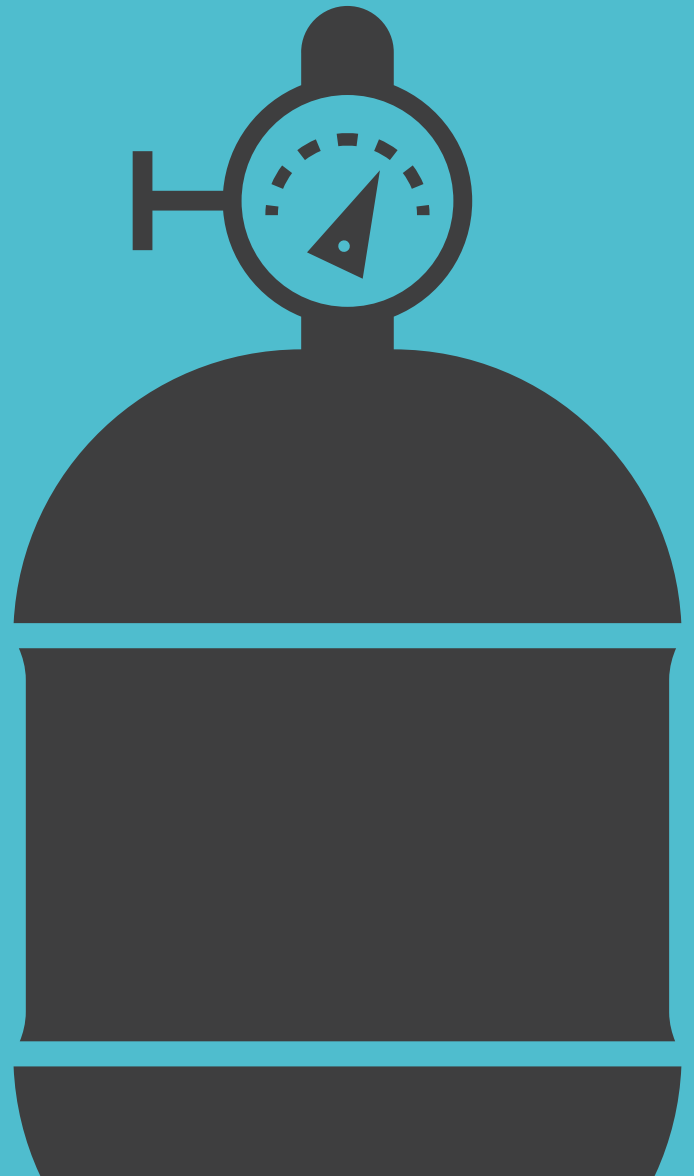


# Compressed Air Systems

A guide to the design,  
operation and maintenance  
of compressed air systems



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# Glossary

**Standard Temperature and Pressure (STP)** – 273k, 1013.25mb

**Normal Temperature and Pressure (NTP)** – 20°C, 1013.25mb

**FAD** - free air delivery is the actual quantity of compressed air converted back to the inlet conditions of the compressor. The units for FAD are generally measured according to the ambient inlet standard conditions Ambient temperature =20°C, Ambient pressure =1 bar abs, Relative humidity =0%, Cooling water/air =20°C and Effective working pressure at discharge valve =7 bar abs.

**CFM** - Cubic feet per minute – Also known as cfm or scfm (standard cfm), a measurement of the flow of compressed air.

**Dew point** – The temperature at which moisture in the air will begin to condense. At this point the relative humidity is 100%. Dew point is often used in pneumatics to represent the amount of moisture in a compressed air system.

**Pressure dew point** – The temperature at which moisture will start to condense back to liquid form. See dew point.

**VSD** - Variable Speed drive - An electronic drive using inverter to provide variable frequency control.

**ISO** - International Standards Organisation.

**BESN** - British Standard European Norm.

**PER** - Pressure Equipment Regulations.

**PED** - Pressure Equipment Directive.

**PSSR** - Pressure Systems Safety Regulations.

**COSHH** - Control of Substances Harmful To Health Regulations.

# 1 Purpose of the guide

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The use of compressed air is wide spread. In the UK however, few would realise that compressed air is the most expensive service on site, costing many times the price of electricity. Despite this, compressed air is often generated inefficiently and then subsequently wasted.

The cost of poorly designed or operated compressed air plant can be very significant - simply because the efficiency of air compressors is not high.

This guide explains, concisely, the basic guiding principles relating to the design, operation and maintenance of compressed air systems. The most efficient operation is achieved with combinations of design, procurement, operation and maintenance. This guide explains how compressed air systems can be designed and operated efficiently.

**1.1 Who is this guide for?**

This guide is primarily intended for companies who currently own, operate or will purchase a compressed system. Energy consultants who are examining and commenting on the efficient configuration of installations may also find this guide a useful reference. The guide covers a wide range of applications in varying degrees of detail and cites reference guidance for specific applications as appropriate.

N.B the guide is not intended as a definitive guide for the use of persons undertaking design who must assure, that in accordance with UK law, they have adequate competence and experience.

**1.2 What is the scope of this guide?**

This guide covers aspects of concept, design and the practical operation of a compressed air system. The guide does not cover specific installations or products other than by way of example.

**1.3 How to use this guidance**

This guide is split into stand-alone sections that may be read in isolation or in sequence.

<b>Guidance section 3</b>	Explains compression and compressor characteristics
<b>Guidance section 4</b>	Explains compressor control
<b>Guidance section 5</b>	Air Quality and the impact on cost
<b>Guidance section 9</b>	Summarise the key design opportunities
<b>Guidance section 10</b>	Summarise the key maintenance opportunities

**1.4 Additional sources of guidance**

This guide contains a list of additional sources of guidance.

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## 2.1

### The headlines

- Typically 10-15% of site energy consumption is used for compressed air.
- Typically 5-10% of that is wasted as leakage.
- Capital cost is less than 25% of total life operational cost.

## 2.2

### Strategy for energy efficiency

To drive efficiency up and business energy cost down, it is important to have an effective energy management strategy. A strategy that addresses energy management at design, procurement, construction and operational phases for all plant, process and indeed buildings. An effective management strategy will incorporate MM&T (Metering, Monitoring and Targeting – refer to the relevant Invest NI Guidance).

## 2.3

### Holistic design

With any plant or process, it is essential to adopt a holistic approach to energy management and to consider the efficiency of individual system components, as well as the whole system. That applies to compressed air where individual components can limit system performance or detrimentally influence the operation of other components.

System improvements must always be carefully evaluated, lest these also have detrimental effect on the performance of other components. Whilst it may be possible to make changes to individual components e.g. to replace the pipework, valves fittings, dryers etc. – these may have an impact on the system as a whole.

The best way to save energy and cost is not to use it in the first place. Demand management is usually the single biggest opportunity for most customers. Reducing the compressed air consumption by ensuring that compressed air is only used for essential purposes and by reducing leakage, can often represent the best energy saving opportunities.

## 2.4

### Retrofitting

In retrofitting or upgrading an existing system, it will often be possible to modify components to an existing system. For example it may be possible to reduce compressor cycling by fitting a larger accumulator. This guide will help you understand the changes that can potentially be made and the influences those changes may have on other system components.

It is often the case that small changes in operational practice, can cumulatively make big changes in energy efficiency.

Design modifications should only be undertaken by an experienced compressed air engineer. Understanding the technology will allow you to make informed choices about upgrade or replacement and to prioritise operational and maintenance activity.

## 2.5

### New design

Careful specification of new plant can have a marked influence on the operational life cost of plant. It is essential that new plant is procured in accordance with energy efficiency best practice in mind. That will mean spending more capital but will reduce the life cycle cost to the business. For example the cost of a PLC based control and VSD driven compressors, may be twice that of a conventional layout – But the life cost savings will dwarf the additional capital cost.

# 3 Compressing air

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**3.1 Pressure?**

Air is actually a mixture of gases, principally those shown below.

Gas	Molecular Weight	Proportion by volume
Nitrogen	28	78%
Oxygen	32	21%
Argon	40	0.9%
Water Vapour	18	0-4%
Carbon Dioxide	44	0.035%

The molecules of these individual components have differing weights and by virtue of their movement at any given temperature, they exert a force correspondent to the molecular weight and % composition. At sea level this mixture generally exerts a total pressure (force/area) of 1013 millibar (mb).

Each individual gas in the mixture exerts a component of that total pressure (a partial pressure) by virtue of the weight and movement of molecules of that individual gas. The arithmetic addition of all the partial pressures results in the total pressure as measured for example by a pressure gauge.

If the mixture is squeezed into a smaller space then the pressure increases with reducing volume and increasing temperature. The energy added by compression can be stored as compressed air and released when the air is expanded to provide mechanical movement.

This is particularly useful in mechanical applications where direct electric actuation is difficult, dangerous or in harsh operational environment or where there are uneven or varying pressure demands e.g. the operation of hand tools like wrenches, grinders, sanders etc.

Unfortunately the process of compression is not particularly efficient and the water vapour in air presents a problem as is addressed in this guidance. Moreover storing and transmitting compressed air often results in significant losses.

**3.2 Compressor types**

The differences between compressor types and particularly the performance at varying speeds and part loads are clearly misunderstood by many – often resulting in the ill-considered recommendation for the retrospective installation of VSD. So it is worthwhile dwelling on the operation of these types and machines. Compressors may be of two different generic types.

- Positive displacement machines
- Dynamic compressors

Positive displacement compressors trap a volume of air in a space and then decrease the volume of that space. Examples include reciprocating, screw, rotary slide or vane and scroll compressors. Scrolls are used principally in refrigeration applications and are not covered in this guidance.

Dynamic compressors work in the same way a fan does by imparting momentum to air and creating flow so the dynamic pressure (pressure due to directional flow) is increased. When that air is slowed down, the dynamic energy is converted into stored or static pressure. Examples of this type of compressor are the axial compressor (the first stages of a jet aircraft engine) or a centrifugal compressor. Axial compressors are not common, largely because of limitations associated with the effective pitch and span wise (radial blade direction) flow at speed and pressure differential from blade root to tip and stall characteristics.

**3.3 Positive displacement compressors types**

There are four main types of compressor

- Piston compressors
- Screw compressors
- Rotary or sliding vane compressors
- Lobe (roots type) blowers

These are all positive displacement compressors and they are all capable of being driven with variable speed drives, although other methods of capacity control are sometimes preferable. The power consumption does not reduce as a cube function with motor speed and generally the efficiency of compression reduces with operating speed. The roots type blower is not considered in this guide - for these generally are for specialist aeration or combustion related applications.

There are other types of compressor but the use is limited and this guide addresses the common types

listed. These compressors have different characteristics and the use depends on the application, pressure and the flow profile.

### 3.4 Reciprocating compressors

The piston compressor remains a popular choice for small or high pressure applications but the reciprocating compressor is largely superseded by the screw compressor (not least a situation driven by manufacturers). Generally reciprocating compressors are now only used for applications up to a few tens of e.g. 22kW. Piston compressors are positive displacement, meaning that a quantity of air is trapped and displaced. Most compressors are single acting (compress only in one stroke direction) and may incorporate one or more pistons arranged in a V-bank, either to provide capacity or to afford the incorporation of two compression stages (usually with intercooling to be discussed).

The compressor is similar to a car engine in that as the piston descends an inlet valve opens drawing in gas from the suction line, the piston reaches the bottom of the stroke and the inlet valve is closed, the piston then proceeds up the cylinder reducing the volume in the cylinder and compressing the gas at or near the top of the stroke depending on valve arrangements and settings, the exhaust valve opens discharging the compressed air to the discharge line.

The operation of a piston compressor is typically a variable volume ratio operation. That is to say the piston stroke to discharge valve opening point will vary depending on the system pressure and valve preload settings.

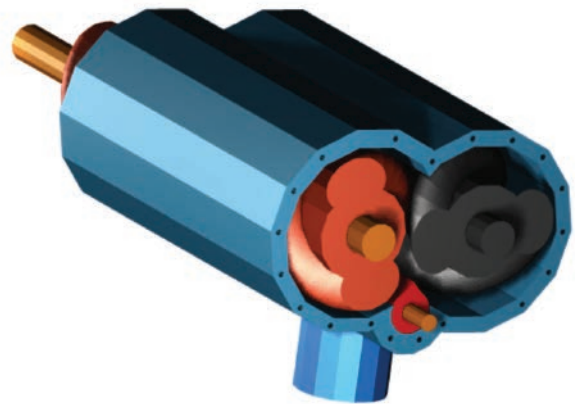
Some smaller reciprocating compressors are generally not designed for continuous operation. These have simple lubrication or dry systems and they have a design duty cycle. In simple terms they have a limitation on continuous loaded use and must be operated or unloaded intermittently, so careful sizing is required to ensure that the intermittent output can be buffered to meet the actual demand.

### 3.5 Screw compressors

The screw compressor is also a positive displacement machine. Air is drawn in through an intake port and trapped between lobes of two intermeshing screw rotors. The intermeshed volume decreases along the rotor length and the air is progressively compressed until discharge. The rotor length is fixed and thus the compression ratio (volume ratio) essentially fixed,

regardless of speed. The variation of speed does not change the volume ratio or lift in pressure.

Because the rotor length and hence the compression ratio is ostensibly fixed, the screw compressor is designed with an operational pressure in mind e.g. 7.5barg. The performance of the compressor in terms of kWh/volume compressed will vary with the operating pressure that the compressor must operate against. Accordingly it is usually best to operate a screw compressor at the design pressure. For other operational pressures, there will be a separate performance characteristic.



In order to overcome some of the issues associated with demand management, two variants have been derived. The first is to use VSD control. This works very well because the compression ratio is not diminished. However as the screws slow the momentum is lost and air leakage between the rotors and the case becomes a significant factor. The ratio of leakage at low rpm and the inverter losses, together with under speed motor losses combine to reduce efficiency significantly and screw compressors are not efficient at low speeds. Thus in using a VSD it is best to ensure that these are not used for very small loads and operation is maintained in the 60-100% range where at all possible and certainly not be used below say 40% of design load (subject to examination of the specific manufacturer's performance curves).

Because air demand changes it is necessary to change the output of the compressor. In fixed speed machines this is achieved typically by throttling the inlet. Most machines are equipped with load /no load inlet throttling which means effectively a damper is closed across the inlet and the machine cannot draw air or deliver air. There is an allied power reduction but the off load power is some considerable % of the normal operational power e.g. 35% or more.

Sliding ports, turning ports or poppet type valves may be used to either restrict the inlet condition or effectively reduce the screw length over which compression takes place (reduces compression ratio) are less common. If the screw length is reduced then the outlet pressure reduces.

In most cases the compressor cannot simply be switched off because there will be a restriction on the number of motor starts that can be allowed without damaging the electric motor. Thus in the simple screw application the airflow to the compressor intake is successively restricted to effect capacity control, switched in response to a pressure set point and control differential.

Oil injected screws are able to work at higher pressures for the oil provides good seal between rotors and the casing (particularly dealing with end float). Compression ratios of 7:1 are possible with a single stage compression. The oil injected machine is also cooled by the injected lubricant. Because there are fewer moving parts, seals and tolerances are greater, the oil injected screw is the lowest cost machine to manufacture. The compression efficiency performance is nearly as good as a double acting reciprocating compressor. This is the most common kind of screw compressor. Oil separator performance should be able to achieve 3-15ppm.

In dry screw compressors the rotors do not actually touch and are geared externally. The dry screw compressor has to work at very high speeds to maintain the directional airflow through the unit and operating temperatures are higher. To achieve medium pressures two compression stages are often used. To improve the volumetric efficiency of the second stage, the air exhausted from the first stage is cooled at pressure to increase the density at second stage suction. Air leaving the compressor is cooled with an after cooler.

Because the dry screw has to be geared and has to work without rotor lubrication the tolerances are much tighter. The costs are generally much higher-particularly where multistage units with inter and after cooler are deployed.

- Meeting low loads with VSD is not an efficient option
- Power consumption does not reduce as a cube function with speed
- Linear reduction in top 30% only
- Below 70% fixed operational losses are high

- VSD screws should not be operating below say 60-70%
- VSD is lower cost than slide or turn valves but is not a substitute.

### 3.6

#### Rotary vane compressors

The rotary vane compressor (sliding vane compressor is also a positive displacement machine.) An eccentric rotor with sliding vanes rotates within a fixed rotor housing. As the rotor revolves the sliding vanes maintain contact with the rotor housing progressively reducing the volume from intake to discharge.

Again the rotary vane compressor can be speed controlled to an extent. But rotary vane operation relies on speed to hold the vanes in tight contact with the housing and there are speed limits to the range of speeds and pressures these can operate at.

Oil free vane compressors are restricted to low pressures, because the lack of lubrication results in higher operating temperature and sealing difficulty. Oil injected machines form an excellent seal between vane and rotor which improves with slight wear (as the vane tip profile wears to match the rotor housing).

### 3.7

#### Centrifugal compressors (Turbo compressor)

The centrifugal compressor is used for larger volume medium pressure (7.5 barg) air flows and works in a similar way to the turbocharger on a car. Air is drawn into the central eye of an impellor and is accelerated through the impellor vanes. The air compresses as it passes through the impellor vanes and increases in speed. As the vanes impart energy from the very high rotational speed, the air flow gains kinetic energy. The kinetic energy is subsequently converted from velocity to pressure in a diffuser stage.

These compressors may be considered as complex fans. The operation does follow the affinity laws and thus a reduction of speed is associated with a cube power reduction. However, the pressure drop induced with speed reduction has to be considered because the machine is not positive displacement, a reduction in speed can only be tolerated with a reduction in pressure – simple capacity control is not possible. For any given rotor speed there is a maximum tolerable pressure differential before the compressor will stall (high pressure surges backwards through the fan). Surge Stall is accompanied by very undesirable flow and vibration which can damage or destroy the compressor. Most compressors are therefore fitted with anti-surge valves



which will bleed high pressure from the discharge side in a controlled fashion when the pressure differential is becoming too high. The pressure differential will become too high if there is a sudden and significant reduction in compressed air demand.

At the other end of the scale if the system pressure is reduced (and the rotor is spun at maximum RPM) the airflow will get faster and faster until the flow becomes supersonic at some location within the impellor. At this point the maximum volume flow is reached, and the compressor cannot produce more flow.

The centrifugal compressor is not a very flexible machine and is most effective when running at or near to full capacity. These machines are better suited to situations where there is a large stable base load and where a VSD screw (operating at constant pressure) within the centrifugal machines operating range, can be used for top up. However the multistage centrifugal compressor is much more efficient than the screw and similar to the most efficient (double acting) reciprocating compressors.

### 3.8 Range and application

Compressor type	Application	Range (nm <sup>3</sup> /hr)	Performance (kWh/nm <sup>3</sup> )	Comments
Reciprocating	Smaller scale variable loads or high pressure applications.	2-25 25-250	14.2 11.8	Good part load efficiency with start stop or unloading and step controlled multiples
Oil free reciprocating	Smaller scale variable loads or high pressure applications where clean air is required.	2-25 25-250	15.3 13.0	Good part load efficiency with start stop or unloading and step controlled multiples
Oil injected screw/ Vane	Ubiquitous use - air is cleaned with oil separation and thus suitable for most applications.	2-25 25-250 250-1000	14.2 12.4 11.2	Good turndown upper ranges with VSD poorer efficiency at low turndowns suitable for pressures up to typically 8.5barg
Oil free screw	Suitable for lower pressure applications e.g. 3.5 to 4 barg unless multistage, intercooling and typically after cooling required.	2-25 25-250 2-25 25-250 250-1000	12 10.6 10.6	Good efficiency at larger size restriction to stage operating pressure (multistage with cooling possible)
Centrifugal	Suitable for medium pressure applications e.g. 6-8 barg and stable base load operation.	250-1000 1000 – 2000 2000+	12.4 10.6 10	Best efficiency where the plant is operated near to full capacity and in the allowed modulating range. Relatively inflexible



### 3.9

#### Compressor sizing

Compressor sizing is not straightforward – the golden rule is not to end up with an under loaded compressor – particularly an under loaded screw compressor (VSD driven or not). A screw compressor that is working way down the range will have substantially higher kWh/m<sup>3</sup>/hr than when operating at higher outputs.

Generally constant speed machines regardless of type are best operated near or at full load. Reciprocating compressors achieve best efficiency at slightly less than full load (because there is a small volume of air that gets compressed every stroke but cannot be cleared through the valve and re-expands each stroke). If the compressor is sized to meet peak load it is likely to operate inefficiently at part loads – However if the machine cannot meet peak flow then the pressure will drop at periods of high demand.

Using multiple compressors affords capacity and efficient turndown whilst also allowing for a degree of back up, planned maintenance and so on. However again efficiency has to be considered for (with reference to the table in 4.3) the part load efficiency of all compressors reduces with size. Moreover the capital cost has to be considered.

The demand may be estimated by preparing an inventory of the air consuming plant, equipment and tools and applying a diversity factor to account for intermittent use (a load factor). Thus the peak flow x the load factor for each item of equipment will derive an expected airflow and this can be summed to provide a total expected air flow. It is necessary to allow for leakage and leakage should be assumed as 10% (although ideally it should be a lot less).

Where manufacturer's data is not available then it will be possible to assess the likely air demand of tools or plant.

# 4 Compressor control

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## 4.1

### Single compressors

#### 4.1.1

##### Start/stop control

At the very simplest end of the scale with single and relatively small compressors, start stop control might be employed. A pressure switch responding to discharge pressure (likely to be receiver pressure in a simple small system) controls the current flow to the electric motor and the compressor switches in and out in response to the pressure. The receiver buffers demand somewhat but the pressure will oscillate and the mean pressure setting must be sufficiently high so that the lowest system pressure is not too low for downstream equipment. This type of control is adequate for small reciprocating systems where the pressure is not critical. The compressor size must be selected so as to prevent excessive compressor cycling or else the motor will fail.

A variant of this (which reduces the number of motor starts) is to simply unload the compressor by shutting the inlet valve (in the case of the screw compressor) or holding the inlet valve open in the case of a reciprocating compressor (preventing compression). Larger older reciprocating compressors required unloading to arrange capacity control and the load/unload transition was evident by the sudden change in noise, line current and vibration as the cylinders were loaded. Load unload is used frequently with low end oil injected screw compressors and again the load/unload can be detected by rising pressure, noise and the sudden increase in line current (where monitored).

#### 4.1.2

##### Modulating control

Fully modulating control can in the case of the screw/vane be achieved by partially or wholly throttling the inlet, which reduces the power somewhat but typically to no less than 35% of full load power. In some cases the power consumed by rotating vane will not fall below approximately 50% full power. Note that whilst the main power reduction may be broadly linear until internal gas leakage destroys efficiency, the total power is not linear because of all the auxiliary loads associated with cooling fans, internal dryers, inverters etc. So whilst capacity control can be achieved the power reduction is not as dramatic.

In the rotary vane compressor, the inlet throttling is typically accomplished by oil servo operated valve. As system pressure rises, that pressure is sensed and used to regulate the flow of pressurised oil to a valve

servo mechanism. The servo progressively closes the inlet damper causing restriction and thereby a reduced pressure at suction with attendant flow reduction. This typically offers a capacity control to 40% output but at the expense of increased compression ratio (although flow is reduced) and the combination does not allow significant power reduction.

Shutting inlet valves on a reciprocating compressor simply results in a fixed volume of air being repeatedly squeezed with attendant rise in temperature and is not an efficient way to control capacity on this type of machine.

A much more effective (if not expensive and relatively unusual) means of capacity control is variable displacement. The screw length is effectively altered by varying the location of the inlet or discharge port using a slide valve. This reduces the compression ratio with a commensurate and broadly linear proportional reduction in power consumption. The arrangement achieves something similar to VSD but the compression ratio is reduced.

#### 4.1.3

##### Variable speed drive (VSD)

The variable speed drive provides reduced volume at ostensibly the same compression ratio (and thus discharge pressure) for screws and to a less widely adopted extent reciprocating and vane compressors. The VSD is not a panacea for performance. In the lubricated screw compressor the VSD can provide efficient modulation above say 50% design capacity. But where a screw is turned down to meet a very low load the performance will be poor.

Essentially load/offload control results on a continuous under and overshoot of the setpoint pressure whereas continuous modulated control it is possible to track the desired setpoint pressure with tight control – the latter being generally more efficient subject to the variances in compressor type.

## 4.2

### Multiple compressor control

Because, generally speaking, compressors of all types perform best when well loaded, it is better to use multiples of compressors to meet loads with large load swings where a single large compressor would otherwise be underloaded. Therefore, in a typical compressed air system there will be several compressors. The combined output of these machines is intended to meet the maximum system demand (notwithstanding any storage to cover short duration

peaks in demand).

Control is required to vary the contribution from one or more compressors to match the prevailing demand. Conventionally compressed air systems are designed to operate within a fixed pressure range (control range) around a setpoint or desired pressure. Very typically that pressure is measured at the compressor (which may not have any bearing on pressures experienced by a distant user). When the system pressure falls below the setpoint value by a predetermined control value below the setpoint value the compressors will be brought on line until the pressure reached the upper control value somewhere slightly higher than the setpoint value. The control bandwidth might typically be 2-20psi and thus a system with a set point of 7.5barg may work between 100psig and 110psig – these being the cut in and cut out pressures and the difference being the control bandwidth.

To arrange for the staggered operation of multiple compressors using simple pressure differential controllers requires that the set points and bandwidths are staggered or otherwise arranged to ensure maintenance of pressure. Depending on the relationship of compressor capacity to system volume and demand that arrangement can lead to substantial degree of pressure swing as compressors cut in and out to service individual setpoints and control bandwidths. To meet minimum pressure requirements the pressure will generally have to be set higher than actually required and reduced in the system for consumption. The use of accumulation or compressed air storage is generally required to manage the compressor cycling and to maintain a degree of pressure stability.

#### 4.2.1

##### **Differential pressure arrangements**

Accurate control with relatively simple differential pressure arrangements can be difficult. Typically the manufacturer will set up compressors based on discharge pressure (individual compressor discharge pressure).

Using a simple cascaded pressure control, the staggered set points for individual compressors allows the sequential loading of compressors in response to falling pressure, and the shutdown of successive compressors with rising pressure. Where the pressure sensing for each machine is at the discharge of the individual compressor (which is typical), then the influence of fittings to the point of common connection

has to be taken into account so as to ensure the pressure “seen” by each compressor is an accurate reflection of the common and desired output and that compressor influence is balanced – or the well staggered bandwidth may not work quite as intended.

With simple pressure sensing that problem may also manifest where pairs (or multiples of compressors) of compressors are operated as base load and one to modulate using inlet valve. However depending on the pressure drops to point of common supply, and where the compressors have individual driers, filters etc. then supplying common header and receiver, the modulating compressor may drive the system pressure (by virtue of lowered pressure drop at reduced flow) to a point where the base load machine cuts out long before the modulating machine is carrying the full load. The result is that the pressure then drops rapidly and the base load compressor re-loads. The control bandwidth for modulation must be narrowed to allow effective base and modulated operation. With reduced control bandwidth the receiver volume may also have to be increased.

Cascaded pressure control, is governed by differential pressure switches incorporating cut in, cut out, and setpoint pressure value. However, this can equally be managed by simple electronic controller, with the additional benefit that the lead compressor can be rotated to avoid excessive wear on any specific machine.

The problem with using cascaded pressure control is that between the first compressor engaging and the last compressor dropping out the total staggered bandwidth is large and as a result there is a significant variation in system pressure. Where the air pressure is critical for equipment that means that the system pressure is generally set higher and dropped to the desired end user pressure. An undesirable situation since higher pressure is less efficient with allied energy and cost.

Considering the pressure /flow characteristics of the turbo compressor (refer to 3.7), cascaded pressure control would be unsuitable (flow increases with falling pressure).

The cascade arrangement does not take account of the relative efficiency of compressors at part load and the arrangement would not work for VSD compressors unless modified to suit.

Where combinations of compressors are to be deployed, an alternative solution is to use fixed speed machines to meet base load and smaller machines to meet loads in excess of the base by setting the control bandwidth for the smaller compressor inside that of the base load control bandwidth.

Here if the load is small the trim compressor will carry the load, however if the load increases and results in a pressure decrease to below the trim cut in the main base load, compressor will cut in and the flow met by both compressors. If the load is not sustained the pressure will rise and initially the trim compressor will be stopped and very shortly thereafter the base load machine. As the pressure falls again the trim compressor will be restarted first to give it the opportunity to meet the load. This strategy will work for both fixed and VSD trim compressors but it is important to ensure that the trim compressor is sized correctly for it is pointless running an oversized trim compressor on the VSD (the specific power consumption will be poor) or under sizing the base to the extent that is frequently switched.

With a VSD trim that control will allow the trim unit to service demand above the flow capacity of the base compressor.

**4.3 Demand profile**

The demand profile is critical to understanding the combination and disposition of compressors that should be used.

The important factor is to ensure that in selecting the compressor sizes and the control strategy that the compressors are working, insofar as is practically possible with a low specific power consumption.

Note that the cost of VSD compressors is not significantly more than a fixed machine and there is really no requirement to use a fixed speed machine unless the type prevents use of VSD. The table to the right summarises.

Potential Compressor solution	Example Load
Single large compressor, VSD not essential but little extra cost.	<p>A line graph with 'Demand m3/hr' on the y-axis (0 to 80) and 'Hours' on the x-axis (1 to 23). A blue trapezoidal area represents demand, starting at 0 at hour 5, rising to 70 at hour 7, staying constant until hour 17, and then falling back to 0 at hour 19.</p>
Potentially 2 similar sized compressors, VSD not essential but little extra cost.	<p>A line graph with 'Demand m3/hr' on the y-axis (0 to 120) and 'Hours' on the x-axis (1 to 23). A blue rectangular area represents a base demand of 50 m3/hr. An orange trapezoidal area is stacked on top, starting at hour 5, rising to 100 at hour 7, staying constant until hour 17, and falling back to 50 at hour 19.</p>
Potentially 3 similar sized compressors, or 1 base and 1 larger VSD providing VSD is not continuously operated at low load.	<p>A line graph with 'Demand m3/hr' on the y-axis (0 to 100) and 'Hours' on the x-axis (1 to 23). A red rectangular area represents a base demand of 30 m3/hr. Three green trapezoidal peaks are stacked on top, each peaking at 90 m3/hr. The peaks occur between hours 4-7, 10-13, and 16-19.</p>
Large base +top up VSD or single large VSD if operated in efficient range.	<p>A line graph with 'Demand m3/hr' on the y-axis (0 to 120) and 'Hours' on the x-axis (1 to 23). A blue rectangular area represents a base demand of 70 m3/hr. Multiple orange trapezoidal peaks are stacked on top, each peaking at 100 m3/hr. The peaks occur between hours 5-7, 9-11, 13-15, 17-19, and 21-23.</p>

*Courtesy of the original energy efficiency best practice guidance and updated appropriately.*

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#### 4.4

##### **PLC based Intelligent Control**

The problems associated with operating differing sizes of compressor at optimal efficiency (lowest specific energy consumption) can be best overcome using an advanced PLC based controller – There are many of these products on the market.

These controllers will allow the programming to account for the variation in specific compressor performance at part load, enabling prioritisation for specific machines if required.

Moreover, the controller may be equipped and configured to examine pressure and flow at remote points in the distribution system and accept the input from individual compressor sensors, current transformers.

By examining, for example, the rate of change of pressure and or flow, PLC control allows the accurate and optimal control of multiple machines, ensuring that compressors are not cycled unnecessarily in response to system pressure change.

Where remote compressors are connected to the system these may also be controlled optimally ensuring pressure balance in the system and optimised duty for all compressors.

Additional functionality might typically incorporate the control of zone valves and or timed operation or timed pressure scheduling (so as to allow pressure reduction for a night time or weekend operation if pressure requirements are lower).

Given that 75% of the life cost of a compressor will be the electrical power consumption - investment in a good control system would be minimum best practice.

# 5 Air quality

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**5.1 Air contamination**

The air that is compressed is simply fresh air drawn into the compressor contaminants and all. These contaminants might include water, oil, smoke, solvents, bacterial contamination and any other airborne particulate. Particularly in the context of food preparation or electronic manufacture, the contamination of food preparation and clean rooms may be well considered and prepared – but the compressed air supplied to these facilities must also be clean.

The air intake, location, cleanliness, temperature and other factors do not only have bearing on air quality, but also on compressor wear and power consumption. (The impacts on performance are discussed in due course).

The compressor type may give rise to air contamination if for example lubrication or sealing oil is entrained but subsequently not separated effectively. Corrosion from the distribution system may unfortunately be entrained in the air flow.

The problem with all contamination is that it will cause blockage or malfunction of equipment or give rise to unacceptable health issues where used in the food or medical industries.

**5.2 Removing particulate before the compressor**

The air is filled with tiny particulate matter, dust, smoke etc. and in industrial areas this burden is higher. As the air is compressed the concentration of particulate is increased. Apart from the health issues that may be associated with the delivery of contaminated air, the concentration of particulate increases wear on the compressor and the end use equipment supplied from the compressed air distribution system.

Particulate must be removed to protect the compressor and the downstream equipment. The compressor intake filter should be rated to remove dust and particulate at typically 4-10 microns and rated to high efficiency e.g. 90-95%. For centrifugal compressors (Turbo compressors) the higher velocities involved require a higher standard of filtration and 0.2 microns is preferential. The larger particulate can be removed by filter and will assist in reducing wear on cylinder walls or screws/vanes or lobes.

Minimising the particulate is good practice and prevents wear but filtration does result in pressure drop so filters must be sized so as to present the least

pressure drop, cleaned and or changed regularly. It is important to maintain both the cleanliness of the compressors operating environment and the intake filtration. Additional intake resistance increases the specific power consumption required and intake energy penalties of up to 3% could manifest.

**5.3 Water contamination**

Air is made up of various gases, principally those shown below.

Gas	Molecular Weight	Proportion by volume
Nitrogen	28	78%
Oxygen	32	21%
Argon	40	0.9%
Water Vapour	18	0-4%
Carbon Dioxide	44	0.035%

At sea level this mixture generally exerts a total pressure (force/area) of 1013 millibar (mb).

Each individual gas in the mixture exerts a component of that total pressure (a partial pressure) by virtue of the weight and movement of molecules of that individual gas. The arithmetic addition of all the partial pressures results in the total pressure as measured for example by a pressure gauge.

The **saturated vapour pressure** is the partial pressure of the gas in question (water in the form of steam) at the point when the air is saturated with water vapour and therefore cannot hold any more without condensation occurring. If the partial pressure of water vapour approaches the vapour equilibrium pressure then the air reaches saturation (it cannot hold any more water) and condensation will occur. The dew point has been reached.

The saturation pressure is a function of temperature and for any given temperature there is a correspondent saturation pressure. So for example if the intake air has a dew point of 5°C (and therefore 9mb partial pressure) compressing the air to 7.5barg would raise that to 67.5mb where the dew point would be approximately 40°C - water would begin to condense from the system at 40°C.



The intake air will unavoidably (unless pre dried) contain water vapour. That water vapour will condense within the compressed air system and must be removed either by drying or by trapping (to be discussed in due course).

The amount of water entrained is significant, for example if a 100nm<sup>3</sup>/hr machine was drawing air at 21°C and 70% rh the total water drawn into the system would be 31 litres. Over the space of a few days that will quickly fill a small receiver. Unless the water is removed from the system it will in part, be transmitted to the points of use potentially damaging equipment, controls or contaminating end products e.g. paint in spray paint guns etc. Condensing water may corrode pipelines, receivers and if the temperature is sufficiently low on external pipework, the water may freeze causing blockage and damage.

To remedy this problem and to cater for air qualities required by industry it is common practice to dry the air to a sub- zero dew point, e.g. class 1 (-70°C) class 2 (-40°C) and class 3 (-20°C). Typically -40°C is required for food industry.

Drying is expensive so air should only be dried to the standard that is absolutely necessary for the production activity. Desiccant drying is required to achieve sub-zero pressure dew points.

**5.4 Removing oil liquids and vapours**

Eliminating oil from the airstream may to a large extent be determined by compressor choice. Where the oil injected screw, vane or a reciprocating compressor are used there will be some oil entrained in the air and carried from the compressor. The amount of oil carried over varies from type to type and manufacturer but even a relatively small screw compressor might add 20-30 litres of oil to the air system over the period of a year's operation. This is added simply from the need to inject oil and not the result of any intake contamination. Compressors operating in a harsh or contaminated environment will draw whatever contamination they are exposed to.

Removing the oil is achieved using a coalescing filter, typically a post compressor, replaceable cartridge type filter. This type of filter will remove solids and coalesce liquid droplets. For food applications, the residual oil content should be reduced to less than 0.007ppm. For additional reduction, an activated carbon filter can be used. To achieve class 1, a combination of cartridge coalescent filter and Carbon filtration will be likely.

**5.5 Standards for quality**

Compressed air is used in so many different applications it has become necessary to define standards for quality and for quality testing.

The most relevant and important of these in the context of resource efficiency is the ISO (international Standards organisation) 8573 series comprising parts 1 through 9, where:

Part	Specified parameters
Part 1	total allowable contamination per unit volume
Part 2	test methodology for aerosol content
Part 3	test methodology for humidity content
Part 4	test methodology for particulate content
Part 5	test methodology for oil vapour and organic solvent content
Part 6	test methodology for gaseous contaminants
Part 7	test methodology for micro biological content
Part 8	test methodology for solid particulate
Part 9	test methodology for liquid content

The requirements are there to ensure quality for air in food, medical, precision and many other applications.

There is a cost benefit analysis to be undertaken because air quality costs money. The benefits of high quality air are.

- Reduced equipment and plant maintenance (less wear and tear).
- Less breakdown with allied production cost.

On the other hand

- Filtration and drying capital and operating cost.
- Filtration increase pressure drop with allied energy cost.
- Filters require replacement with associated cost.

For air quality classification the contaminants are determined as three groups as illustrated in the table to the right.

**5.6**  
**Air Quality Standards (ISO 8573 2010)**

ISO Class	A - Solid Particulate			B - Water		C - Oil	
	Max No. of particles per m <sup>3</sup> air			Concentration mg/m <sup>3</sup>	Vapour pressure dewpoint at 7barg	Liquid g/m <sup>3</sup>	Total aerosol and liquid vapour mg/m <sup>3</sup>
	0.1-0.5 micron	0.5-1micron	1-5micron				
0				--		-	
1	≤20,000	≤4,00	≤10		≤-70oC	-	0.01
2	≤400,000	≤6,000	≤100	-	≤-40oC	-	0.1
3	-	≤90,000	≤1,000	-	≤-20oC	-	1
4	-	-	≤10,000	-	≤/+3oC	-	5
5	-	-	≤100,000	-	≤+7oC	-	25
6	-	-	-	≤5	≤+10oC	-	-
7	-	-	-	5-10	-	≤0.5	-
8	-	-	-	-	-	5	-
9	-	-	-	-	-	5-10	-
X	-	-	-	>10	-	>10	>10

The specification for air is denoted by the ISO class required for each of Particulate, Water and Oil. So if the specification was 2.1.1, then the solid particulate content would be as highlighted above (these standards are onerous but might well be required for Pharmaceutical clean air systems or food and drink)

- Air quality costs money. Drying is particularly expensive and over drying should be avoided.
- Particulate filtration is cheaper to arrange and good pre and post compressor filtration is necessary to prevent compressor wear, oil contamination and for health and safety. For pharmaceutical applications then specialist pre and post drying filtration will be required. Additional point of use filtration may be required.
- The essential factor is consultation during procurement.
- System maintenance and regular filter replacement (including air intake filtration).

# 6 Drying compressed air

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This section of the guidance considers the differing methods of drying air and allied best practice.

The water burden within a compressed air system can be significant considering that on a summer's day when the air temperature was 21°C and perhaps 70% relative humidity, each m<sup>3</sup> of air drawn into the compressor will be holding 12.8g of water. Thus for every 100m<sup>3</sup>/hr flow the charge of water is 1.28kg, which has to be removed from the system. Because the pressure dew point is higher as pressure increases this water will condense from the air and fill all points at low level e.g. receivers. With oil carry over this will form an emulsion.

### 6.1 After cooling

Because, the process of compression involves the addition of energy, the air leaves the compressor at an elevated temperature. In some cases this can be well in excess of 100°C. Cooling the air after compression serves useful purpose for it:

- Reduces the moisture level in the compressed air
- Increases the system capacity
- Protects filters/dryers from excessive heat

The higher temperature of the compressed air increases the capacity to hold water. The relative humidity is reduced and the dew point is raised. If the air is cooled after compression to say 35°C then a significant amount of the water (typically 75%) will condense out in any wet receiver. After cooling (sometimes incorporated in the compressor air to air exchanger) is limited by the ambient conditions prevailing (in the plant room) which is why, amongst other things, it is important for the compressor to draw cool, clean air. The process of condensation affords some oil and particulate removal. The water is separated from the air using a separator which may be incorporated with the compressor or fitted as an in line separator. The separator will typically incorporate some form of momentum separation (e.g. cyclonic separation) and additionally a trap to discharge separated water.

The wet receiver should therefore be located somewhere cool, preferably outside so as to promote further cooling and to facilitate the additional drop out of water in the cooled air. However, it is important to protect against freezing within the vessel.

### 6.2 Receiver condensate discharge

A reduction in receiver volume is debilitating because it reduces the compressor cycle periods and limits the buffered capacity. To prevent corrosion, carry over and to ensure correct function it is necessary to remove condensate that has accumulated in the receiver.

“Cracking” a manual drain valve or a timed blowdown with solenoid is not best practice. The cost of these exercises is prohibitive. The receiver should be fitted with a drain and a proprietary trap. The type of trap should be selected for the nature of the condensate – if a high viscosity emulsion is prevalent then the trap must be selected to suit.

Electronic condensate drain traps may not be satisfactory for very high viscosity liquids but are the preferred option for efficiency for these only operate on condensate detection and have very little if any air consumption.

Condensate has to be disposed.

### 6.3 Dryer types

There are four main types of dryer:

- Refrigerant dryers
- Desiccant Dryers
- Deliquescent
- Membrane

#### 6.3.1 Refrigerant dryers

Use a small direct expansion refrigeration system to cool the airflow causing the water in the airflow to be condensed out. The water in condensing gives up latent heat and this is removed by the refrigerant. The cooled compressed air flow is used to precool air entering the dryer. Heat from the refrigerant circuit is dissipated by the system condenser.

The main disadvantage of the refrigeration system is that these cannot cool the air flow below zero or the water will freeze. However the dryer type is the common industrial choice for relatively low spec airflows (refer to section 6) and is very robust, e.g. not susceptible to oil contamination.

The advantages of the refrigerant dryer are

- Low initial capital cost.
- Low operating cost.
- Low maintenance costs.
- Not damaged by oil in the air stream.

The principal disadvantage is that a sub-zero dew point cannot be achieved.

### 6.3.2

#### Desiccant dryers

A desiccant is a material that has a physical affinity for water (silica gel for example has an affinity for water and is a desiccant). The desiccant dryer relies on a column or bed of desiccant material through which the compressed air is passed.

As the airflow passes through the bed /packed tower the water is drawn to adhere to the desiccant material and the dried air passes out the desiccant dryer. The desiccant eventually becomes saturated and the water adsorbed must be driven off by regenerating the desiccant bed. The dryer can then only operate for a specific period of time before becoming saturated and in practice two or more desiccant beds are required to allow continuous operation. The airflow is swapped between towers as the water content is measured or on a timed cycle based on duty.

To regenerate the dryer beds a flow of purge air can be used but this requires a fairly high air volume (it is the reverse of the desiccant process and drying airflow can be 15-20% of the total airflow). Therefore, this is an expensive process. The amount of purge air required can be reduced by heating the air steam, (sometimes electrically), so as to allow a higher supported water burden and affording a 50-75% reduction in purge air depending on the exact design of the regeneration system.

Waste heat from the compression process can be used in what is termed waste heat regenerative drying and this substantially reduces the heat used for regeneration.

To achieve the contamination standards required for chemical, pharmaceutical and electronics industry (some food applications) regenerative desiccant drying is essential to achieve the low dew points required.

The principal advantage is the fact that this type of dryer can achieve such a low dew point. The air is sufficiently dry to decrease microbiological growth and reduces the risk of pipeline corrosion.

The disadvantages of desiccant dryers are

- High initial capital outlay.
- The desiccant material must be replaced from time to time.
- The desiccant is usually sensitive to oil , grease and particulate so a range of pre-filtering must be used.
- There is an air load associated with operation.

### 6.4

#### Deliquescent dryers

These work in much the same way as the desiccant dryer but the desiccant actually absorbs the water. The diluted absorbent is discharged from the dryer and there is therefore a liquid effluent. Deliquescent Dryers will typically provide a  $-6.6^{\circ}\text{C}$  to  $-3.8^{\circ}\text{C}$  dew point reduction below the compressed air temperature entering the dryer. Deliquescent dryers might typically be used in applications such as sandblasting. They are cheaper than desiccant dryers but the performance is not suitable for most industrial applications, there is a cost for absorbent and a cost for effluent disposal.

The main advantage of the deliquescent dryer is the low capital cost.

### 6.5

#### Membrane dryers

Membrane dryers work in a similar way to a very efficient filter (and in a very similar manner to a reverse Osmosis unit). Because different gases have different molecular sizes they can be separated on the basis of molecular weight if a sufficiently good filtration medium is used. In the case of the cartridge type membrane filter water vapour is passed preferentially through a fibre membrane and air is retained as the core flow. The water coalesces and is drained away. A dew point of  $-40^{\circ}\text{C}$  can be achieved with some air cost.

The principal advantages of the membrane dryers are:

- The low cost.
- Low maintenance.
- No electrical power requirement.
- Intrinsic safety.

The principal disadvantages are:

- Total flow – Limited really to low flow or in line drying applications.

## 6.6

### Dryer selection

The type of dryer selected depends on the application and in turn;

- The dew point required.
- The pressure.
- The inlet temperature.

Each of the dryer types discussed above has a cost associated with the operation. On the one hand the sensitivity of the filter (desiccant types) will require efficient pre-filtration (and thus there will be an allied pressure drop requiring more compressor power) and on the other the application of external heating must be weighed against the cost of purge air. The table below sourced from GPG 216 (Energy Efficiency Best Practice Programme).

Dew point	Type	Pre-filtration	Extra Energy consumption
+10	Deliquescent	Not essential	1%
+3	Refrigeration	General	5%
-20	Membrane filter	High efficiency	25%
-20	Desiccant Waste heat regenerative	High efficiency pre and dust removal afterwards	3-5%
-40	Desiccant using electric or steam heating (heatless)	High efficiency pre and dust removal afterwards	10-15%
-40	Desiccant with purge air	High efficiency pre and dust removal afterwards	8-12%
-70	Desiccant with purge air (heatless)	High efficiency pre and dust removal afterwards	20+%

A decision regarding dew point is critical because over-drying the air could cost a very significant amount of money (in terms of energy and maintenance over the life of the compressors/Air system.)

The dew point requirements will dictate (to a large extent) the type of dryer that must be used. However the important factor is to consider the utilisation at low dew points and to determine if all air must be dried by the same amount. Clearly if it is one process or system that requires very dry air then it would be pointless drying the entire compressed air flow to the same onerous level and the use of local membrane drying might be satisfactory.

Thus depending on the application, separate and differing quality airstreams could be derived from a base air quality – a solution that would provide significant saving over a blanket approach and global quality constraint.

The older Energy Efficiency Best Practice Guide G216 is now, in parts, out of date – however the information contained in relation to air quality requirements for specific applications is invaluable – refer to <http://www.bcas.org.uk/files/pdf/6.pdf>.

## 6.7

### Filtering before drying (extracted from GPG216)

Depending on the type of dryer required then special treatment of the air will be required to protect the dryer.

#### 6.7.1

##### Solar Hot Water Storage

Prior to the dryer the air should be filtered using a coalescing filter element. Typically this will comprise of several layers of glass micro-fibre, which captures dirt particles, water aerosols, oil aerosols, micro-organisms and rust and pipe scale down to 1 micron in size (0.3 – 1micron). Typically the maximum remaining oil aerosol content should be less than 0.6 mg/m<sup>3</sup> at 21°C, with a low pressure drop e.g. <70mbar.

Coalesced liquid condensate collects in the base of the filter housing where it is automatically removed from the system by a solenoid operated / timed drain valve.

#### 6.7.2

##### Piping and wiring arrangements

If improved levels of oil removal are required, it is normal to fit a high efficiency oil removal filter. This will remove 99.999% of the particles down to a size of 0.01 µm, including water and oil aerosols, giving an air quality as good as can be delivered by an 'oil-free' machine.

### 6.7.3

#### **Controller Location**

The desiccant may be carried over into the airstream. To mitigate against duct carry over a filter is fitted after the desiccant dryer. These filters will typically offer 0.3 -3 Micron filtration and are generally cartridge type filters with a replaceable or reusable filter (can be polyester, polypropylene, cotton, fiberglass, Nomex, or paper). For higher temperatures fibreglass is generally used. These should be low pressure drop e.g. (7-70mbar).

# 7 Heat recovery

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## 7.1

### General

As much as 90% of the electrical power supplied to a compressor is directly or indirectly lost as heat, that includes:

- motor losses,
- radiated and convected losses,
- oil cooler,
- after cooler and
- heat remaining in the compressed air.

Heat recovery from very small compressors is not normally viable but a case by case basis should be considered. For larger compressors the potential for heat recovery depends on the extent to which the machine must be modified to recover the heat. In most lubricated screw compressors the oil is cooled and in some the airflow is after cooled. The waste heat from the oil and from the air is ducted away and is dissipated in a cooling airstream. A fan assisted airflow ducts cool air into the compressor housing and warm air is vented from the compressor housing.

As a rough rule of thumb approximately 1kW of waste heat is available for every 10m<sup>3</sup>/hr of compressed air flow at 7barg. The airflow can be recovered at 20 - 25°C above the ambient cooling air temperature. This warm air can be ducted for process heating, drying or for space heating as required with the provisos that:

- Fire regulations are considered and fire dampers are fitted to ductwork if required to meet building regulations
- Oil injected rotary screw compressors will have a safety valve fitted. If this valve lifts the air recovered will be contaminated with an oil aerosol. The safety valve should be piped outside of the heat recovery stream
- The airflow through oil cooler and after cooler must not change so an additional fan may be necessary if ductwork is extensive
- If the heated air is drawn from the compressor house then care should be taken not to reduce the static pressure within the compressor house - or risk reduced compressor performance
- Consideration to transmitted sound and duct attenuation should be given.

The very simple ducting for warm air from a compressor can often provide a very cost effective support for space heating albeit seasonally. In summer months a simple duct diversion can allow the waste heat to be dumped to atmosphere once again.

Where waste heat is used as preheated air for burners it is important to check that the combustion air fans can remain within their fan curves using reduced density of preheated air.

Where waste heat is used for an industrial drying process then it is important to ensure that the back pressure and total flow characteristics remain within acceptable limits - or risk damaging the compressor.

The lubricated screw compressor tends to operate at a much lower temperature than oil free screws. The integration of heat recovery from the water cooled compressor with wet heating systems designed to operate at higher temperatures is therefore sometimes difficult. The water from inter, after and oil coolers may be diverted via heat exchanger so as to preheat a water supply, whether for heating, process heating, feed preheating, makeup heating or some other solution. This is not a new concept and heat was often recovered from the large reciprocating compressors (inter and oil coolers).

For any sizeable compressor the oil is diverted through a little cooler radiator and a low cost conversion to allow water cooling with custom heat recovery is possible and often will return an acceptably short payback e.g. <5years if there is match for the waste heat so that recovery can take place over most of the year and for a high number of operational hours. The waste heat is satisfactory for boiler feed heating, make up heating, space heating, and so on.

The water for cooling can (depending on type and manufacturer) be of 80°C or 90°C.

The opportunity for oil free screws (refer to section 4) is potentially more significant because the oil free screw works at a higher speed and temperature and thus there are bigger inter and after cooler loads to be recovered. Heat recovery from the lubricant oil (non injected gear oil) may also be possible.

## 7.2

### Assessing the feasibility of heat recovery

To assess the feasibility of heat recovery it will be necessary to calculate the heat available from the compressor and evaluate:

<b>Heat Output</b>	
How much heat is available?	
Can it be recovered as warm air or hot water?	
What is the maximum temperature that heat may be recovered at?	
Will modification allow cooling as per the manufacturer's stipulated requirements?	
<b>Demand Profile</b>	
What is the compressors operational profile?	
What is the average compressor loading?	
Is there a heat demand that matches output?	
<b>Heat Sink Profile</b>	
What are the available heat loads?	
What are the demands for these load?	
Do these demands occur at the same time as compressor use?	
<b>Location</b>	
Where is the compressor?	
Where is the heat demand?	
Is heat recovery physically practical?	
<b>Cost</b>	
How much will it cost to modify the compressor?	
How much is the heat output worth?	
How much will it cost to run the heat recovery system?	
What is the alternate cost of providing the heat?	

### 7.3

#### **Calculating the savings potential**

Consider a larger VSD two stage compressor rated to deliver 31.5m<sup>3</sup>/minute and consuming 5.98kW/m<sup>3</sup>/minute (Inclusive of motor and fan power) drawing approximately 188.7kW at 100 to 105psig. In theory, as much as 150kW could be recovered as heat at low grade, displacing a space heating load.

If the machine were operating 4,000hrs per annum then that heat amounts to 600,000kWh of waste heat.

If there is a seasonal heat sink of 2,000hrs per annum (or half of the operational hours) then it might be possible to usefully displace 300,000kWh of heat. If the existing heating boilers were 75% efficient then the displaced gas might be calculated as 400,000kWh and the savings potential at 3p/kWh for gas as £12,000!

It is therefore easy to see why heat recovery can be an attractive option for larger compressors e.g. 55kW or more (although some smaller opportunities may be possible).

# 8 Compressed air leakage

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## 8.1

### General

Compressed air is rarely metered and is invisible. More often than not, leakage goes undetected. Leakage occurs because hoses, couplings traps etc. suffer repeated movement and wear and eventually leak. Where there are flexible couplings or snap fittings, the spring loaded and self-sealing couplings eventually wear.

Given that the process of compression results in 70% or more energy expended as heat, compressed air is also the most expensive service/utility consumed at most sites. At larger industrial sites with extensive compressed air distribution systems leakage might typically be 5-10%, (compressed air leakage is widely reported to be significantly higher but such generalisations are inaccurate) but this obviously depends on the care and maintenance exercised. A formal leak detection and repair programme (an ongoing maintenance programme) will substantially reduce that leakage and the associated cost.

## 8.2

### Calculating the cost of leakage

If the compressors use start/stop or load/unload controls, it is relatively easy to estimate leakage. If the compressors are run at a time of no process demand, the cycle time may be monitored and the leakage calculated accordingly. Once the system is pressurised, the time on load and the time off load is monitored over a period of several cycles. The compressor will simply load and unload to service the leakage. In the case of the unloading screw compressor the line current or pressure rise can be monitored on the control panel. Having established the ratio of loaded to unloaded or on/off time the leakage may be evaluated as:

The pressure rated FAD x loaded time / (loaded +unloaded time).

The associated power consumption and cost can then be calculated.

So for example if the a 55kW compressor operating at 9.6kW/m<sup>3</sup>/hr (17.7kWh/100acfm) is providing air to a system and found to cycle 18 seconds on then 162 seconds off the leakage rate might be evaluated as 10% or 44,000kWh in the working year worth say £5,720 per annum at 13p/kWh – for larger systems, the loss will be proportionately greater.

If the compressors modulate to a target pressure, then it is much more difficult to physically calculate the leakage but assuming all process loads have been stopped, the power drawn by the compressor can be monitored over time and the cost of leakage thus evaluated.

The cost of the electrical power associated with leakage is only part of the story, because the additional airflow requires drying, filtration, and results in extended compressor hours, maintenance and so on.

## 8.3

### Leak detection

Leakage can be audible and listening for leaks during a production shut down can be a useful means of identification. However if the leakage is masked by production noise or is ultrasonic then the job is more difficult. Often in a busy factory or production area this is the case.

When air leaks from the high pressure pipeline or coupling to the low pressure ambient the friction and turbulence of that flow creates an ultrasonic (inaudible) noise. This ultrasonic noise is generated local to leakage and can be detected by an instrument that is designed to respond to that frequency of noise. Importantly the microphone or detector has to be able to detect in the relevant ultrasonic region (the relevant range being well in excess of 30kHz and most microphones have a range that reaches only 20kHz) – thus, a standard microphone will not necessarily detect leakage - but actual pipe flow.

High frequency ultrasound emissions occur locally, but may also be generated from section changes etc – so care in identification is still required. However a good quality ultrasonic leak detection system with appropriate calibration and software will allow detection of the relevant ultrasound and conversion to an audible output for the operator.

In purchasing an ultrasonic leak detector the quality of the equipment is therefore vital. In interpreting the output, the skill and experience of the operator are essential. It is always important to check the results for order of magnitude.

Using a specialist third party contractor is also possible, however it is important they have specific expertise of compressed air systems and are not relying solely on a software based analysis of leakage.

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## 8.4

### Where will the system leak?

- Old pipework - if the pipework has corroded there will likely be leakage at screwed or flanged joints
- Other pipe fittings that have been stressed or incorrectly supported or that have been subject to movement or vibration
- Inaccessible connections where correct fitting and tightening would have been (is) difficult.
- Flexible hoses with cuts, nicks or which have simply worn through.
- Quick or push fit couplings where the seals will wear
- Trapped drain or separator non returns which eventually foul and stick open
- Tools with internal seal wear.

## 8.5

### Managing a programme

The most important factor in reducing leakage is to have a system for detection, notification and repair. Identifying leakage is worthless unless there is a system in place to repair the leakage. Adopting a formal strategy for leak reporting, tagging and repair is therefore a vital element of cost reduction. Typically this system will incorporate a simple severity coding and a maximum time to repair.

Staff should be encouraged or required to report leakage or damage and the maintenance staff (internal or external) will be required to respond and repair the leakage within an agreed timeframe.

# 9 Efficient system design and operation

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To achieve the maximum operational benefit the demand must be minimised (refer to section 10 of this guidance) and the compressor /control combination must be selected appropriately (refer to sections 3 and 4 of this guidance).

Minimising demand and selecting the correct compressor combination for the installation is a little more complex than first might be imagined and whilst VSD can offer fantastic power reductions the load match and control strategy must be evaluated carefully if the real benefits of this technology are to be realised. The selection process must consider the daily and potentially seasonal demand variations and must incorporate sufficient compressor numbers and disposition to allow optimal specific power consumption.

To ensure maximum efficiency however, the whole system design must be considered and that of course includes the distribution. Pressure loss in the distribution system can undermine any diligent compressor/control selection.

### 9.1

#### Appropriate use and demand management

The most efficient way to use compressed air is not to use it at all and a fundamental starting point is to consider the alternatives to compressed air and decide whether in fact it is really necessary at all.

As a first step to energy and resource efficiency it is necessary to understand the compressed air demand, the extents of the distribution and the pressure constraints.

Compressed air should only be used where essential or where the use of alternate actuation is dangerous. For example:

- It is sometimes the case that a compressor and entire air system or extended distribution is installed to cater for the operation of a single or small number of air actuated rams. Unless these are used in an intrinsically safe environment there may be an opportunity to replace these actuators with simple electric equivalents.
- Often air driven pumps are used for clean environments or explosive atmospheres. The use of air driven pumps should be avoided in all but essential situations for these are least efficient means of pumping. In many cases the use of variable speed drive can effect much more accurate capacity control.

- Compressed air is often used for cleaning – a practice to be avoided where possible because it is expensive and the particulate blown from the product or surface of the cleaned item becomes an airborne particulate contaminant – there is an allied health and safety (COSHH) risk.
- Compressed air is used for cooling and shaping in some extrusion processes. In some particularly high temperature processes steam can actually be used at a fraction of the cost. For lower temperature cooling only applications fans or blowers can shift a larger volume of air with lower pressure and improved heat transfer coefficient at a fraction of the cost.

Appropriate use of compressed air is essential if cost is to be controlled and the first step is a survey of equipment, air demand and utilisation.

### 9.2

#### System pressure

The system pressure is potentially dictated by the compressor type and the market choice available (refer to Section 3). Compression to higher pressure has an associated specific power penalty. However, one of the most easily accommodated means of buffering load variation is to generate at high pressure and utilise air at a lower pressure. The stored energy or accumulation provides an inherently useful buffer capacity.

Unfortunately compression to a higher pressure requires considerably more energy. Assuming a nominal pressure of 7.5 barg a 1 bar reduction in generation pressure would afford (approximately) a 7.25% reduction in power consumption (very approximately 1% for every 2psig reduction).

It is therefore vital that the system pressure is as low as possible, whilst retaining sufficient buffer or stored energy.

- That is achieved by ensuring that the control system has the lowest overall operational bandwidth (refer to section 4). This means the average generation pressure is the lowest.
- The system losses are no more than 10% by ensuring pipes and fittings are adequately sized to prevent excessive pressure drop.
- Leakage is managed to a minimum.
- The system is maintained effectively (Principally ensuring that coalescing/cartridge type filters are changed on a regular basis).
- The compressor air intake filters are kept clean.



### 9.2.1

#### Mixed pressure systems

Do not sacrifice a low overall system pressure to service a small high pressure load. Install a separate compressor or use pressure amplification. A pressure amplifier is simply a compressor driven by your low pressure air that produces high pressure air. For small high pressure loads in an otherwise low pressure environment, these allow the main system pressure to be dropped.

For larger loads a dedicated compressor is probably advisable but every case should be evaluated on individual merit.

### 9.3

#### Pipework design

Compressed air pipework offers frictional resistance to flow. The amount of friction depends on the roughness of the pipe surface and the velocity of the airflow. The losses are a squared function of velocity – it is therefore important to keep velocity as low as possible whilst being economic.

It must be accepted that unless the pipeline is of infinite diameter and the fittings of infinitely small resistance, there will be a system pressure drop. The generally accepted (existing good practice guidance) maximum should not typically exceed 10psig from compressor to remote point of use. That however requires the compressor to operate with an inherent 5% efficiency penalty. So for example if the required pressure is 7barg and the system drop is 10psig then the compressor discharge is going to be 7.7 barg and the averaged inherent loss circa 5%. On large system (say a 160kW compressor with a 50% duty running 8,000hrs) that might cost £50,000 over a 10 year life span. It is therefore easy to see why designing for efficiency is valuable.

In evaluating the potential benefits it is not the cost of the pipe work that must be offset it is the marginal cost of making the pipework larger. The life cost from derived pressure drop has to be assessed on a component by component basis and to take account of practical constraints. For example the compressor house pipework may offer a very low pressure loss at 200mm but the cost or practical installation of 200mm pipework is not justifiable or easily installed and a 150mm header may have to be accepted. In many cases this will be a relatively trivial in the overall life cost.

If a target system drop of 10psig (or as illustrated 0.7 barg, approximately 10%) is accepted the life cost penalty is significant. To maximise efficiency a lesser pressure drop should be targeted in design – subject to the marginal cost being outweighed by life operational cost. A target of 2.5% or 5% losses is more appropriate as electricity costs rise.

The pressure drop through a section of pipe can be calculated manually or more normally determined from tables or a Darcy chart for any particular pipe material and of course the velocity (and associated pressure drop) must be estimated for the maximum anticipated flow through the relative section. The pressure drop differs for differing pipeline materials because the roughness of the pipe varies. Black iron (commonly used for large mains) is a great deal rougher than say extruded aluminium or plastic.

The pressure drop through fittings is generated by turbulence and momentum change and flow through and immediately before and after the fitting is disrupted. Accordingly the disruption will have an impact on the flow in the straight pipe before and after the fitting. For this reason the pressure drop through a fitting is normally calculated as a friction loss for the flow calculated for an equivalent length of piping.

Compressor output and sometimes equipment consumption is usually expressed as FAD (Free Air Delivery) this is the air consumed at ambient conditions. The pressure flow is lower and must be calculated by compression ratio.

### 9.3.1

#### Acceptable velocities

Because the pressure drop is a square function of velocity, two approaches might be used to calculate system pressure. Initially it is useful to estimate all line sizes based on a notional acceptable velocity (A velocity which is known to give reasonably small pressure drop). However after the design is assembled in this way it is important to check that the cumulative system does not generate excessive drop.

As a rough rule of thumb:

- Header and main line velocities should not exceed 6ms<sup>-1</sup>
- Shorter outlying branches and drops should not exceed 9ms<sup>-1</sup>
- The system should be designed for peak anticipated flow

- The overall system drop should be checked
- Use whole life costing for economic appraisal.

### 9.3.2

#### Material selection

Friction loss will also be affected by internal corrosion and steel pipe (referred to as black iron) will corrode. As it does so the roughness of the inside increases dramatically and the pressure losses increase. Steel pipe is heavy and thus relatively expensive to install. Moreover the installation requires training and specialist assembly with typically screwed or welded connection.

Advances in materials technology now allow the use of plastic (AABS or similar) and aluminium systems which have extremely low roughness, low friction and are much lighter than steel allowing cheaper installation. Some systems incorporate proprietary jointing techniques e.g. clamping that eliminates the requirement for welding. Moreover such materials do not corrode and there is little increase in flow resistance over time. Installation does not require skilled labour and this serves to mitigate the overall cost of installation.

The use of aluminium or plastic low resistance pipework may improve the opportunity to reduce system pressure drop to and improved design pressure drop and the additional cost of these systems may be evaluated against the expected operational life cost benefit.

The discharge temperature of some compressor types can be very high and here it is difficult to use plastic pipe (or in some cases aluminium pipe at these sizes). The use of steel pipe may be necessary for discharge pipework.

### 9.3.3

#### The concept of the ring main

Where ever practical a ring main should be used to service multiple common pressure users. The concept effectively reduces the index (worst route pressure drop) and allows the lowest operational pressure to be used. The concept of ring main also provides a degree of redundancy.

## 9.4

### Compressed air storage

The compressed air receiver provides several functions (directly or indirectly) but essentially allows for the buffer of air so that the compressor does not see the demand directly. This storage function flattens the worst peaks reducing in pressure as it does so and prevents the compressor offloading when the demand diminishes for short periods by allowing the storage to pressurise. The wet receiver also allows a degree of condensation before the dryer if a colder location can be established. Regardless, and in order to retain a low and optimised generation pressure and small control differential, the receiver must be sized generously.

The wet receiver will typically be sized (as a minimum) on the difference in load and unload pressures, the maximum number of on/off load cycles with a correction for the volume increase resulting from additional heat of compression, using (Courtesy Atlas Copco)

$(0.25 \times Q_c \times T_i) / (F_c \times P_d \times T_o)$  where:

$Q_c$  = Compressor capacity in Litres/second

$T_i$  = Temperature into compressor in Kelvin

$F_c$  = Maximum cycle frequency in Cycles/seconds e.g. 1 cycle every 30 seconds

$P_d$  = The control differential in bar e.g. 0.3bar

$T_o$  = Temperature into receiver in Kelvin

Accordingly if a 158l/s compressor was operating with a maximum cycle frequency of 1 cycle every 45 seconds and the control differential was 0.5bar, with an inlet temperature of 10°C and an expected outlet temperature after compression, aftercooling and water separation of 25°C - the volume would be 3.37m<sup>3</sup>.

A dry receiver (post drying installation) does much the same thing but there is a lesser condensation function, particularly in low dew point drying systems. A central receiver buffers the compressor and local or distributed receivers may be used to flatten the pressure profile experienced at remote points of use.

Where the dry receiver is used as a buffer then the required size (to cover say a transient load) could be estimated using:

$$V = T \times D \times P_a / (P_f - P_i)$$

V = Receiver volume, m<sup>3</sup>.

T = time allowed (minutes) for pressure drop to occur

D = Air demand, m<sup>3</sup>/minute of free air

P<sub>a</sub>= Absolute atmosphere pressure, bar

P<sub>i</sub>= Initial receiver pressure, bar

P<sub>f</sub>= Final receiver pressure, bar

If compressor is delivering to this system and this is buffering peak flow then the compressor output Q<sub>c</sub> can be subtracted as:

$$V = T \times (D - Q_c) \times P_a / (P_f - P_i)$$

**9.5 Compressor location**

The compressors operating conditions can influence system efficiency. Hiding the compressor away in an overheated plant room will have a marked effect on efficiency. As a rule of thumb (and for the operating temperatures over which most compressors function) an increased intake temperature will reduce volumetric efficiency an increase power consumption by approximately 1% for every 3.5°C increase in suction temperature.

So for example if the 55kW compressor is providing air to a system and working with a 50% load factor and 4,000hrs per annum. The air temperature is 25°C well in excess of the temperature the compressor is rated at. The density of air is lower at 25°C than it is a 15°C and the mass of air drawn is reduced. The compressor will have to work harder to provide the same mass flow. As a rule of thumb every 3.5°C temperature increase will reduce the compressor performance by 1%.

The intake filter must be kept clean, as this presents a pressure restriction and an air filter which is caked with dust and paint will increase the pressure restriction at intake – effectively increasing the compression ratio and the work required.

Design checklist	
Key Issues	
Note this is simply a list of reminders for energy efficiency and does not replace your suppliers recommendations	
<b>Demand Reduction</b>	Assess Demand (diversified peak flow)
	Eliminate demand
	Use blowers instead of compressed air
	Use electric actuation instead or air
	Use electric pumps instead of air pumps
	Don't use compressed air for cleaning

## Design checklist (continued)

### Key Issues

Note this is simply a list of reminders for energy efficiency and does not replace your suppliers recommendations

<b>Compressor</b>	Select the correct compressors for volume and pressure
	Use multiples of compressors and do not rely on a single variable speed compressor for very variable loads. Optimise the specific power consumption
	Ensure the quality of airflow to compressor is clean filtered air. Ensure filter pressure drop is a minimum
	Select compressor location to allow the draft of clean cold air
	Generate at the lowest available system pressure
	Use PLC multi compressor controller with remote pressure sensing and proactive compressor management
<b>Air Quality</b>	Evaluate the required air quality for the majority of plant - Meet the general air quality standard for the bulk of supply
	Meet local air quality requirements with local drying and filtration
	Do not over dry the air supply
	Do not over filter or clean the air
	Oversize filters to achieve lowest pressure drop
	Check and replace cartridge and coalescing filters on a regular basis
<b>Operational Pressure</b>	Use compressor control to minimise the set point spread and to allow the lowest average operational pressure
	Use a wet receiver to buffer compressor output and improve drying
	Check for leakage to minimise pressure drop
	Generate at lowest pressure for majority use. Use local compressors or pressure amplification for high pressure uses
	Use remote pressure sensing
	***Distribute compressors to best serve the loads*** Use distributed compressors only if the installed control system is capable of remote pressure sensing and remote compressor management.
	Design using directional flow T's and low pressure drop fittings, Design for low target system pressure drop Restrict velocities to reduce pressure drop

<b>Design checklist (continued)</b>	
<b>Key Issues</b>	
Note this is simply a list of reminders for energy efficiency and does not replace your suppliers recommendations	
<b>Distribution</b>	Size for minimal overall system pressure drop
	Use directional T's and low pressure drop fittings
	Oversize filters to assure lowest pressure drop
	Size hoses and flexible connections for the duty – Minimise the lengths of flexible connections
	Adopt ring main configuration
	Choose Low pressure drop valves – minimise use of high pressure drop globe or angle pattern valves
	Enable the isolation of system components that have intermittent, no overnight or no week end use
<b>Storage</b>	Design to minimise compressor size
	Use wet receiver to reduce drying load and buffer compressor
	Locate wet receiver in cool location – preferably externally
	Use separator prior to wet receiver
	Use Electronic condensate drain trap or specialist sludge trap for poor environmental conditions
	Use dry receiver to buffer air flow but assure that compressor control remains effective
	Use dry receivers to buffer transient on site loads and to prevent excessive local pressure reductions
<b>Heat recovery</b>	Design for heat recovery
	Can the compressors be selected (water or air cooled) to effect best heat recovery
	Will the compressors be located to effect best heat recovery
	Consider heat recovery for space heating, drying, process heating, DHW heating, air drying
	Do ensure that seasonal heat demand variations are accounted in appraising the economic case

# 10 Maintaining for Efficiency

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### 10.1

#### **Maintenance frequency**

The frequency with which maintenance must be conducted is a function of utilisation – the more hours the more maintenance. That periodicity can be determined in consultation with your compressed air equipment suppliers and in accordance with the specific requirements stipulated by the manufacturer. However, effective maintenance will have a profound effect on efficiency cost and operational reliability.

### 10.2

#### **Compressors**

The cost of the compressors may constitute 20% or more of the life cost of the plant. Failure to maintain that asset will result in eventual failure. The periodic service, oil change and tolerance checking is vital if best efficiency and best lifespan are to be achieved.

Oil change and cleanliness are key factors and maintaining the good condition of intake air is key to reducing wear and increasing system efficiency. The inlet filter for the compressor should be selected for minimum pressure drop whilst providing adequate particulate screening. Compressor capacity will reduce proportionally with inlet pressure drop.

Lubricated screw compressors or rotary vane compressors are particularly vulnerable to particulate in the inlet air because that particulate is blended with the sealing oil and causes rapid wear.

### 10.3

#### **Conditioning plant**

The aftercooler (integral or otherwise), water separator, pre-filter, dryer, post drying filters and wet and dry receivers all must be inspected and maintained for each component has the potential to cause damage in the event of failure.

Depending on the dryer type (refer to section 6) the pre-filtration must be carefully managed and maintained if damage to the dryer is to be prevented.

The receivers are unfired pressure vessels and subject to the provisions of the PED (Pressure Equipment Directive) and the Pressure Equipment Regulations, the Pressure Systems Safety Regulations. These are subject to mandatory inspection by a notified body.

### 10.4

#### **Traps and drains**

Traps incorporated within inline filters and separators, line traps and receiver traps should all be regularly checked. Fail open results in excessive leakage with associated cost penalty. Fail closed results in reduced capacity and excessive moisture levels in the air. Both have cost and reliability consequences.

### 10.5

#### **Line filters**

Blocked or heavily obscured filter units will result in excessive pressure drop, equipment underperformance or malfunction all with associated cost.

### 10.6

#### **Leakage**

Leakage and the cause of leakage are addressed in section 8 of this guidance.

## Maintenance checklist

### Key Issues

Note this is simply a list of reminders for energy efficiency and does not replace your suppliers recommendations

<b>Compressor</b>	Intake filter appropriate periodic clean or replacement
	Oil filters appropriate periodic clean or replacement
	Compressor oil periodic replacement
	Injected oil periodic replacement
	Interim and full service as per manufacturers schedule essential
	Drive maintenance – Belts, motors, bearings
	Inter cooler heat exchange surfaces clean
<b>Treatment</b>	After cooler inspection and operation check pre dryer temperature
	Maintain critical pre dryer filtration and water separation
	Dryer performance check
	Receiver condition and drainage
	Post dryer temperature
	Regularly replace post dryer filtration
<b>Distribution</b>	Evaluate pressure and monitor system pressure drop
	Check the correct operation of traps and separators
	Change cartridge filters regularly
<b>Leakage</b>	Check and monitor system pressure drop over time
	Institute a leak management programme
	Conduct audible and ultrasonic surveys on a regular basis
	Generate at lowest pressure for majority use. Use local compressors or pressure amplification for high pressure uses
	Repair leaks promptly
	Minimise flexible hose lengths
	Check and or replace quick couples



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