is fed in turn by risers from the compressor central. A larger compressed air network can be divided into four main parts: Risers, distribution pipes, service pipes and compressed air fittings. The risers transport the compressed air from the compressor central to the consumption area.

Distribution pipes divide the air across the distribution area. Service pipes feed the air from the distribution pipes to the workplaces. The compressed air fittings are the connections between the service pipe and the compressed air consumer.

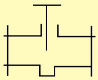
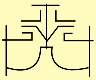
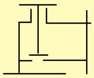
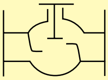
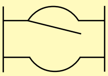
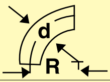
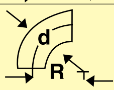
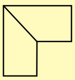
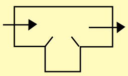
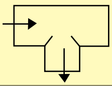
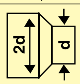
3.6.3 Dimensioning the compressed air network

The pressure obtained immediately after the compressor can generally never be utilised fully, accordingly, you must calculate that the distribution of compressed air claims some losses, primarily friction losses in the pipes. In addition, throttling and changes in the direction of flow occur in valves and pipe bends. Losses, which are converted to heat, result in a pressure drop that for a straight pipe can be calculated with the relation:

$$\Delta p = 450 \times \frac{q_v^{1,85} \times l}{d^5 \times p}$$

Δp = pressure drop (bar)
qv = air flow, free air (l/s)
d = internal pipe diameter (mm)
l = length of the pipe bar(a)
p = absolute initial pressure

3:36

Equivalent length in metres												
Component		Inner pipe diameter in mm (d)										
		25	40	50	80	100	125	200	250	250	300	400
Ball valve (full flow)		0.3 5	0.5 8	0.6 10	1.0 16	1.3 20	1.6 25	1.9 30	2.6 40	3.2 50	3.9 60	5.2 80
Diaphragm valve fully open		1.5	2.5	3.0	4.5	6	8	10	-	-	-	-
Angle valve fully open		4	6	7	12	15	18	22	30	36	-	-
Poppet valve		7.5	12	15	24	30	38	45	60	-	-	-
Flap check valve		2.0	3.2	4.0	6.4	8.0	10	12	16	20	24	32
Elbow R = 2d		0.3	0.5	0.6	1.0	1.2	1.5	1.8	2.4	3.0	3.6	4.8
Elbow R = d		0.4	0.6	0.8	1.3	1.6	2.0	2.4	3.2	4.0	4.8	6.4
90° angle		1.5	2.4	3.0	4.5	6.0	7.5	9	12	15	18	24
Tee through-flow		0.3	0.4	1.0	1.6	2.0	2.5	3	4	5	6	8
Tee side-flow		1.5	2.4	3.0	4.8	6.0	7.5	9	12	15	18	24
Reducing nipple		0.5	0.7	1.0	2.0	2.5	3.1	3.6	4.8	6.0	7.2	9.6

Some fittings and their influence on losses in pipes of a different diameter. The losses are recalculated to a corresponding increase in the length of the pipe network (m).

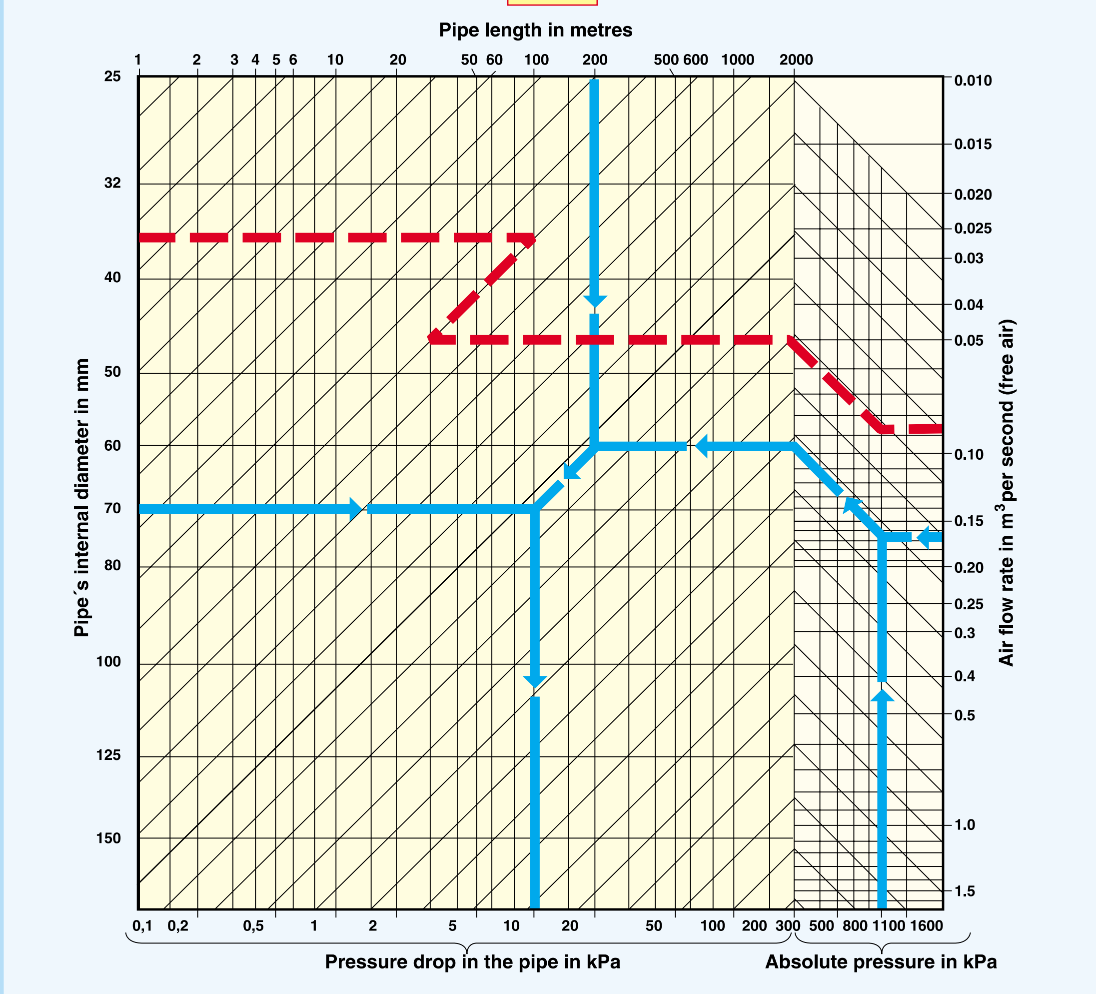
When calculating different parts of the compressed air network the following values can be used for the permitted pressure drop:

Pressure drop across service pipes	**0.03 bar**
Pressure drop across distribution pipes	**0.05 bar**
Pressure drop across risers	**0.02 bar**
Total pressure drop across the fixed pipe installation	**0.10 bar**

The requisite pipe lengths for the different parts of the network (risers, distribution and service pipes) are estimated. A scale drawing of the probable network plan is a suitable basis. The length of the pipe is corrected through the addition of equivalent pipe lengths for valves, pipe bends, unions, etc as set out in figure 3:36. When calculating the pipe diameter a nomogram, as set out in figure 3:37, can be used to find the most appropriate pipe diameter as an alternative to the formula (page 99). The flow, pressure, permitted pressure drop and the pipe length must be known to make a calculation. Standard pipe of the closest, greater diameter is then selected for the installation.

The equivalent pipe lengths for all

3:37



the parts of the installation are calculated by using a list of fittings and pipe components as well as the flow resistance expressed in pipe length. These "extra" pipe lengths are added to the starting pipe length. The network's selected dimensions are then recalculated to ensure that the pressure drop will not be too great. The individual sections (service pipe, distribution pipe and risers) should be calculated separately on large installations.

3.6.4 Flow measurement

Strategically placed flow meters permit internal debiting and economic allocation of compressed air utilisation within the company. Compressed air is a production media that should be a part production costs for individual departments within the company. With such a viewpoint it becomes interesting for all concerned to try to reduce consumption within the different departments.

The modern flow meters available on the market can give everything from numerical values for manual reading, to measurement data directly to a computer or debiting module.

The flow meters are generally mounted close to shutoff valves. Ring measurement makes particular demands as the meter needs to be able to measure both forwards and backwards.

3.7 Portable compressors

3.7.1 General

Today, virtually all portable compressors consist of a diesel engine powered, oil injected, screw compressor. Oil-free compressors only occur, for example, with service work in the process industry.

3.7.2 Noise and gaseous emissions

Modern designs of diesel powered compressors have a very low noise level according to applicable EU standards (ISO 84/536/EC) and can therefore be used in populated areas and close to hospitals, etc.

During the passed years fuel economy has been improved dramatically through efficient screw elements and more effective diesel engines. This is especially valuable, e.g. for well drilling, where the compressor works intensively under a long period. At the current time, there are engines with exhaust gas emissions that comply with the stringent demands set out in EURO-1. Contractors carrying out work in large towns must today (1998) use machinery that complies with this standard.

3:38

Pressure range	Pressure (bar)	Application area
Low	⩾7	Contract work
Medium	10 - 12	Blasting, ground work
High	20⩽	Water and energy well drilling, geotechnical investigations

Portable compressors are chiefly available in three different pressure ranges.

3.7.3 Pressure range

Modern portable compressors have a good overall economy through high operating reliability, good service characteristics, compact dimensions and a low total weight. They have a chassis normally designed for a transport speed of 30km/h or 80km/h. As for stationary compressors there is auxiliary equipment such as after-coolers, different filter packs (dust filters, carbon, etc.), after heaters and lubricating oil systems available. They can also be equipped with cold start equipment and a generator 230V/400V. There are portable, diesel powered generators built in a similar way to portable compressors for greater power requirements. The power classes start from 10 kVA and upwards.

3.8 Electrical installation

3.8.1 General

To dimension and install a compressor installation requires knowledge on how component parts affect each other and which regulations and provisions apply.

Here follows an overview of the parameters that should be especially considered to obtain a compressor installation that functions satisfactorily with regard to the electrical installation.

3.8.2 Motors

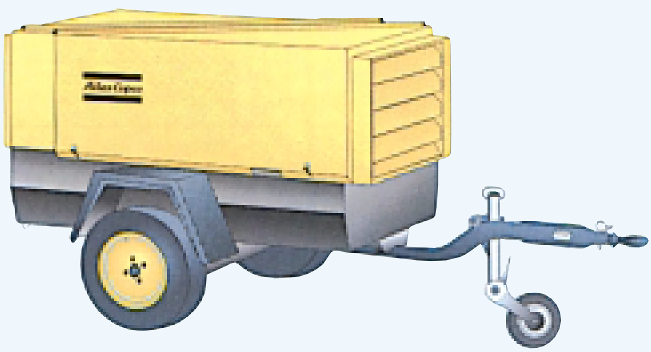
Short-circuited, three phase induction motors are used for compressor operations. Low voltage motors are generally used up to 450 kW and there above high voltage is the best option.

The motor's protection class is regulated by standards. The dust and water spray resistant design (IP54) is preferred to open motors (IP23), which require regular dismantling and cleaning. In other cases, dust deposits in the machine will eventually cause overheating, resulting in a reduced service life.

The motor, usually fan cooled, is intended to work at a maximum ambient temperature of 40°C. At high temperatures the output must be reduced. The motor is normally flange mounted and directly connected to the compressor. The speed is adapted to the compressor, however, in practice 2 pole or 4 pole motors with a speed of 3000 rpm respective 1500 rpm are solely used.

The rated output of the motor is also determined by the compressor and should be as

3:39



Example of a portable compressor

close to the compressor's requirement as possible.

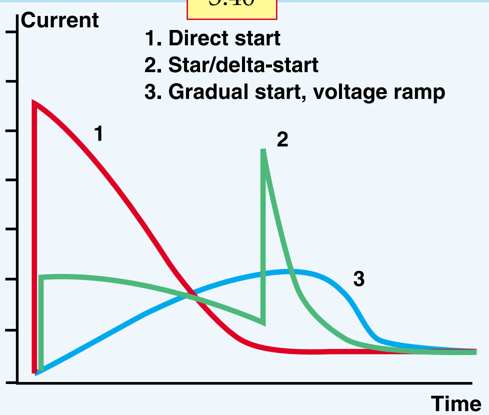
A too large motor is more expensive to buy, requires an unnecessarily high starting current, requires larger fuses, has low output factor and somewhat inferior efficiency. A too small motor becomes overloaded and consequently a risk for breakdown.

The starting method should also be included as a parameter when selecting a motor. The motor is only started with a quarter of its rated torque with star/-delta–start, which is why a comparison between the motor's and the compressor's torque curves can be justified to ensure that the compressor starts correctly.
See 3.8.3.

3.8.3 Starting methods

The most common starting methods are direct start, star/delta–start and gradual start. Direct start is easy and only requires a contactor and overload protection. The disadvantage is the high starting current, 6–10 times the motor's rated current and sometimes the high starting torque, which can, for example, damage shafts and couplings.

3:40



Connection current/time at different starting methods

The star/delta–start is used to limit the starting current. The starter consists of three contactors, overload protection and a timer. The motor is started with the star connection and after a set time (when the speed has reached 90% of the rated speed) the timer switches the contactors so that the motor is delta connected, which is the operating mode. See 1.6.5.7.

The star/delta–start reduces the starting current to approximately 1/3 compared with a direct start, however, the starting torque falls at the same time to a quarter. The relatively low starting torque means the motor's load should be low during the starting phase, so that the motor virtually reaches its rated speed before switching to the delta connection. If the speed is too low, a current/torque peak as great as with direct start, will occur during switching to the delta connection.

Gradual start, which can be an alternative start method to star/delta–start is a starter built up of semiconductors (thyristors) instead of mechanical contactors. The thyristors are controlled according to a time ramp, so that an equal rising current feeds the motor. The start is gradual and the starting current is limited to approx. three times the rated current.

The starters for direct start and star/delta-start are in most cases integrated in the compressor. It can be motivated with large compressor plant to place the units separately in the switchgear, due to space requirements, heat development and access for service.

A starter for gradual start is usually set-up separately next to the compressor. High voltage fed compressors always have their start equipment in the switchgear.

3.8.4 Control voltage

Normally no separate control voltage is connected to the compressor, as most compressors are fitted with an integrated control transformer. The transformer's primary side is connected to the compressor's power supply. This arrangement gives more reliable operations. In the event of disturbances in the power supply the compressor will be stopped immediately and blocked for restart.

This function, with one internally fed control voltage, should be copied in those cases where the starter is placed away from the compressor.

3.8.5 Short-circuit protection

Short-circuit protection, which is placed on one of the cables' starting points, can be made up of fuses or a circuit-breaker. Irrespective of which of the solutions you select it will give, if correctly matched, good protection.

Both methods have advantages and disadvantages. Fuses are well-known and work better than a circuit-breaker with large short-circuit currents, but they do not make a fully isolating break and have a long tripping time with small fault currents. A circuit-breaker breaks fully isolating and rapidly even with small fault currents, but demands more work during the planning stage compared to fuses. Dimensioning of the short-circuit protection is based on the expected load, but also on the limitations of the starter unit.

With regard to the starter's short-circuit protection see the standard according to IEC (International Electrotechnical Commission) 947-4-1 Type 1 & Type 2. How a short-circuit will affect the starter is determined by which of the options, Type 1 or Type 2, is selected.

Type 1: "damage to contactors and overload relays can occur. The replacement of components may be necessary".

Type 2: "Damage does occur to the overload relays. Light welding of the contactors is permitted. It shall be possible using basic measures to reset the starter in the operating mode."

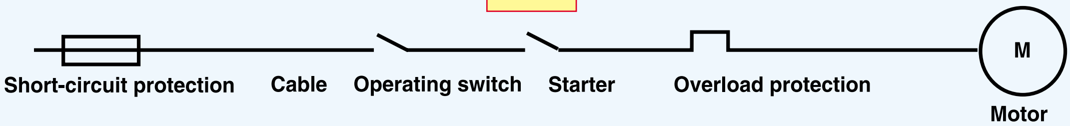
3.8.6 Cables

Cables shall, according to the provisions "be dimensioned so that during normal operations they do not accept hazardous temperatures and that they shall not be damaged thermally or mechanically with a short-circuit". The dimensioning and selection of cables is based on the load, permitted voltage drop, routing method (on a rack, on a wall, etc.) and the ambient temperature. Fuses can be used, for example, to protect the cables and can make up both a short-circuit protection and an overload protection. For motor operations a short-circuit protection is used (for example, fuses) and a separate overload protection (usually the motor protection included in the starter).

The overload protection protects the motor and motor cables by tripping and breaking the starter, when the load current exceeds the pre-set value. The short-circuit protection protects the starter, overload protection and the cables. How cables are dimensioned taking the load into consideration is set out in IEC 364 5 523 (SS 4241424).

There is a further parameter to bear in mind when dimensioning cables and the short-circuit protection, namely the "tripping condition". This condition means the installation shall be designed so,

3:41



A basic system for the connection of an electric motor to the mains supply.

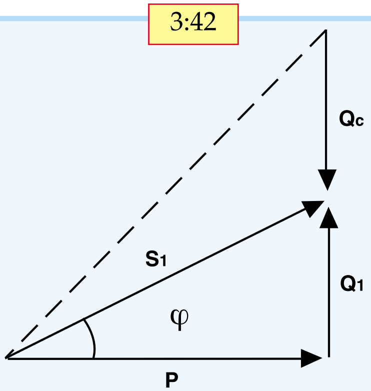
a short-circuit anywhere in the installation shall result in quick and safe breaking. Whether the condition is met is determined by, among others, the short-circuit protection, the length and cross-section of the cable.

3.8.7 Phase compensation

The electric motor does not only consume active power, which can be converted to mechanical work, but also reactive power, which is needed for the motor's magnetisation. The reactive power loads the cables and transformer. The relation between the active and reactive power is determined by the power factor, cosφ. This usually is between 0.7 and 0.9 where the lower value refers to small motors.

The power factor can be raised to virtually 1 by generating the reactive power directly by the machine using a capacitor. This reduces the need of drawing the reactive power from the mains. The motive for phase compensation can be that the power supplier may charge for drawing reactive power over a predetermined level and that heavily loaded transformers and cables need to be off-loaded.

3:42



Reactive power Qc is supplied to increase the motor's power factor (cosφ) to close to 1.

3.9 Sound

3.9.1 General

Noise is an energy form that propagates in a room as longitudinal waves through the air, which is an elastic medium. The wave movement causes changes in pressure, which can be registered by a pressure sensitive instrument, for example a microphone. The microphone is therefore one of the main parts in all equipment for sound measurement.

To measure the sound power in the SI unit Watt is difficult, due to the range covered by the sounds that surround us. Within acoustics they speak of levels instead, and measure sound in relation to a reference point. Measurement becomes manageable by apply a logarithm to the relation. The formula is:

$$L_W = 10 \times \log_{10} \times W / W_0$$

L_W = sound power level (dB)
W = actual sound power (W)
W_0 = reference power,
usually 10–12 (W)

3.9.2 Sound pressure

The sound pressure level is a measurement of the sound's intensity. The relation is:

$$L_p = 20 \times \log_{10} \times p/p_0$$

L_p = sound pressure level (dB)
p = actual sound pressure (bar)
p_o = reference sound pressure, usually 0.0002×10^{-6} (bar)

The sound pressure level always refers to a specific distance to the power source, e.g. a machine. For a stationary compressor the distance is 1 metre and for a portable compressor the distance is 7 metres (according to CAGI Pneurop).

Information about the sound pressure level must always be supplemented with a room constant for the room where the measurement was made. Otherwise the room is assumed to be limitless, i.e. an open field. In a limitless room there are no walls that can reflect the sound waves, which would affect the measurement.

3.9.3 Absorption

When sound waves come into contact with a surface, some of the waves are reflected and some absorbed into the material of which it consists. The sound pressure at a certain moment therefore always consists partly of a sound that the sound source generates, partly of sound that is reflected from surrounding surfaces (after one or more reflections).

How effectively a surface can absorb sound depends on the material it is made up of and is usually stated as an absorption factor (between 0 and 1).

3.9.4 Room constant

A room constant is calculated for a room with several surfaces, walls and other surfaces, which depends on the different surfaces' absorption characteristics. The relation is:

$$K = \frac{A \times \bar{\alpha}}{1 - \bar{\alpha}}$$

$$\bar{\alpha} = \frac{\textit{total absorption}}{\textit{total area}} = \frac{A_1 \times \alpha_1 + A_2 \times \alpha_2 +}{A_1 + A_2 +}$$

K = room constant
$\bar{\alpha}$ = average absorption factor for the room (m^2)
A = total room area (m^2)
A_1, A_2, etc. are the parts of the room surface that have absorption factors α_1, α_2, etc.

3.9.5 Reverberation

The reverberation time is defined as the time it takes for the average sound pressure to decrease by 60 dB once the sound source has become silent. The average or equivalent absorption factor for the room is calculated as:

$$\bar{\alpha} = \frac{0{,}163 \times V}{T}$$

V= volume of the room (m^3)
T = reverberation time (s)

The room constant is obtained if this expression in put in relation to:

$$K = \frac{A \times \bar{\alpha}}{1 - \bar{\alpha}}$$

A = total room area (m^2)

3.9.6 Relation between sound power and sound pressure

If sound is sent out from a point sound source in a room without reflecting surfaces, the sound is distributed equally in all directions and the measured intensity will therefore be the same at all points at the same distance from the sound source. Accordingly, the intensity is constant at all points on a spherical surface around the sound source.

From this you can derive that the sound level falls by 6dB for each doubling of the distance to the sound source. However, this does not apply if the room has hard, reflective walls. You must then take the sound reflected by the walls into consideration. If you then introduce a direction factor the relation becomes:

$$L_p = L_W + 10\log\frac{Q}{4\pi r^2}$$

L_p = sound pressure level (dB)
L_W= sound power level (dB)
Q = direction factor (m^2)
r = distance to the sound source

For Q the empirical values apply (for other positions of the sound source the value of Q must be estimated):

Q = 1	**if the sound source is suspended in the middle of a large room.**
Q = 2	**if the sound source is placed on a hard, reflective floor, close to the center of a wall or ceiling.**
Q = 4	**if the sound source is placed close to the transition wall-floor or wall-ceiling.**
Q = 8	**if the sound source is placed in a corner close to the intersection of three surfaces.**

If the sound source is placed in a room where its border surfaces do not absorb all the sound, the sound pressure level will increase due to the reverberation effect. This addition is inversed proportionally to the room constant:

$$L_p = L_w + 10\log\left[\frac{Q}{4\pi r^2} + \frac{4}{K}\right]$$

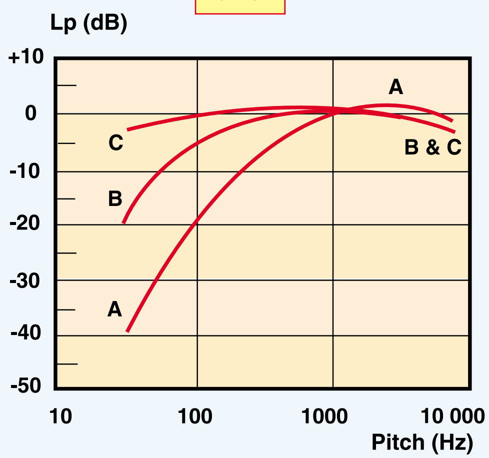
If this relation was drawn as a serie of curves it would show that in the proximity of the power source the sound pressure level drops by 6 dB for each doubling of the distance. However, at greater distances from the power source the sound power level is dominated by the reflected sound and thereby there is no decrease at all with increased distance.

The machines, which transmit sound through their bodies or frames, do not behave as point sources if the listener is at a distance greater than 2–3 times the machine's greatest dimension from its centre.

3.9.7 Sound measurements

The human ear distinguishes sound at different frequencies with different clarity. Low or very high frequency sounds must be stronger than sounds around 1000–2000 Hz to be perceived as equally strong.

3:43



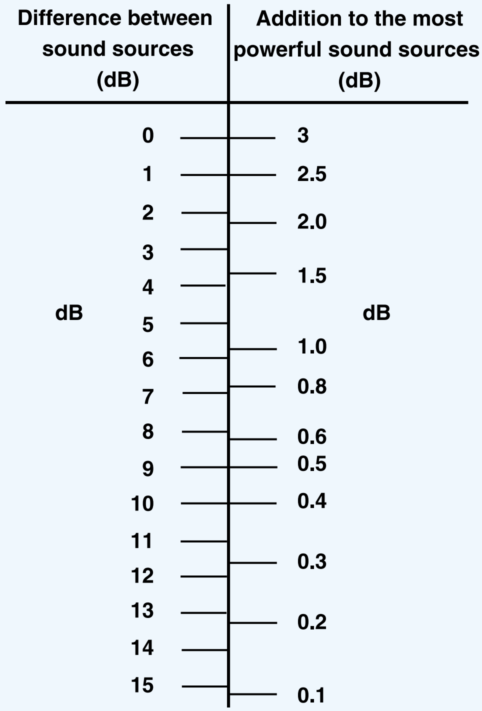
This is how the frequency curves appear for the different filters used to weight sound levels when measuring sound.

Different filters that adjust the measured levels at low and high frequencies are used to emulate the human ear's ability to hear sounds. When measuring noise an A-filter is usually used and the sound is measured as dB(A).

3.9.8 Interaction of several sound sources

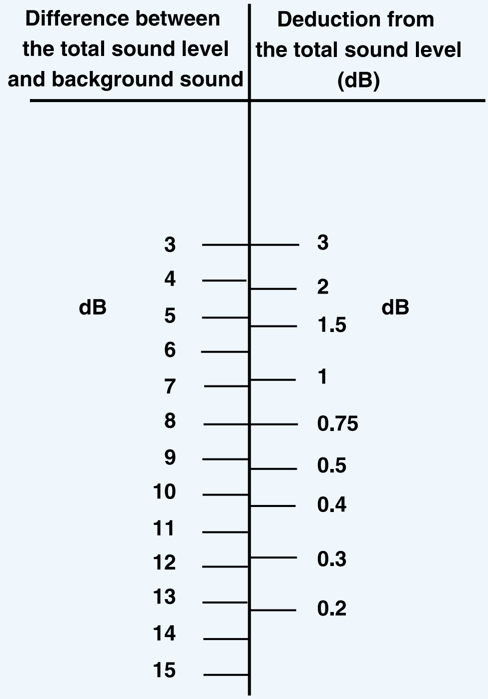
When there is more than one sound source in a room the sound pressure increases. However, as the sound pressure level is defined logarithmically you can not add the sound pressure levels algebraically. When more than two sound sources are active, you start by adding two and thereafter the next is added to the sum of the first and so on. As a mnemonic rule, when two sound sources with the same levels shall be added the result is an increase by 3 dB. When ten sound sources with the same levels shall be added the increase is 10 dB.

3:44



The nomogram defines how many dB shall be added to the most powerful sound source when the power of two sound sources shall be added.

3:45



The nomogram defines how many dB shall be deducted from the total sound level at different background sound levels in order to estimate the total noise level.

Background sound is a special case. It is treated as a separate sound source and the value is deducted from the total of the other sound sources in order to give these special treatment.

3.9.9 Sound reduction

There are five different ways to reduce sound. Sound insulation, sound absorption, vibration insulation, vibration dampening and dampening of the sound source. Sound insulation involves an acoustic barrier being placed between the sound source and the receiver.

This means that only a part of the sound can be insulated, depending on the area of the barrier and the insulation characteristics. A heavier barrier is more effective than a lighter barrier.

Sound absorption involves the sound source being surrounded by light, porous absorbents attached to a barrier. Thicker absorbents are more effective than thinner absorbents and typical densities are approx. 30 kg/m^3 for polyurethane foam respective approx. 150 kg/m^3 for mineral wool. Vibration insulation is used to prevent the transfer of vibrations from one part of a structure to another. A common problem is the transfer of vibrations from a built-in machine to the surrounding barrier or down to the floor. Steel springs, cork, plastic, and rubber and examples of material used for vibration insulation. The choice of material and dimensioning is determined by the frequency of the vibration and demands of stability on the machine set-up.

Vibration dampening involves a structure being fitted with a dampening external surface of an elastic material with a high hysteresis factor. When the dampening surface is sufficiently thick, a wall, for example, is effectively prevented from vibrating and thus starting to emit sound. Dampening of a sound source gives small results, yet a good exchange in relation to the cost. In this way a reduction in the machine's total sound level by approx. 5 dB can be achieved, while integration can mean a reduction by approx. 15–25 dB.

3.9.10 Noise with compressor installations

A compressor's noise level is measured on a machine in free field. When it is installed in a room the noise level is affected by the properties of the room. The size of the room, material in the walls and ceiling as the presence of other equipment (and its possible noise level) in the room are all significant.

Furthermore, the positioning of the compressor in the room also affects the noise level, precisely as the set-up and connection of pipes and the like. Sound radiating from compressed air pipes are frequently a more problematic noise source than the noise from the compressor and its power source. It can be a question of vibration transferred mechanically to the pipe, often in combination with vibration transferred through the compressed air. It is therefore important to fit vibration insulators and even enclose part of or the entire pipe system with a combination of sound reducing material and a sealed barrier.

3.10 Standards, laws and provisions

3.10.1 General

In the compressed air sector, as in many other sectors, there are regulations that apply. It can be demands that are defined in laws and provisions as well as optional regulations, as in national and international standards. Sometimes the regulations in standards are also binding, for example, when they come into force through legislation. If a standard is quoted in an agreement it can thereby also be made binding.

Binding regulations can apply, for example, to safety for people and property while optional standards are used to facilitate activities such as work with specifications, selection of quality, measurements, manufacturing drawings etc.

3.10.2 Standards

Standards are quoted in many cases by legislators as a way of creating the desired level of safety. You can be considered as complying with the legislation's different demands if you follow the detailed directives given by the standards with regard to design, equipment and testing. Standards are useful to the manufacturer and the consumer. They increase interchangeability between components from different manufacturers and the possibility of comparison under the same conditions.

Standards are produced and issued nationally as well as on European and international levels. International standards, ISO or EN usually come into force as national standards (SIS in Sweden). It is the standardization body ISO (International Organization for Standardization) with its base in Geneva respective CEN (Commission Européenne pour la Standardization), that organise the international work.

SIS (Standardiseringskommissionen i Sverige) is the Swedish party in this work and in addition manages national standardisation in Sweden with the help of a number of specialised standardisation bodies. You can purchase all standards, national as well as international from SIS.

Besides official standards there are also documents produced by trade bodies such as PNEUROP an association of European manufacturer's of compressed air equipment. An example of such a document is the measurement standards for, e.g. compressor capacity, oil content in the compressed air, etc. which are issued while awaiting a standard to be drawn up.

Chapter 4 Economy

4.1 Economy

4.1.1 Costs for compressed air production

4.1.1.1 General

Electrical energy is the dominant energy type with virtually all industrial compressed air production. In many compressed air installations there are often significant and unutilised energy-saving possibilities through, e.g. energy recovery, pressure lowering, leakage reduction and by optimising operations through the choice of the control and regulation system.

It is profitable to look to the future as far as possible and try to assess the affects of new situations and demands that might apply to the installation when planning a new investment. Typical examples are environmental demands, energy saving demands, increased quality requirements from production and future production investments.

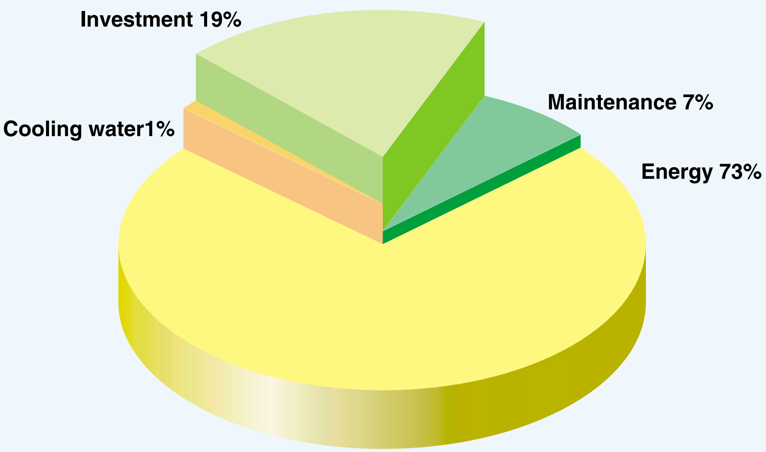
Optimised compressor operations are becoming more important, especially for larger, compressed air dependent industries. Production changes over time in a developing industry and thereby the conditions for compressor operations. It is therefore important that the compressed air supply is based both on the actual requirement and on plans for the future. Experience shows that an extensive and unbiased analysis of the operating situation results, on nearly every occasion, in improved overall economy.

Energy costs are clearly the dominating factor for the installation's overall economy. It is therefore important to concentrate on finding solutions that comply with demands of performance and quality as well as demands on efficient energy utilisation. The added cost involved with acquiring compressors and other equipment that comply with both of these demands will been seen in time as a good investment.

As energy consumption often represents approx. 80% of the overall cost you should exercise care when selecting the regulation system. The difference in regulation systems overshadows the difference

4:1

When analysing different expenses for the production of compressed air you will find a breakdown similar to the one in the diagram. However, the apportionment between different types of costs change with the number of operating hours/year, the equipment included in the calculation, the types of machine and the selected cooling system, etc.



between types of compressor. The ideal situation is when the compressor's full capacity is adapted exactly to balanced consumption, something frequently applied in process applications. Most types of compressors are supplied with their own control and regulation system, but the addition of equipment for co-control with other compressors in the installation can further improve the operating economy.

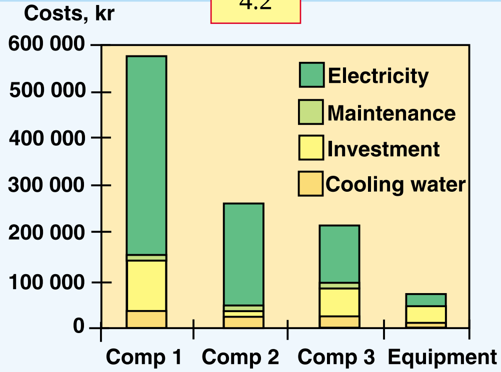
Speed regulation is becoming a popular regulation method, due to the power requirement being virtually proportional to the speed, drawn capacity. To think carefully and allow the requirement govern the selection of regulation equipment gives good results.

If a small amount of compressed air is required during the night and weekends, it can be profitable to install a small compressor adapted to this requirement. If, for some reason, you need another working pressure, the requirement should be analysed to discover whether the entire production can take place from a compressor centre or whether the network should be divided up for different pressure levels. Sectioning of the compressed air network can also come into question, in order to shutdown certain sections during the night and at weekends, to reduce air consumption or when you wish to apportion costs internally based on flow measurements.

4.1.1.2 Apportioning costs

Investment costs are a fixed cost made up of the purchasing price, building costs, installation and insurance. The cost of the investment as a part of the overall cost is connected partly with the selection of the quality of the compressed air and partly with the amortisation period and the calculation's interest rate. The size of energy costs are linked to the operating time per year, degree of utilisation and the energy price, etc. Some acquisition costs, for example, equipment for energy recovery give a direct pay off in the form of reduced operating and maintenance costs.

4:2



This is how costs can be divided between 3 compressors and auxiliary equipment. The large differences can be due to how the included machines are valued, that the capital value for an older machine may be higher than for a newly purchased machine, that the selected level of safety can affect the accounted maintenance costs, etc.

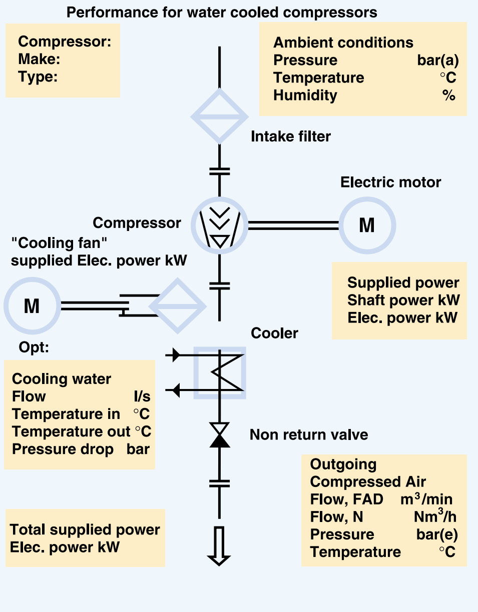
4.2 Opportunities for saving

4.2.1 Power requirement

When performing calculations it is important to bear in mind the overall power requirement. All energy consumers that belong to machines, should be observed, for example, intake filters, fans and pumps.

With comparisons between different investment options particular importance must be placed on the use of comparable values. Therefore you must be assured that the values are stated in accordance with

4:3



The figure shows a simple, but extremely useful model that can be used to give a true picture of a compressor's electrical energy requirement.

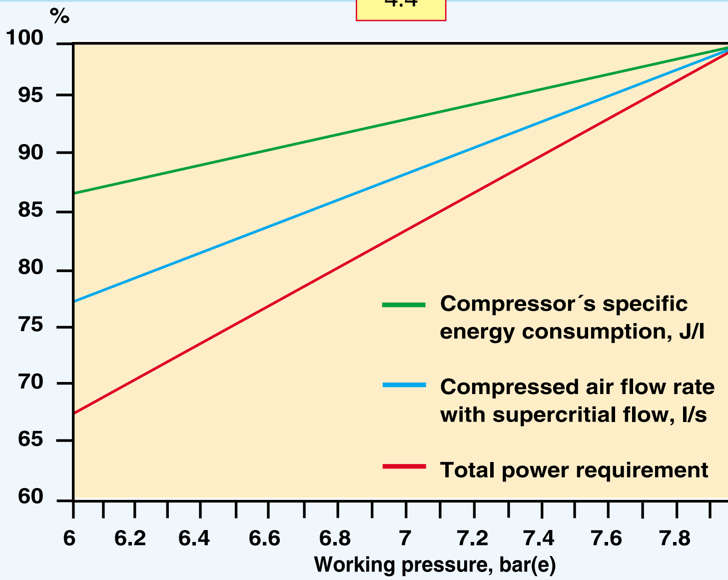
internationally agreed regulations, for example, as set out in ISO 1217. Ed3, supplement C-1996.

4.2.2 Working pressure

The working pressure directly affects the power requirement. High pressure means higher energy consumption. To increase the working pressure to compensate for pressure drop always results in impaired operating economy.

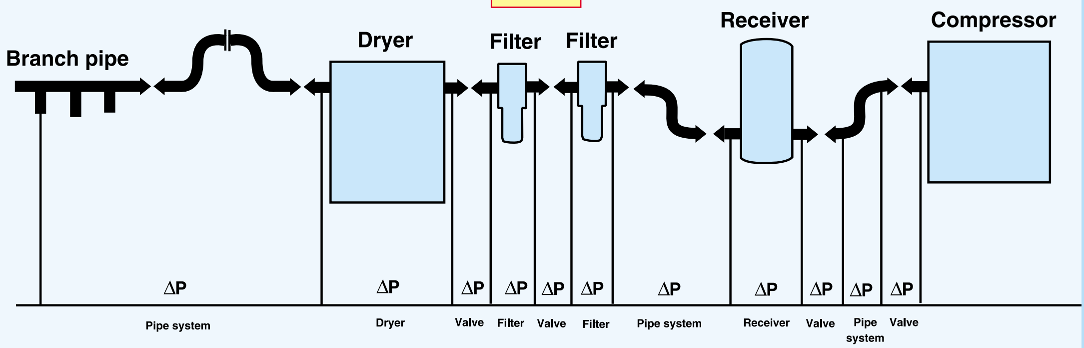
Despite this it is a common method, for example, with pressure drop caused by a too small pipe system or blocked filters. As an example , an increase in the working pressure by 1 bar, brings about an increase in the power requirement by approx. 6% . In an installation with several filters, especially if they have been operational for a long period of time without being replaced, the pressure drop can be significantly higher and therefore very costly.

4:4



The relation between pressure and electrical energy consumption. For a compressor of 15 m³/min a reduction of the working pressure from 7 to 6 bar means that the power requirement decreases from 91 to 85 kW. With a running time of 4000 hours/year this represents 24,000 kWh/year.

4:5



This is how the pressure drop across different components in the network affects the requisite working pressure.

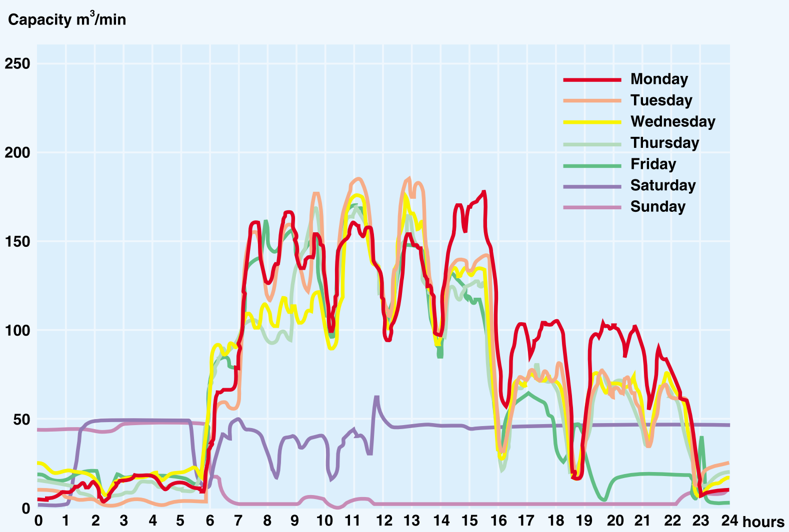
In many installations it is not possible to implement large pressure reductions, but using modern regulation equipment it is frequently fully realistic to lower the pressure by 0.5 bar. This means a saving of perhaps a few per cent, which may seem to be low, but if you consider that the total efficiency of the installation is increased to an equivalent degree it becomes more evident what the pressure reduction means.

4.2.3 Air consumption

By analysing routines and the use of compressed air, you can find solutions that give a more equal load on the compressed air system. The need of increased air production can thereby be kept low, which reduces operating costs.

Unprofitable consumption, which usually depends on leakage, worn equipment, processes that have not been adap-

4:6



The diagram shows how consumption can be spread during a week and 24 hours per day. Consumption is low during the night shift, high during the day shift, it drops during breaks, but is constant during the weekends (leakage or production).

ted or the incorrect use of compressed air is best rectified by increasing general awareness. Dividing the compressed air system into sections, that can be separated using valves, can be a method of reducing consumption during the night and at weekends. In most installations there is some leakage, which is a pure loss that must therefore be minimised. Frequently leakage claims 10-15% of the produced compressed air sometimes even more. Leakage is proportional to the working pressure, which is why one method of reducing leakage is to repair leaking equipment and lower the working pressure, e.g. at night.

A lowering of the pressure by only 0.3 bar reduces leakage by 4%. If the leakage in an installation of 100 m^3/min is 12% and the pressure is reduced by 0.3 bar, this represents a saving of approx. 3 kWh/hour, which is equivalent to the electricity consumption in a normal electrically heated home. Even the air consumption for machines and equipment increases with an increased working pressure.

4:7

Hole diameter		Output flow at 7 bar working pressure	Power requirement for the compressor
Size	mm	l/s	kW
●	1	1.2	0.4
●	3	11.1	4.0
●	5	31	10.8
●	10	124	43

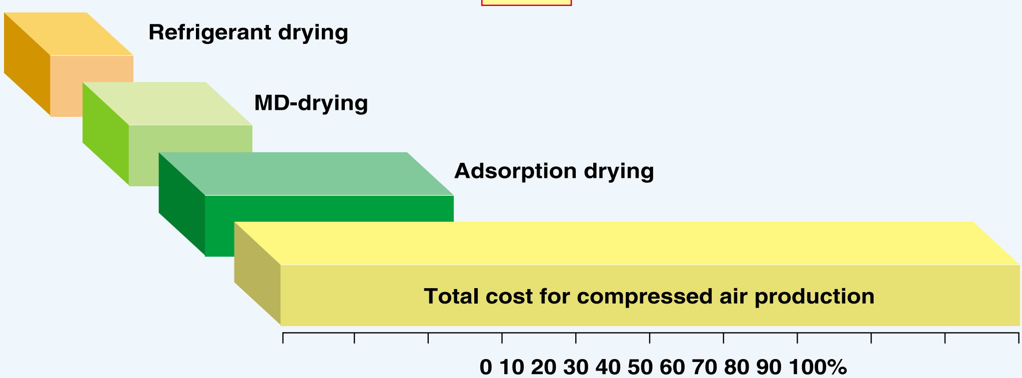
The table shows the relation between leakage and power consumption for some different (small) holes at a system pressure of 7 bar.

4.2.4 Regulation method

Using a modern, master control system the compressor central can be run optimally for different operating situations at the same time as safety and availability increase.

Selecting the right regulation method allows the supplied quantity of energy to be reduced through a lower system pressure and the degree of utilisation is optimised for each machine in the installation. At the same time availability increases, which reduces the risk of unplanned downtime. Besides, central control allows programming for automatic pressure reduction in the entire system, e.g. during operations at night and weekends.

4:9



A cost comparison between different drying methods.

As compressed air consumption is seldom constant, the compressor installation should have a flexible design, for example, through the use of compressors with different capacities and speed controlled motors. Compressors with a prudent design can be run with speed control and screw compressors are especially suited for this, as their capacity and power requirement are virtually proportional to the speed.

4.2.5 Air quality

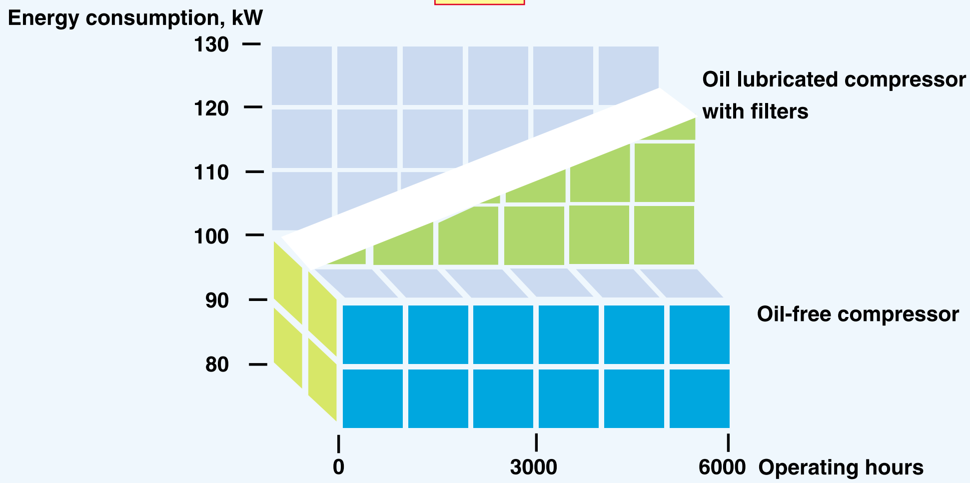
High quality compressed air reduces the need of maintenance, increases operating reliability of the pneumatic system, control system and instrumentation at the same time as wear to the air-powered machines is reduced.

If the compressed air system is adapted for dry compressed air from the outset the installation will be less expensive and simpler, as the pipe system does not need to be fitted with a water separator and so-called, swan-necks. When the air is dry it is not necessary to discharge air to the atmosphere to remove condensation. Condensation drainage on the pipe system is also not required, which means lower costs for installation and maintenance. The best economy can be gained by installing a central compressed air dryer. Decentralising air treatment, with several smaller units placed in the system, is more expensive and makes the system harder to maintain.

It is normally calculated that the reduced installation and maintenance costs for a system with dry compressed air cover the cost of the drying equipment. Profitability is very good, even when it is a question of supplementing existing installations with drying equipment.

Oil-free compressors do not need an oil separator or cleaning equipment for condensation. At the same time filters are not needed and therefore the cost for filter replacement is avoided. Consequently, there is no need to compensate for the pressure drop in the filter so compressor's working pressure can be lowered, which improves the installation's economy yet further.

4:10



An oil-free compressor gives constant compressed air quality to a fixed energy cost.

4.2.6 Energy recovery

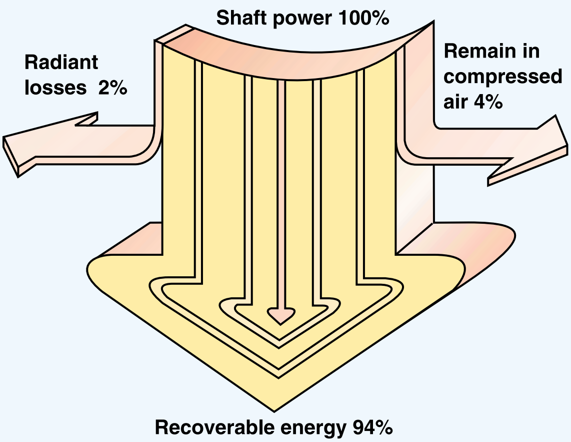
If you use electricity, gas or oil for any form of heating within the production facilities or in the process, there is good reason to investigate the possibilities of fully or partly replacing the energy with surplus energy from the compressor installation. The decisive factors are the alternative value in SEK/kWh for the surplus energy, degree of utilisation and the amount of additional investment necessary. A well planned investment in energy recovery often gives a repayment time of only 1–3 years. Over 90% of the power supplied to the compressor can be recovered in the form of highly valuable heat. The temperature level of the recovered energy determines the possible application areas and thereby the value.

The highest degree of efficiency is generally obtained from water cooled installations, when the compressor installation's outgoing cooling water can be connected directly to a continuous heating requirement, for example, the heating boiler's return circuit. Surplus energy can then be effectively utilised all year round. Different compressor designs give different prerequisites. In situations requiring a large heat flow, long transporting distances to the point of utilisation, a generally lower requirement or a requirement that varies during the year, it can be of interest to look at the possibilities of selling the recovered energy.

4.2.7 Maintenance

As all other equipment, even a compressor

4:11



Recovered energy kWh/year:

W $= [(K_1 \times Q_1) + (K_2 \times Q_2)] \times T_R$

Saving/year: $W \times e_p/\eta$

Saved oil m³/year: $W/6800 \times \eta$

- **W** = Recovered energy (kWh/year)
- T_R = Time per year when there is a need of power recovery (hours/year)
- K_1 = Part of T_R with loaded compressor
- K_2 = Part of T_R with off-loaded compressor
- Q_1 = Available power in coolant with loaded compressor (kW)
- Q_2 = Available power in coolant with off-loaded compressor (kW)
- e_p = Energy price
- η = Normal heat source's efficiency.

At the same time as the compressor produces compressed air, it also converts the supplied energy to heat, which is transferred to the coolant, air or water. Only a small part follows with the compressed air and is emitted as radiation from the machine and pipes. An air cooled compressor involves the simplest recovery system, while a water cooled compressor involves more efficient and more flexible recovery possibilities.

installation requires some form of maintenance. However, maintenance is low in relation to other costs, but can be reduced further through planning measures. The choice of the level of maintenance is determined by the installation's reliability and performance.

Maintenance makes up the smallest part of the installation's total cost. This is connected to a high degree on how the installation has been planned in general and the choice of compressor and the auxiliary equipment.

You can reduce costs by combining monitoring with other functions when using equipment for fully automatic operations and monitoring of the compressor central. The total budget for maintenance is affected by:

- **Type of compressors**
- **Auxiliary equipment (dryers, filters, control and regulation equipment)**
- **Operating situation**
- **Installation conditions**
- **Media quality**
- **Maintenance planning**
- **Choice of safety level**
- **Energy recovery/cooling system**
- **Degree of utilisation**

The annual cost usually lies between 5–10% of the machine's investment value.

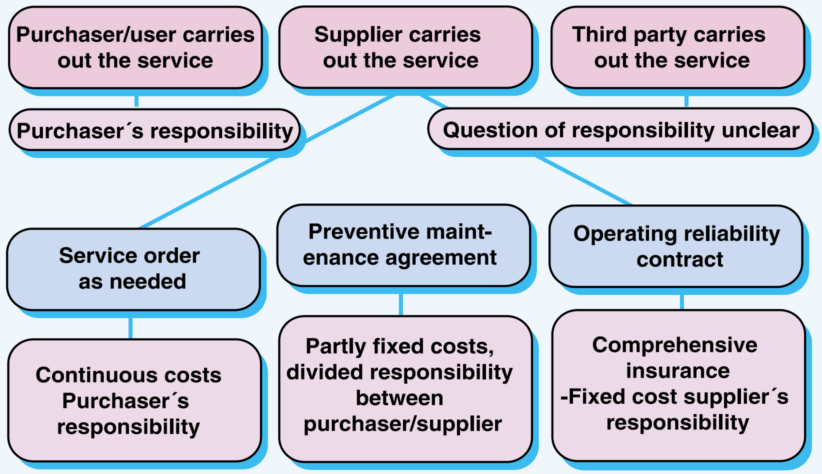
4.2.7.1 Maintenance planning

Well planned maintenance means that you can anticipate costs and extend the service life of the machine and auxiliary equipment. At the same time costs for repair of small faults decrease and downtime becomes shorter.

By utilising advanced electronics to a greater degree machines are equipped with instruments for diagnostic examination. This means that component parts can be utilised optimally and replacement takes place only when there is a need. The

4:12

Different forms of maintenance



The highest possible degree of utilisation reduces the cost of service and maintenance, counted in SEK/operating hour. It is realistic to plan for 100% utilisation and at least 98% availability.

need of reconditioning can be discovered at an early stage before becoming immense, thus avoiding subsequent damage and unnecessary downtime.

By utilising the supplier's service staff and original spare parts, you can expect the machine to have a high technical standard and you have the possibility of adopting modifications based on the latest findings during the machine's service life. The assessment of the maintenance requirement is made by specially trained technicians, who also answer for the training of in-house maintenance staff. You should always have you own skilled staff to take care of daily inspection, as ears and eyes can hear and see things that monitoring equipment cannot.

4.2.7.2 Auxiliary equipment

It is easy to expand an installation with numerous pieces of auxiliary equipment to, for example, increase the air quality or monitor the system. However, even auxiliary equipment needs service and brings about costs for maintenance, e.g. in the form of filter replacement, drying agent replacement, adaptation to other equipment and the training of staff.

In addition there are secondary maintenance costs, for example, the distribution network and production machines, which are affected by the quality of the compressed air and deposit costs for oil and filter cartridges. All of these costs must be evaluated in the calculation that forms the basis of an investment.

4.3 Other economic factors

4.3.1 General

You can describe and analyse a product, a material or service in a systematic way (yet simplified), with the help of a life cycle analysis, LCA. The LCA analyses all the stages in the product's live cycle. This means everything, from the selection of the raw material to the final waste depositing are included in the study.

The analysis is often used as a comparison between different options, for example, products with an equivalent function. The result is often used to provide guidance in issues concerning processes or product design. LCA can also be used by companies in communication with subcontractors, customers or the authorities to describe their product's characteristics.

The results from an LCA primarily act to form the basis for making decisions in the work minimise a product's effect on the environment. LCA does not give the answer to every question, which is why other aspects such as quality and the available technology must be examined to provide comprehensive background material.

4.3.2 LCC

LCC calculations (LCC = Life Cycle Cost) are used more and more as a tool to evaluate the different investment options. Included in the LCC calculation are the product's combined costs during a specific period, thus the capital cost, operating cost and the service cost.

The LCC calculation is often implemented based on a planned installation or a working installation and with this as a basis, defines the requirement level for the new installation. However, it is right to point out that a LCC calculation is often only a qualified guess with regard to future costs and is limited as it is based on today's knowledge of an installation's condition and energy price changes.

Neither does it bear in mind "soft" values that can be just as important, for example, production safety and subsequent costs.

To make an LCC calculation requires knowledge and preferably experience of compressed air installations. For the best result it should be made in consultation between the purchaser and the salesman. Central issues are, e.g. how different investment options affect factors such as production quality, production safety, the need of subsequent investment, maintenance of production machines and the distribution network, environment, the quality of the final product, risk assessment for downtime and rejections. An expression that must not be forgotten in this context is LCP, Life Cycle Profit, i.e. the earnings that can be made through, e.g. energy recovery and reduced rejections.

4:13

Example of a compressor calculation

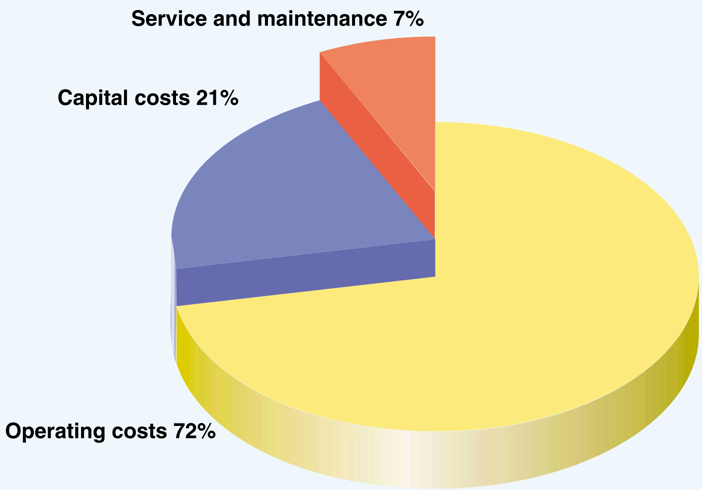
Input data						
Price of electricity	SEK/kWh	0.45				
Calculation interest	%	12				
Depreciation period	years	10				
Operating time	hours/year	6 000				
		Comp 1	**Comp 2**	**Comp 3**	**Dryers**	**TOTAL**
Annual consumption						
Electricity	MWh/year	1 200	555	406	133	2 294
Water (circulation system)	m^3/year	-	-	-	-	-
Divided operating costs						
Electricity	kSEK/year	540	250	183	60	1 033
Water	kSEK/year	6	4	2	0	12
Annual costs without energy recovery	**kSEK/year**	**760**	**383**	**197**	**114**	**1 454**
Operating costs	kSEK/year	547	254	185	60	1 046
Capital cost	kSEK	167	99	2	44	312
Service and maintenance	kSEK/year	45	30	11	11	97
Air production, total	**Mm^3/year**	**12 600**	**5 760**	**3 670**	**-**	**22 030**
Energy recovery						
Energy price (for the alternative use)	SEK/kWh	0.4	0.4	0.4	-	-
Recovery period	months/year	10	10	8	-	-
Degree of recovery	%	94	94	94	-	-
Quantity of recovered energy	MWh/year	893	395	233	-	1 521
Annual cost with energy recovery	**kSEK/year**	**413**	**234**	**109**	**114**	**870**
Saving with energy recovery	**kSEK/year**	**347**	**149**	**88**		**584**
Specific air costs 1, without energy recovery	öre/m^3	6.0	6.6	5.4	0.5	6.6
Specific air costs 2, with energy recovery	öre/m^3	3.3	4.1	3.0		3.9

Note:values have been rounded off.

When assessing service and maintenance costs you must also consider the equipment's expected condition after the calculation period has passed, that is whether it should be seen as consumed or be restored to its original condition.

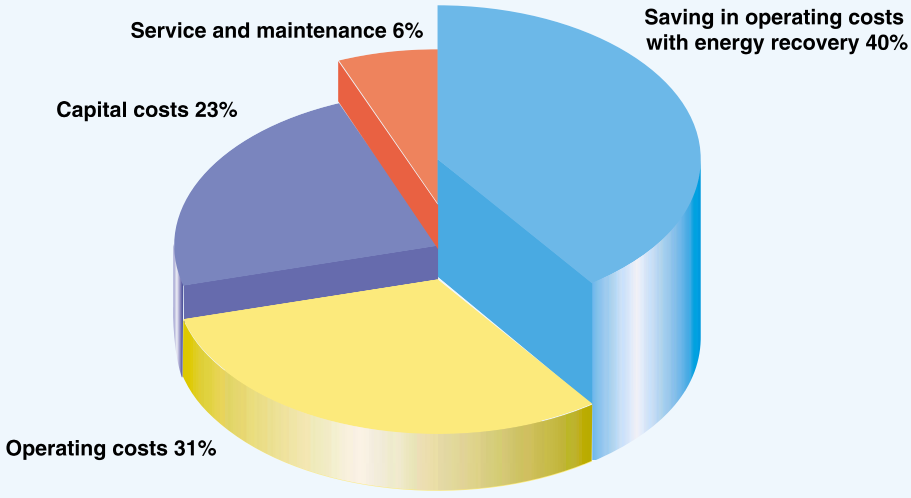
Furthermore, the calculation model must be adapted to the type of compressor. The example below can serve as a model for the economic evaluation of a compressor installation, with or without energy recovery.

4:14



This is how costs are divided without energy recovery.

4:15



This is how costs are divided with energy recovery.

Chapter 5 Calculations

5.1 Example of dimensioning compressed air installations

Below follows some normal calculations for dimensioning a compressed air installation. The intention is to show how some of the formulas and data from previous chapters are used. The example is based on a desired compressed air requirement and result in dimensioned data, based on components that can be chosen for the compressed air installation. After the example are a few additions that show how special cases can be handled.

5.2 Input data

The compressed air requirements and the ambient conditions must be established before dimensioning is started. In addition to this requirement, a decision as to whether the compressor shall be oil lubricated or oil-free and whether the equipment shall be water cooled or air cooled must be made.

5.2.1 Requirement

Assume that the need consists of three compressed air consumers.
They have the following data:

Consumer	Flow	Pressure	Dew point
1	12 Nm^3/min	6 bar(e)	+5°C
2	67 l/s (FAD)	7 bar(a)	+5°C
3	95 l/s (FAD)	4 bar(e)	+5°C

5.2.2 Ambient conditions (dimensioning)

Dimensioning ambient temperature: 20°C
Maximum ambient temperature: 30°C
Ambient pressure: 1 bar(a)
Humidity: 60%

5.2.3 Miscellaneous

Air cooled equipment
Compressed air quality from an oil lubricated compressor is regarded as sufficient.

5.3 Component selection

It is a good idea to recalculate all input data from the requirement table under 5.2.1 so that it is uniform with regard to type before dimensioning of the different components is started.

Flow: In general the unit l/s is used to define the compressor capacity, which is why consumer 1, given in Nm^3/min, must be recalculated.

$12\ Nm^3/min = 12 \times 1000/60 = 200\ Nl/s.$

Insertion of the current input data in the formula gives:

$$Q_{FAD} = \frac{Q_N \times (273 + T_i) \times 1.013}{273 \times P_i} = \frac{200 \times (273 + 35) \times 1.013}{273 \times 0.74} \approx 309\ l/s\ (FAD)$$

Pressure: The unit generally used to define pressure for the compressed air components is overpressure in bar, i.e. bar(e).

Consumer 2 is stated in absolute pressure, as 7 bar(a). The ambient pressure shall be detracted from this 7 bar to give the overpressure. As the ambient temperature in this case is 1 bar the pressure for consumer two can be written as 7-1 bar(e) = 6 bar(e).

With the recalculations set out above the table for uniform requirement:

Consumer	Flow	Pressure	Dew point
1	225 l/s (FAD)	6 bar(e)	+5°C
2	67 l/s (FAD)	6 bar(e)	+5°C
3	95 l/s (FAD)	4 bar(e)	+5°C

5.3.1 Dimensioning the compressor

The total consumption is the sum of the three consumers 225 + 67 + 95 = 387 l/s. A safety marginal of approx. 10-20% should be added to this, which gives a dimensioned flow rate of 387 x 1.15 ≈ 445 l/s (with 15% safety marginal).

The maximum required pressure for consumers is 6 bar(e). The dimensioned consumer is the one, including the pressure drop, that requires the highest pressure.

A reducing valve should be fitted to the consumer with the requirement of 4 bar(e). Assume at the moment that the pressure drop in the dryer, filter and pipe together does not exceed 1.5 bar. Therefore a compressor with a maximum working pressure of 7.5 bar(e) is suitable.

5.3.2 Assumption for the continued calculation

A compressor with the following data is selected:

Maximum pressure = 7.5 bar(e)
State flow at 7 bar(e) = 450 l/s
Total supplied power at 7 bar(e) = 175 kW
Supplied shaft power at 7 bar(e) = 162 kW
The compressed air temperature out of the compressor = ambient temperature +10°C.

Furthermore, the selected compressor has loading/unloading regulation with a maximum cycle frequency of 30 seconds. Using loading/unloading regulation the selected compressor has a pressure that varies between 7.0 and 7.5 bar(e).

Q_c = Compressor's capacity (l/s) = 450 l/s
P_1 = Compressor's intake pressure (bar(a)) = 1 bar(a)
T_1 = Compressor's maximum intake temperature (K) = 273 + 30 = 303 K
f_{max} = maximum cycle frequency = 1 cycle/30 seconds
$(p_U - p_L)$ = set pressure difference between loaded and unloaded compressor (bar) = 0.5 bar.
T_0 = Compressed air temperature out of the selected compressor is 10°C higher that the ambient temperature which is why the maximum temperature in the air receiver will be (K) = 273 + 40 = 313 K.

Compressor with loading/unloading regulation gives the right formula for the air receiver volume:

$$V = \frac{0.25 \times Q_c \times T_0}{f_{max} \times (p_U - p_L) \times T_1} = \frac{0.25 \times 450 \times 313}{1/30 \times 0.5 \times 303} = \underline{6\,972\text{ l}}$$

This is the minimum recommended air receiver volume.
The next standard size up is selected.

5.3.4 Dimensioning of the dryer

As the required dew point in this example is +6°C, a refrigerant dryer is the most suitable choice of dryer. When selecting the size of the refrigerant dryer a number of factors must be taken into consideration by correcting the refrigerant dryer's capacity using correction factors. These correction factors are unique for each refrigerant dryer model. Below the correction factors applicable for Atlas Copco's refrigerant dryers are used and are stated on the Atlas Copco's data sheet. The four correction factors are:

1. **Refrigerant dryer's intake temperature and pressure dew point.**
   As the compressed air temperature out of the selected compressor is 10°C higher than the ambient temperature, the refrigerant dryer's intake temperature will be maximum 30 + 10 = 40°C. In addition, the desired pressure dew point is +5°C.
   The correction factor 0.95 is obtained from Atlas Copco's data sheet.

2. **Working pressure**
   The actual working pressure in the compressor central is approx. 7 bar, which represents a correction factor of 1.0.

3. **Ambient temperature**
   With a maximum ambient temperature of 30°C a correction value of 0.95 is obtained.

Accordingly the refrigerant dryer should be able to handle the compressor's fully capacity multiplied by the correction factors above.
450 : 0.95 x 1.0 : 0.95 = $\underline{406\ l/s}$.

5.3.5 Assumptions for the continued calculation

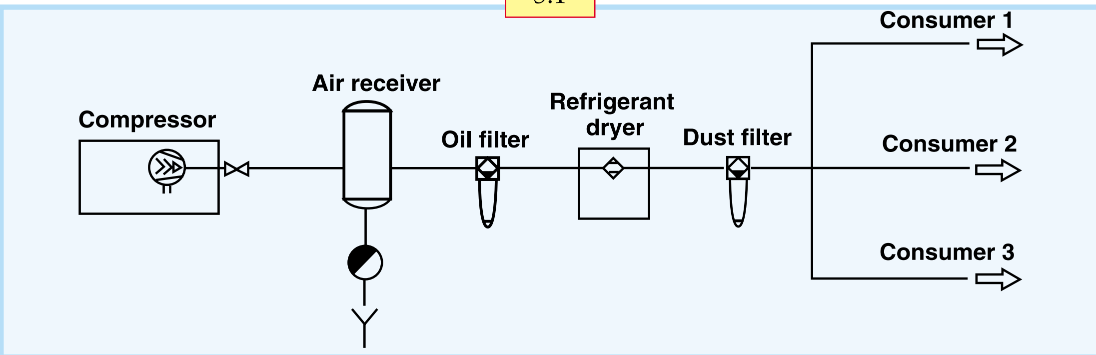
An air cooled refrigerant dryer with the followed data is selected:

Capacity at 7 bar(e) = 450 l/s
Total power consumption = 5.1 kW
Emitted heat flow to surroundings = 14.1 kW
Pressure drop across the dryer = 0.09 bar

When all the components for the compressor installation have been chosen, it should be checked that the pressure drop is not too great. This is done by adding together all the pressure drops for the components and pipes.

It may be appropriate to draw a schematic diagram of the compressed air installation as shown in fig. 5:1.

5:1



The pressure drop for the components is obtained from the component suppliers, while the pressure drop in the pipe system should not exceed 0.1 bar.

The total pressure drop can now be calculated:

Component	Pressure drop (bar)
Oil filter (pressure drop when filter is new)	0.14
Refrigerant dryer	0.09
Dust filter (pressure drop when filter is new)	0.2
Pipe system in compressor central	0.05
Pipe system from compressor central to consumption points	0.1
Total pressure drop:	**0.58**

The maximum pressure of 7.5 bar(e) and on-load pressure of 7.0 bar(e) for the selected compressor gives a lowest pressure at the consumers of 7.0 - 0.58 = 6.42 bar(e). You should add to this, the pressure drop increase across the filter that occurs over time. This pressure drop increase can be obtained from the filter supplier.

5.4 Other dimensioning

5.4.1 Condensation quantity calculation

As an oil lubricated compressor has been chosen, the water condensation separated in the compressor and refrigerant dryer will contain oil. The oil must be separated before the water is released into the sewer, which can be done in an oil separator. Information on how much water is condensed is needed in order to dimension the oil separator.

The total flow of water in the air taken in is obtained from the relation:
f_1 = relative humidity x the amount of water (g/litre) the air can carry at the maximum ambient temperature 30°C x air flow = 0.6 x 0.030078 x 445 ≈ 8.0 g/s.

The amount of air remaining in the compressed air after drying is subtracted from this quantity (saturated condition at +6°C).

$$f_2 = \frac{1 \text{ x } 0.007246 \text{ x } 445}{8} \approx 0.4 \text{ g/s}$$

The total condensation flow from the installation f_3 then becomes

$$f_1 - f_2 = 8.0 - 0.4 = 7.6 \text{ g/s} \approx \underline{27.4 \text{ kg/hour}}$$

With help of the calculated condensation flow the right oil separator can be chosen.

The principle that the supplied power to the room air shall be removed with the ventilation air is used to determine the ventilation requirement in the compressor room.

For this calculation the relationship for the power at a specific temperature change for a specific mass of a specific material is used.

$$Q = m \times c_p \times \Delta T$$

Q = the total heat flow (kW)
m = mass flow (kg/s)
c_p = specific heat capacity (kJ/kg, K)
ΔT = temperature difference (K)

The formula can be written as:

$$m = \frac{Q}{c_p \bullet \Delta T}$$

where:

ΔT = the ventilation air's temperature increase; presuppose that an increase of the air temperature with 10K can be accepted $\Rightarrow \Delta T = 10$ K.
c_p = specific heat capacity for the air = 1.006 kJ/kg x K (at 1 bar and 20°C)
Q = the total heat flow (in kW) = (94% of the supplied shaft power to the compressor + the difference between the supplied total power to the compressor and the supplied shaft power to the compressor + the stated heat flow from the refrigerant dryer) = 0.94 x 162 + (175 - 162) + 14.1 ≈ 180 kW

which gives the ventilation air

$$m = \frac{Q}{c_p \times \Delta T} = \frac{180}{1.006 \times 10} \approx 17.9 \text{ kg/s}$$

which at an air density of 1.2 kg/m^3 is equivalent to 17.9/1.2 ≈ 15m^3/s.

5.5 (Addition 1) At high altitude

Question: Presuppose the same compressed air requirement as described in the previous example at height of 2500 metres above sea level with a maximum ambient temperature of 35°C. How large compressor capacity (expressed as free air quantity) is required?

Answer: Air is thinner at altitude, which must be considered when dimensioning the compressed air equipment that has a compressed air requirement specified for a normal state (e.g. Nm^3/min). In those cases that the consumer's flow is stated in free air quantity (FAD) no recalculation is necessary.

As consumer 1 in the example above is given in the unit Nm^3/min the requisite flow for this consumer must be recalculated. The state at which the compressor's performance is normally stated is 1 bar and 20°C, which is why the state at 2500 metres above sea level must be recalculated into this state.

By using the table the ambient pressure 0.74 bar at 2500 metres above sea level is obtained. If the flow is recalculated to Nl/s (12 Nm^3/min = 12000/60 Nl/s = 200 Nl/s) and is set in the formula the following is obtained:

$$Q_{FAD} = \frac{Q_N \times (273 + T_i) \times 1.013}{273 \times P_i} = \frac{200 \times (273 + 35) \times 1.013}{273 \times 0.74} \approx 309 \text{ l/s (FAD)}$$

The total compressor capacity demanded is then 309 + 67 + 95 = 471 l/s (FAD).

5.6 (Addition 2) Intermittent output

Question: Presuppose that in the calculation example above there is an extra requirement from consumer 1 of a further 200 l/s for 40 seconds on the hour. It is accepted during this phase that the pressure in the system drops to 5.5 bar(e). How large should the receiver volume be to meet this extra requirement?

Answer: It is possible, during a short period, to take out more compressed air than what the compressor can manage by storing the compressed air in an air receiver. However, this requires that the compressor has a specific over capacity. The following relation applies for this:

$$V = \frac{Q \times t}{P_1 - P_2}$$

where

Q = air flow during the emptying phase = 200 l/s
t = length of the emptying phase = 40 seconds
$P_1 - P_2$ = permitted pressure drop during the emptying phase = normal pressure in the system - minimum accepted pressure during the emptying phase = 6.46 - 5.5 = 0.96 bar

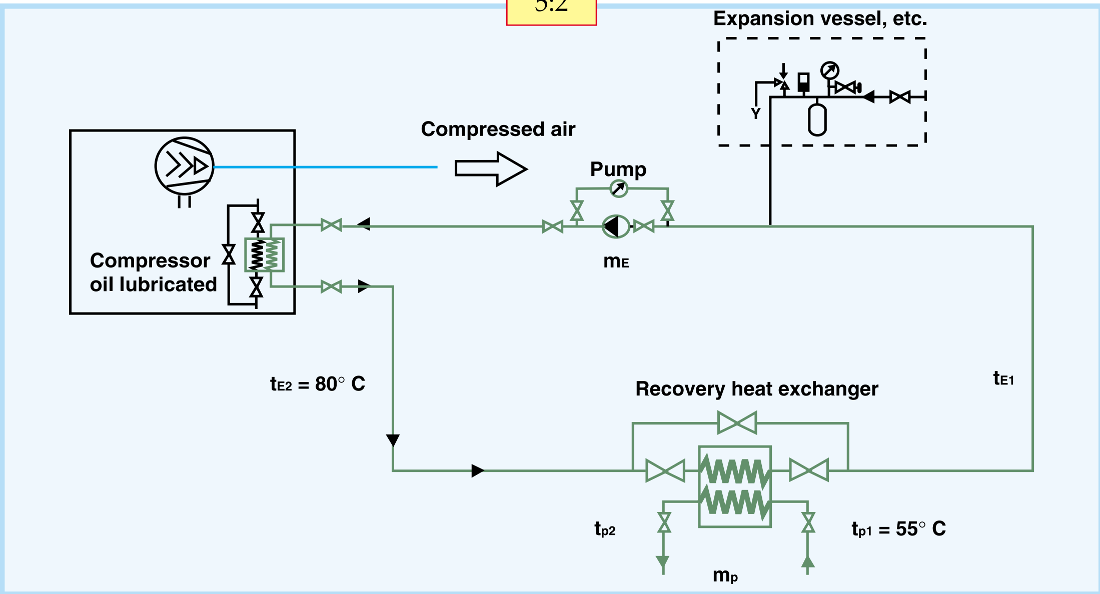
Inserted in the formula to give the requisite air receiver volume:

$$V = \frac{Q \times t}{P_1 - P_2} = \frac{200 \times 40}{0.96} = 8\,340 \text{ l}$$

In addition, it is required that the compressor has a specific over capacity, so that it can fill the air receiver after the emptying phase. The selected compressor has an over capacity of 5 l/s = 18000 litres/hour. As the air receiver is emptied once per hour the compressor's over capacity is more than sufficient.

5.7 (Addition 3) Water borne energy recovery

5:2



Question: A water borne energy recovery circuit is to be built for the compressor in the example. Presuppose that the water to be heated is in a warm water return (boiler return) with an ingoing return temperature of 55°C. Calculate the flow required for the energy recovery circuit and the power that can be recovered. Also calculate the flow and outgoing temperature for the boiler return.

Answer: Start be drawing the energy recovery circuit and name the different power, flow and temperatures. Now follow the calculation below.

Q_E = power transferred from the compressor to the energy recovery circuit (kW)
Q_P = power transferred from the energy recovery circuit to the boiler return (kW)
m_E = water flow in the energy recovery circuit (l/s)
m_P = water flow in the boiler return (l/s)
t_{E1} = water temperature before the compressor (°C)
t_{E2} = water temperature after the compressor (°C)
t_{P1} = ingoing temperature on the boiler return (°C)
t_{P2} = outgoing temperature on the boiler return (°C)

5.7.1 Assumption

The following assumption has been made:

A suitable water temperature out of the compressor for energy recovery can be obtained from the compressor supplier and is assumed to be in this example t_{E2}= 80°C.
Assumption for the water circuit through the energy recovery heat exchanger:

$$t_{E1} = t_{P1} + 5°C$$

$$t_{P2} = t_{E2} - 5°C$$

In addition it is assumed that the pipe and heat exchanger have no heat exchange with the surroundings.

5.7.2 Calculation of the cooling water flow in the energy recovery circuit

$$Q = m \times c_p \times \Delta T$$

ΔT = temperature increase across the compressor = $t_{E2} - t_{E1}$ = 80°C - 60°C = 20°C
c_p = specific heat capacity for water = 4.18kJ/kg x K
Q = Q_E = the power that can be taken care of = 70% of the supplied shaft power = 0.70 x 162 = 132 kW.
This is the power possible to recover for the selected compressor.
m = mass flow in the energy recovery circuit = m_E.

The formula can be written as:

$$m_E = \frac{Q_E}{c_p \times \Delta T} = \frac{113}{4.18 \times 20} = \underline{1.35 \text{ kg/s}}$$

5.7.3 Energy balance across the recovery heat exchanger

For the recovery heat exchanger applies:

$$Q_E = m_E \times c_p \times (t_{E2} - t_{E1})$$

$$Q_p = m_p \times c_p \times (t_{P2} - t_{P1})$$

However, as it has been presupposed that no heat exchange shall take place with the surroundings, the power transferred to the energy recovery circuit from the compressor will be equal to the power transferred in the recovery heat exchanger, i.e. $Q_P = Q_E = 113$ kW.

The formula can be written as:

$$m_P = \frac{Q_P}{(t_{P2} - t_{P1}) \times c_p} = \frac{113}{(75 - 55) \times 4.18} \approx \underline{1.35\ l/s}$$

5.7.4 Compilation of the answer

It can be established from the calculation that the power which can be recovered is 113 kW. This requires a water flow in the energy recovery circuit of 1.35 l/s. An appropriate flow for the boiler return is also 1.35 l/s with an increase of the boiler temperature by 20°C.

5.8 (Addition 4) Pressure drop in the piping

Question: A 23 metre pipe with an inner diameter of 80 mm shall lead a flow of 140 l/s. The pipe is routed with 8 elbows that all have a bend diameter equal to the inner diameter. How great will the pressure drop across the pipe be if the absolute initial pressure is 8 bar(a)?

Answer: First the equivalent pipe length for the 8 elbows must be determined. The equivalent pipe length of 1.3 metres per elbow can be read off from the table 3:36. The total pipe length is then 8 x 1.3 + 23 = 33.4 meters. The following formula is used to calculate the pressure drop:

$$\Delta p = 450 \frac{Q_c^{1.85} \times l}{d^5 \times p}$$

Insertion gives:

$$\Delta p = 450 \frac{140^{1.85} \times 33.4}{80^5 \times 8} \approx 0.0054 \text{ bar}$$

Accordingly, the total pressure drop across the pipe will be 0.0054 bar

Chapter 6 Quantities, units and symbols

6.1 The SI-system

Since 1964 SI has been the Swedish standard within the area of quantities and units where fundamental information can be found in SIS 01 61 18 (General principles and writing rules) and SIS 01 61 26 (Prefixes for multiple units) and SIS 01 61 32 (SI-units, derived and additional units).

Units are divided into four different classes:

Base units
Supplementary units
Derived units
Additional units

If a prefix (micro, milli, kilo, mega, etc.) is placed before a unit, the formed unit is then called a multiple unit.

Base units, supplementary units and derived units are called *SI units* and multiple units formed by SI units are called *units in SI*. Note that additional units and not units in SI.

Base units are one of the established, independent units in which all other units can be expressed.

There are 7 base units in SI:

for length	metre	m
for mass	kilogram	kg
for time	second	s
for electrical current	ampere	A
for temperature	kelvin	K
for luminous intensity	candela	cd
for the amount of substance	mole	mol

Supplementary units are units of a basic nature, but are not classified as base units or derived units.

Two supplementary units are included in SI:

for plane angles	radian	rad
for solid angles	steradian	sr

Derived units and formed as a power or products of powers of one or more base units and/or supplementary units according to physical laws for the relation between different units.

15 derived units have been given their own names:

Quantity	Designation	Symbol	Expressed in other SI units	Expressed in base and supplementary units
frequency	hertz	Hz	-	s^{-1}
force	newton	N	-	$m \times kg x s^{-2}$
pressure, mechanical stress	pascal	Pa	N/m^2	$m^{-1} \times kg \times s^{-2}$
energy, work	joule	J	N x m	$m^2 \times kg \times s^{-2}$
power	watt	W	J/s	$m^{-2} \times kg \times s^{-3}$
electrical quantity, charge	coulomb	C	A x s	s x A
electrical voltage	volt	V	W/A	$m^2 \times kg \times s^{-3} \times A^{-1}$
capacitance	farad	F	C/V	$m^{-2} \times kg^{-1} \times s^4 \times A^{-2}$
resistance	ohm	W	V/A	$m^2 \times kg \times s^{-3} \times A^{-2}$
conductivity	siemens	S	A/V	$m^{-2} \times kg^{-1} \times s^3 \times A^2$
magnetic flux density	tesla	T	Wb/m^2	$kg \times s^{-2} \times A^{-1}$
magnet flux	weber	Wb	V x s	$m^2 \times kg \times s^{-2} \times A^{-1}$
inductance	henry	H	Wb/A	$m^2 \times kg \times s^{-2} \times A^{-2}$
luminous flux	lumen	lm	cd x sr	cd x sr
light	lux	lx	lm/m^2	$cd \times sr \times m^2$

Additional units. There are a number of units outside of SI, which for different reasons cannot be eliminated despite that corresponding units in principle can be expressed in SI units. A number of these units have been selected to be used with the units in SI and are called additional units.

There are also a further four additional units primarily for use within astronomy and physics. All of these additional units are approved by Comité International des Poids et Mesures (CIPM) 1969 and are used together with SI units.

The following additional units for technical use occur:

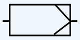
Quantity	Additional unit: Designation	Additional unit: Symbol	Remarks
plane angle	degree	...°	$1° = \frac{\pi}{180}$ rad
plane angle	minute	...’	$1' = \frac{1°}{60}$
plane angle	second	...”	$1'' = \frac{1'}{60}$
volume	litre	l	1 l = 1 dm^3
time	minute	min	1 min = 60 sec
time	hour	h	1 h = 60 min = 3.600 sec
time	day	d	1 d = 24 h
mass	tonne (metric)	t	1 t = 1.000 kg
pressure	bar	bar	1 bar = 10^5 Pa = 10^5 N/m^2

Multiple units. A multiple unit is formed by an SI unit or an additional unit by the unit being preceded by a prefix that involves the multiplication by the power of ten. Fourteen such prefixes are stated in international recommendations (standards) as set out in the table below.

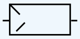
Power	Prefix: Designation	Prefix: Symbol	Example	
10^{12}	tera	T	1 terajoule	1 TJ
10^{9}	giga	G	1 gigawatt	1 GW
10^{6}	mega	M	1 megavolt	1 MV
10^{3}	kilo	k	1 kilometre	1 km
10^{2}	hectoh	h	1 hectogram	1 hg
10^{1}	deca	da	1 decalumen	1 dalm
10^{-1}	deci	d	1 decimetre	1 dm
10^{-2}	centi	c	1 centimetre	1 cm
10^{-3}	milli	m	1 milligram	1 mg
10^{-6}	micro	μ	1 micrometre	1 μm
10^{-9}	nano	n	1 nanohenry	1 nH
10^{-12}	pico	p	1 picofarad	1 pF
10^{-15}	femto	f	1 femtometre	1 fm
10^{-18}	atto	a	1 attosecond	1 as

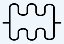
Pressure					
Pa=N/m²	**bar**	**kp/cm²**	**Torr**	**m vp**	**mm vp**
1	10^{-5}	1.02×10^{-5}	7.5×10^{-3}	1.02×10^{-4}	0.102
10^{5}	1	1.02	750	10.2	1.02×10^{4}
9.81×10^{4}	0.981	1	735	10	10^{4}
133,3	1.33×10^{-3}	1.36×10^{-3}	1	1.36×10^{-2}	13,6
9.81×10^{3}	9.81×10^{-2}	0.1	73.5	1	10^{3}
9.81	9.81×10^{-5}	10^{-4}	7.35×10^{-2}	10^{-3}	1
Energy					
J	**kJ**	**kWh**	**kpm**	**kcal**	
1	10^{-3}	2.78×10^{-7}	0.102	2.39×10^{-4}	
1000	1	2.78×10^{-4}	102	0.239	
3.6×10^{6}	3.6×10^{3}	1	3.67×10^{5}	860	
9.81	9.81×10^{-3}	2.72×10^{-6}	1	2.39×10^{-3}	
4.19×10^{3}	4.19	1.16×10^{-3}	427	1	
Power					
W	**kpm/s**	**kcal/s**	**kcal/h**	**hk**	
1	0.102	0.239×10^{-3}	0.860	1.36×10^{-3}	
9.81	1	2.34×10^{-3}	8.43	1.33×10^{-2}	
4.19×10^{3}	427	1	3.6×10^{3}	5.69	
1.163	0.119	0.278×10^{-3}	1	1.58×10^{-3}	
735	75	0.176	632	1	

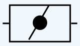
6.2 Drawing symbols

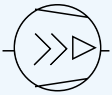

Air filter


Silencer

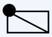

Diffusor

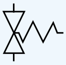

Compensator

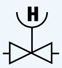

Damper valve


Screw compressor

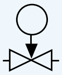

Non-return valve

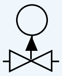

Stop valve

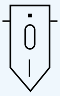

Safety valve

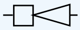


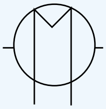
Manual valve

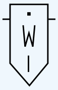

Pressure closing valve


Pressure opening valve


Oil separator


Ejector

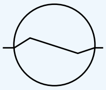

Water cooled cooler

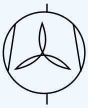


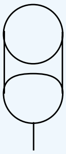
Water separator

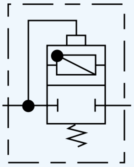

Condensation receiver

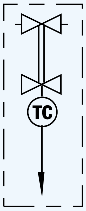

Direction of flow


Air cooled cooler

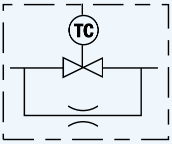

Fan


Expansion vessel


Minimum pressure valve

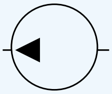


Thermostat valve

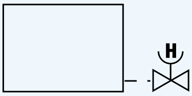


Thermostat valve

Fluid filter



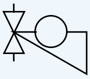
Fluid pump



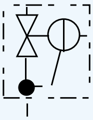
Oil tank with manual drainage

Restrictor

Bypass



Pressure regulator



Oil stop valve

Air

Oil

Water

Drainage

Ventilation ducting

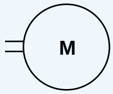
Mechanical connection

Signal air

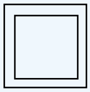
Electrical signal

Border line

Electrical energy



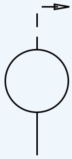
Electric motor



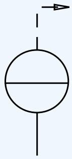
Electrical distribution box

Motor coupling

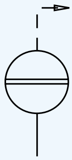
Blind flange



Sensor for pressure, temperature, etc.



Sensor for pressure, temperature, etc.

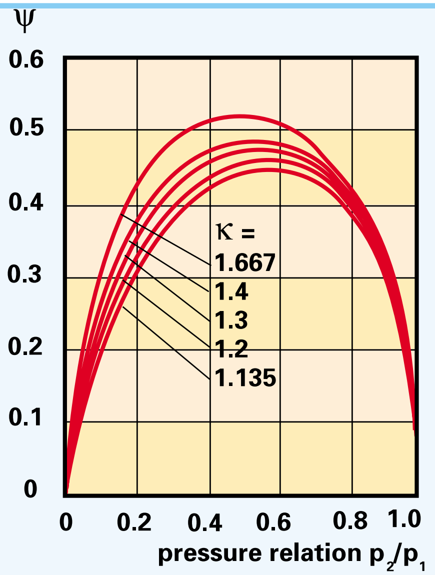


Sensor for pressure, temperature, etc.

6.3 Diagrams and tables

Material	J/kg x K
air (atmospheric pressure)	1 004
aluminium	920
copper	390
oil	1 670-2 140
steel	460
water	4 185
zinc	385

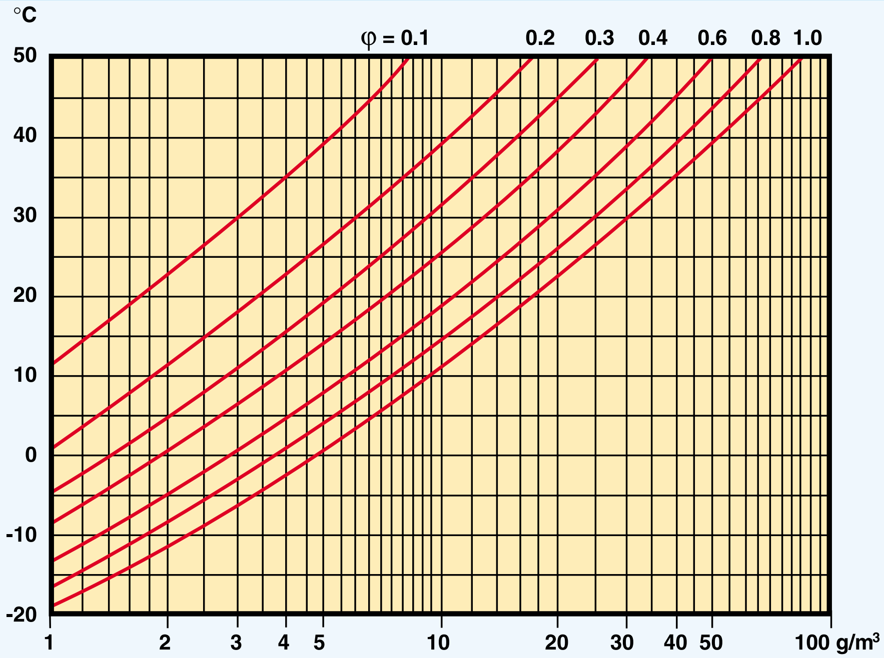
Heat capacity for some materials.



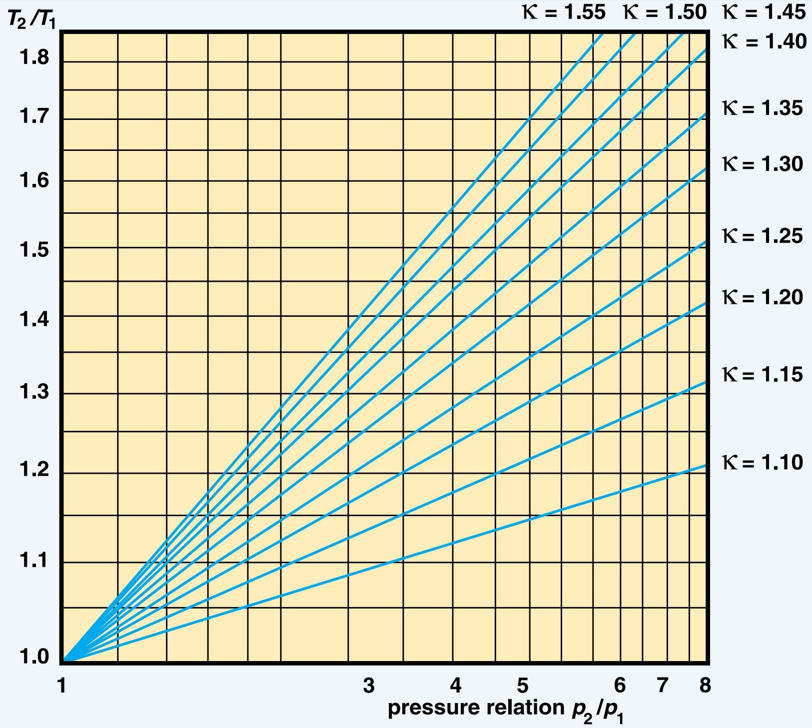
Flow coefficients as a function of the pressure relation for different κ-values.

boiling point	78.8	K
critical pressure (a)	37.66	bar
critical temperature	132.52	K
specific weight	1.225	kg/m^3
dynamic viscosity	17.89×10^{-6}	Pa x s
freezing point	57-61	K
gas constant	287.1	J/(kg x K)
kinematic viscosity	14.61×10^{-6}	m^2/s
molar weight	28.964	(dimensionless)
heat capacity at:		
constant pressure	1.004	kJ/(kg x K)
specific heat capacity ratio	1.40	(dimensionless)
speed of sound	340.29	m/s
thermal conductivity	0.025	W/(m x K)

Some physical constants for dry air at sea level (=15°C and 1013 bar).



Water content in air at different relative vapour pressures (φ).
The maximum water content is doubled with each temperature rise of 11 K.



The diagram show the temperature relationship T_2/T_1 for different gases with different κ-values during isentropic compression.

t °c	Ps mbar	ρw g/m³	t °c	Ps mbar	ρw g/m³
-40	0.128	0.119	5	8.72	6.80
-38	0.161	0.146	6	9.35	7.26
-36	0.200	0.183	7	10.01	7.75
-34	0.249	0.225	8	10.72	8.27
-32	0.308	0.277	9	11.47	8.82
-30	0.380	0.339	10	12.27	9.40
-29	0.421	0.374	11	13.12	10.01
-28	0.467	0.413	12	14.02	10.66
-27	0.517	0.455	13	14.97	11.35
-26	0.572	0.502	14	15.98	12.07
-25	0.632	0.552	15	17.04	12.63
-24	0.689	0.608	16	18.17	13.63
-23	0.771	0.668	17	19.37	14.48
-22	0.850	0.734	18	20.63	15.37
-21	0.937	0.805	19	21.96	16.31
-20	1.03	0.884	20	23.37	17.30
-19	1.14	0.968	21	24.86	18.34
-18	1.25	1.06	22	26.43	19.43
-17	1.37	1.16	23	28.09	20.58
-16	1.51	1.27	24	29.83	21.78
-15	1.65	1.39	25	31.67	23.05
-14	1.81	1.52	26	33.61	24.38
-13	1.98	1.65	27	35.65	25.78
-12	2.17	1.80	28	37.80	27.24
-11	2.38	1.96	29	40.06	28.78
-10	2.60	2.14	30	42.43	30.38
-9	2.84	2.33	31	44.93	32.07
-8	3.10	2.53	32	47.55	33.83
-7	3.38	2.75	33	50.31	35.68
-6	3.69	2.99	34	53.20	37.61
-5	4.02	3.25	35	56.24	39.63
-4	4.37	3.52	36	59.42	41.75
-3	4.76	3.82	37	62.76	43.96
-2	5.17	4.14	38	66.28	46.26
-1	5.62	4.48	39	69.93	48.67
0	6.11	4.85	40	73.78	51.19
1	6.57	5.19	41	77.80	53.82
2	7.06	5.56	42	82.02	58.56
3	7.58	5.95	43	86.42	59.41
4	8.13	6.36	44	91.03	62.39

Saturation pressure (P_S) and density (ρ_W) for saturated water vapour.

Gas	Volume %	Weight %
nitrogen N_2	78.084	75.520
oxygen O_2	20.947 6	23.142
argon Ar	0.934	1.288
carbon dioxide CO_2	0.031 4	0.047 7
neon Ne	0.001 818	0.001 267
helium He	0.000 524	0.000 072 4
krypton Kr	0.000 114	0.000 330
xenon Xe	0.000 008 7	0.000 039
hydrogen H_2	0.000 05	0.000 003
methane CH_4	0.000 2	0.000 1
nitrous oxide N_2O	0.000 05	0.000 08
ozone O_3	summer: 0 to 0.000 007	0 to 0.000 01
	winter: 0 to 0.000 002	0 to 0.000 003
sulphur dioxide SO_2	0 to 0.000 1	0 to 0.000 2
nitrogen dioxide NO_2	0 to 0.000 002	0 to 0.000 003
ammonia NH_3	Ca 0	Ca 0
carbon monoxide CO	Ca 0	Ca 0

Composition of clean, dry air at sea level. This composition is relatively constant up to a height of 25 km.

Machine type and size	Air requirement max. l/s
Drilling machines, Ø = bit diameter (mm)	
Small Ø < 6.5	**6.0**
Medium 6.5 < Ø = < 10	**7.5**
Large 10 < Ø < 16	**16.5**
Thread cutters	**6**
Screwdriver, d = screw size	
Small d < M6	**5.5**
Medium M6 < d < M8	**7.5**
Impact wrench, d = bolt size	
Small d < M10	**5.0**
Medium M10 < d < M20	**7.5**
Large d ≥ M20	**22.0**
Filing machine	**7.5**
Polishers/Die grinders, e = power (kW)	
Small e < 0.5	**8.0**
Large e > 0.5	**16.5**
Grinders, e = power (kW)	
Small 0.4 < e < 1.0	**20.0**
Medium 1,0 < e < 2	**40.0**
Large e > 2	**60.0**
Chipping hammers	
Light	**6.0**
Heavy	**13.5**
Air hoists t = lifting tonnage	
t < 1 tonne	**35**
t > 1 tonne	**45**
Scaler	**5.0**
Cleaning nozzle	**6.0**
Nutrunner, d = bolt size	
d ≤ M8	**9**
d ≥ M10	**19**

Some examples of air consumption for some common power tools and machines based on experience. These values from the basis for calculating the requisite compressor capacity.

Dew point °C	g/m³	Dew point °C	g/m³	Dew point °C	g/m³	Dew point °C	g/m³
+100	588.208	+58	118.199	+16	13.531	−25	0.55
99	569.071	57	113.130	15	12.739	26	0.51
98	550.375	56	108.200	14	11.987	27	0.46
97	532.125	55	103.453	13	11.276	28	0.41
96	514.401	54	98.883	12	10.600	29	0.37
95	497.209	53	94.483	11	9.961	30	0.33
94	480.394	52	90.247	10	9.356	31	0.301
93	464.119	51	86.173	9	8.784	32	0.271
92	448.308	50	82.257	8	8.243	33	0.244
91	432.885	49	78.491	7	7.732	34	0.220
90	417.935	48	74.871	6	7.246	35	0.198
89	403.380	47	71.395	5	6.790	36	0.178
88	389.225	46	68.056	4	6.359	37	0.160
87	375.471	45	64.848	3	5.953	38	0.144
86	362.124	44	61.772	2	5.570	39	0.130
85	340.186	43	58.820	1	5.209	40	0.117
84	336.660	42	55.989	0	4.868	41	0.104
83	324.469	41	53.274			42	0.093
82	311.616	40	50.672	−1	4.487	43	0.083
81	301.186	39	48.181	2	4.135	44	0.075
80	290.017	38	45.593	3	3.889	45	0.067
79	279.278	37	43.508	4	3.513	46	0.060
78	268.806	36	41.322	5	3.238	47	0.054
77	258.827	35	39.286	6	2.984	48	0.048
76	248.840	34	37.229	7	2.751	49	0.043
75	239.351	33	35.317	8	2.537	50	0.038
74	230.142	32	33.490	9	2.339	51	0.034
73	221.212	31	31.744	10	2.156	52	0.030
72	212.648	30	30.078	11	1.96	53	0.027
71	204.286	29	28.488	12	1.80	54	0.024
70	196.213	28	26.970	13	1.65	55	0.021
69	188.429	27	25.524	14	1.51	56	0.019
68	180.855	26	24.143	15	1.38	57	0.017
67	173.575	25	22.830	16	1.27	58	0.015
66	166.507	24	21.578	17	1.15	59	0.013
65	159.654	23	20.386	18	1.05	60	0.011
64	153.103	22	19.252	19	0.96	65	0.0064
63	146.771	21	18.191	20	0.88	70	0.0033
62	140.659	20	17.148	21	0.80	75	0.0013
61	134.684	19	16.172	22	0.73	80	0.0006
60	129.020	18	15.246	23	0.66	85	0,00025
59	123.495	17	14.367	24	0.60	90	0.0001

The water content in the air at different dew points.

6.4 Compilation of current standards and norms

Here follows a compilation of current (1997) standards and norms within the compressed air field. The compilation refers to Swedish regulations, but in most cases there are equivalent national regulations in other countries. The listed standards are all, with some exceptions, European or international. Pneurop documents are usually issued with a parallel CAGI issue for the American market.

It is always important to check with the issuing body that the latest issue is being used, unless the requirement/demand refers to a dated issue.

6.4.1 Safety related regulations and standards

6.4.1.1 Machine safety

EU directive 89/392/EEG, Machinery directive. In Sweden this has been made law as AFS 93:10 (modified as AFS 94:48). National Swedish Board of Occupational Safety and Health regulations for machinery

EN 1012-1 Compressors and vacuum pumps – Safety demands – Del 1: Compressors

EN 1012-2 Compressors and vacuum pumps – Safety demands – Del 2: Vacuum pumps

6.4.1.2 Pressure safety

Directive 87/404/EEG, Simple pressure vessels. In Sweden this has been made law as AFS 93:41 I (modified as AFS 94:53) National Swedish Board of Occupational Safety and Health regulations for simple pressure vessels

Directive 76/767/EEG Covers common legislation for pressure vessels and methods of inspecting. Directive 97/23/EG for pressure equipment (applies from 1999-11-29)

AFS 86:9 (modified as AFS 94:39)) National Swedish Board of Occupational Safety and Health regulations for pressure vessels and other pressure equipment

EN 764 Pressure equipment - Terminology and symbols - Pressure, temperature.

EN 286-1 Simple unfired pressure vessels designed to contain air or nitrogen - Part 1: Design, manufacture and testing

EN 286-2 Simple unfired pressure vessels designated to contain air or nitrogen - Part 2: Pressure vessels for air braking and auxiliary systems for motor vehicles and their trailers

EN 286-3 Simple unfired pressure vessels designed to contain air or nitrogen - Part 3: Steel pressure vessels designed for air braking equipment and auxiliary pneumatic equipment for railway rolling stock

EN 286-4 Simple unfired pressure vessels designed to contain air or nitrogen - Part 4: Aluminum alloy pressure vessels designed for air braking equipment and auxiliary pneumatic equipment for railway rolling stock

6.4.1.3 Environment

Pneurop PN8NTCI, Noise test code for compressors. ISO 84/536/EC. Sound level demands for machinery. A special standard for sound measurements is being drawn up within ISO.

For the emission of exhaust fumes from combustion engines a decision is expected according to "EU stage 1'

6.4.1.4 Electrical safety

ELSAK-FS 1994:9 National Swedish Board of Electrical Safety's regulations for electrical material.

ELSAK-FS 1994:7 National Swedish Board of Electrical Safety's regulations for heavy current installations (equivalent to IEC 364)

EU directive 89/336/EEG Electromagnetic compatibility

ELSAK-FS 1995:5 National Swedish Board of Electrical Safety's regulations for electromagnetic compatibility

EN 60204-1 Machinery, Electrical safety instructions

EN 60439-1 Low-voltage switchgear and control gear assemblies

6.4.2 Technical related standards and norms

6.4.2.1 Standardization

SS 1796 Compressed air technology - Terminology

ISO 3857-1 Compressors, pneumatic tools and machines - Vocabulary - Part 1: General

ISO 3857-2 Compressors, pneumatic tools and machines - Vocabulary - Part 2: Compressors

ISO 5390 Compressors - Classification

ISO 5941 Compressors, tools and machines Preferred pressures

6.4.2.2 Specifications

SS-ISO 1217 Compressed air technology – displacement compressors – delivery tests

ISO 5389 Turbo-compressors - Performance test code ISO 7183-1 Compressed air dryers - Part 1: Specifications and testing

ISO 7183-1 Compressed air dryers - Part 2: Performance ratings

ISO 8010 Compressors for the process industry - Screw and related types - Specifications and data sheets for their design and construction

ISO 8011 Compressors for the process industry - Turbo types - Specifications and data sheets for their design and construction

ISO 8012 Compressors for the process industry - Reciprocating types - Specifications and data sheets for their design and construction

SS-ISO 8573-1 Compressed air for general use - Part 1: Contaminants and quality classes. For enforcement of the current national Swedish regulations there are:

Pressure vessel norms 1987
Piping norms 1978
Air container norms 1991
Issued by the Pressure vessel standardization
EG directive 73/23/EEG Low voltage directive

6.4.2.3 Measurements

ISO 8573-2 Compressed air for general use Part 2: Test methods for oil aerosol content

(Draft) ISO 8573-3 Compressed air - Part 3: Measurement of humidity

(Draft) ISO 8573-4 Compressed air - Part 4: Measurement of solid particles

Index

T

VW

Notes

What sets Atlas Copco apart as a company is our conviction that we can only excel in what we do if we provide the best possible know-how and technology to really help our customers produce, grow and succeed.

There is a unique wayof achieving that - we simply call it the Atlas Copco way. It builds on **interaction**, on long-term relationships and involvement in the customers' process, needs and objectives. It means having the flexibility to adapt to the diverse demands of the people we cater for.

It's the **commitment** to our customers' business that drives our effort towards increasing their productivity through better solutions. It starts with fully supporting existing products and continuously doing things better, but it goes much further, creating advances in technology through **innovation**. Not for the sake of technology, but for the sake of our customers' bottom line and peace-of-mind.

That is how Atlas Copco will strive to remain the first choice, to succeed in attracting new business and to maintain our position as the industry leader.

Atlas Copco

9780 0380-10