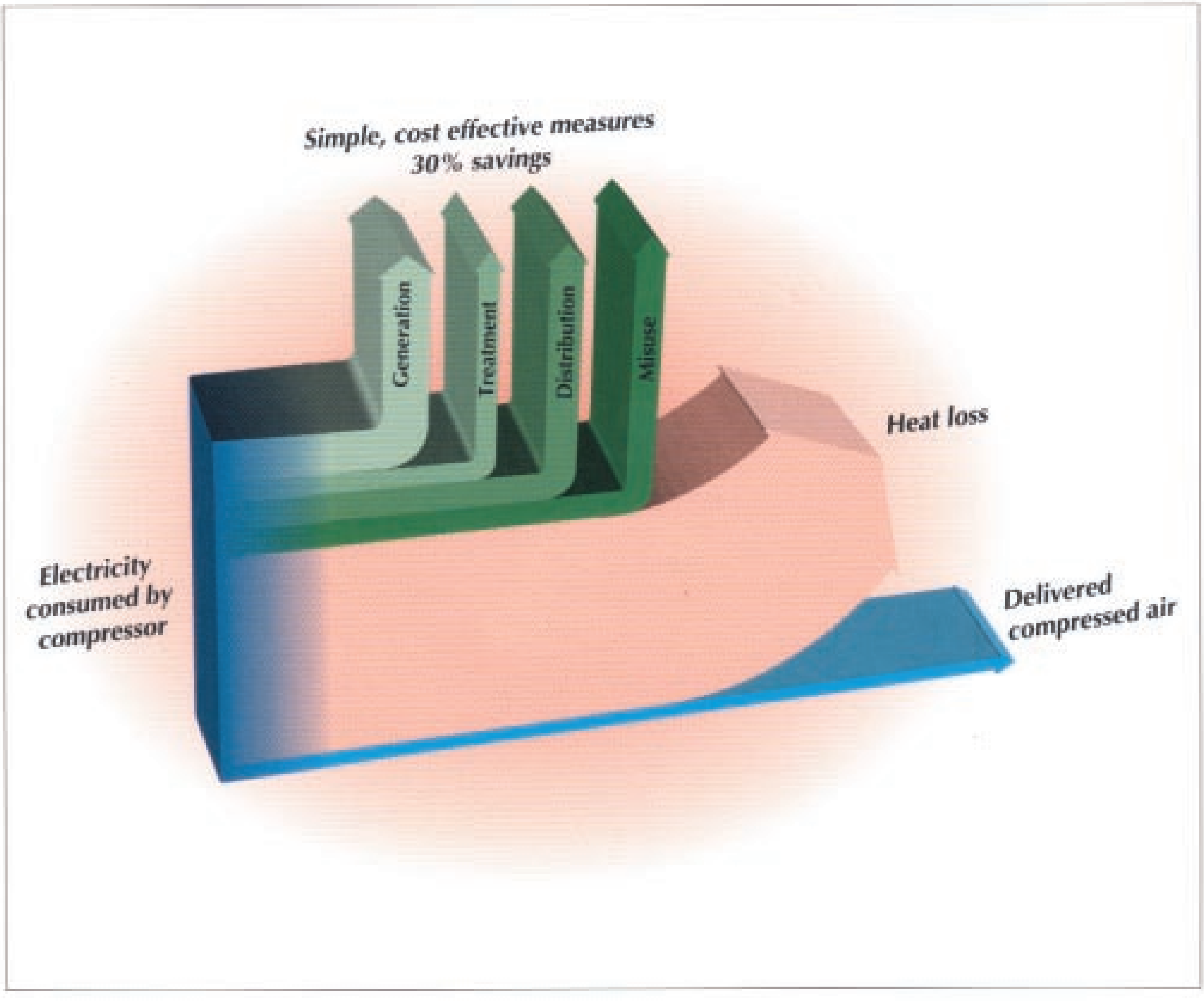
GOOD PRACTICE GUIDE 126

Compressing Air Costs



GOOD PRACTICE GUIDE 126

ENERGY EFFICIENCY

BEST PRACTICE
PROGRAMME

COMPRESSING AIR COSTS

This booklet is No. 126 in the Good Practice Guide series. It provides advice on practical ways of improving energy efficiency in plants generating and distributing compressed air. It considers the various systems currently in use and shows how such systems can be monitored and demand reduced. Case histories are also included, providing practical examples of how savings have been made.

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First published 1994
Reprinted 6/94, 4/95, 6/96, 3/98, 12/98, 9/99, 12/00, 9/01

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FOREWORD

This Guide is part of a series produced by the Department of the Environment, Transport and the Regions under the Energy Efficiency Best Practice Programme. The aim of the programme is to advance and spread good practice in energy efficiency by providing independent, authoritative advice and information on good energy efficiency practices. Best Practice is a collaborative programme targeted towards energy users and decision makers in industry, the commercial and public sectors, and building sectors including housing. It comprises four inter-related elements identified by colour-coded strips for easy reference:

— *Energy Consumption Guides:* (blue) energy consumption data to enable users to establish their relative energy efficiency performance;

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ETSU would like to acknowledge the following people for their comments on this Guide:

Eric Harding (Air Technology)
The British Compressed Air Society (BCAS)
Peter Gale (CompAir Broom Wade)
Brian Johnston (Spirax Sarco)
Paul Ellen (Land Rover)
Mike Birch (Creda)
Sir Savile Burdett (Rapaway Energy)

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COMPRESSING AIR COSTS

1. INTRODUCTION

Compressed air represents approximately 10% of industrial electricity usage and is present, either as a utility or integral process raw material, on the vast majority of industrial sites. The versatility, flexibility and safety of compressed air as an energy transmitting medium ensures its use as an essential service. Typically over a ten year period the total costs are 75% energy, 15% capital and 10% maintenance. An energy efficient system is therefore highly cost-effective, even if it costs slightly more to install.

Research has shown that the annual total cost of generating compressed air in the UK, based on an average price of 4.5 p/kWh, is £340 million equating to a consumption of 7.5 million MWh of electrical energy per annum. It is estimated that 30% of this is wasted and could be saved by introducing simple, cost-effective energy efficiency measures. This corresponds to a potential annual saving of 2.25 million MWh or £100 million.

Compressed air is a very expensive form of energy. Fig 1 shows the relative cost of producing compressed air compared with other energy sources. By the time the air reaches the end user, the equivalent cost is in excess of 50 p/kWh. Most people using compressed air equipment are not aware of this fact, and see it almost as a free and convenient resource.

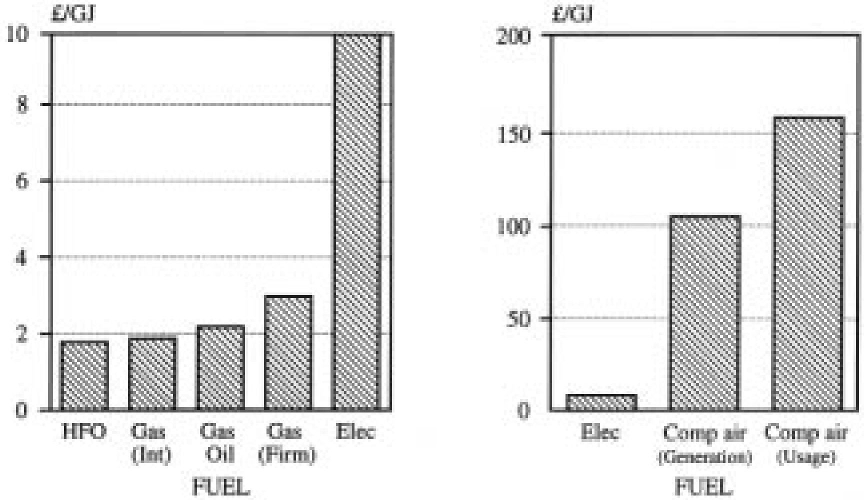

Fig 1 Relative energy cost of compressed air

With the help of this Guide, everybody can make some savings from their compressed air systems. Taking the overall identified saving of 30%, the annual savings potential related to installed capacity of the site compressors (assuming 100% utilisation) would be as follows:

Compressor Capacity			Annual Potential Savings (£) @ 4.5 p/kWh	
kW	**l/sec**	**cfm**	**48 h/wk**	**120h/wk**
18	55	110	700	1,700
85	250	500	3,100	7,800
165	500	1,000	6,000	14,900
320	1,000	2,000	11,400	28,400

These are average figures, and in some cases almost double these savings would be achieved.

Throughout this Guide, a typical example of a factory demand for 500 l/s of compressed air at 7 barg will be used. If the compressor operates on a 48 hour week with power costs of 4.5p/kWh it will cost over £20,000 per annum to generate the air.

The purpose of this Good Practice Guide is to provide information on how to improve the efficiency of your compressed air system, thereby reducing your costs and increasing profit. The Guide concentrates on the areas of:

- end usage;
- distribution;
- control;
- treatment;
- generation.

Examining these areas in this order represents the best approach to maximising savings at the earliest opportunity. Indeed, leakage from distribution networks and the control of compressors account for approximately 70% of the total wastage.

In addition, the Guide investigates methods of recovering the considerable waste heat generated by air compression. Consideration is also given to the benefits of cooling air before it enters the compressor inlet, and a section is included on the Pressure Systems and Transportable Gas Containers Regulations.

A number of case histories are incorporated within the Guide and these serve to highlight typical applications with resulting opportunities to increase the overall energy efficiency of compressed air systems.

This Good Practice Guide discusses compressed air systems operating at normal factory pressures of 7 barg unless otherwise stated.

2. HOW HIGH ARE YOUR COMPRESSED AIR COSTS?

To estimate the annual costs of compressed air on your site you need to know either the rating of the compressor motor(s) (kW) or the rated output of the compressor(s) in l/sec or cfm.

Table 1 below gives guideline consumptions, including allowances for motor losses, depending on machine utilisation.

Table 1 Annual compressed air costs (based on 4.5 p/kWh)

Compressor Capacity			Annual Costs (£) 48 h/week utilisation		Annual Costs (£) 120 h/week utilisation	
kW	l/sec	cfm	75%	50%	75%	50%
18	55	110	1,700	1,100	4,200	2,800
25	75	150	2,300	1,600	5,900	3,900
37.5	110	220	3,500	2,300	8,700	5,800
55	165	330	5,100	3,400	12,700	8,500
85	250	500	7,800	5,200	19,500	13,000
165	500	1,000	14,900	10,000	37,400	24,900
250	750	1,500	22,400	14,900	56,000	37,300
320	1,000	2,000	28,400	18,900	70,900	47,300

Machine utilisation can be calculated by noting the average on-load and average off-load running times of the compressors. If a more accurate calculation is needed of the air usage then the equations included in Section 3.3 can be used.

3. REDUCING END USER DEMAND

3.1 Air Use

Investigating where and how compressed air is used around site will reveal the areas in which major savings can be achieved.

By the time compressed air reaches the end user, its cost as an energy source is very high, around 50p/kWh. It is vitally important that each use is investigated in detail to establish whether:

- it needs to be operated by compressed air at all;
- the supply pressure is greater than necessary;
- there is adequate facility for isolating the supply line when it is not in use.

Some of the case histories in Section 9 include instances where compressed air has been used when a cheaper alternative is available. Two commonplace examples of misuse are using compressed air for cleaning or cooling duties: alternatives exist for cleaning benches, and cooling duties can generally be carried out using high pressure fans or a lower pressure compressed air system.

Air knives are another frequent misuse of compressed air. Dedicated high pressure blowers are far less costly and can be linked to the conveyor, such that the blower is automatically switched off if product stops passing beneath the knife.

Other misuses of high pressure air (7 barg) are:

- air agitation - often used on effluent plants. High pressure blower fans or lower pressure compressed air are often just as effective;
- air jets to eject products from high speed machines. Air intensifying nozzles are excellent energy saving devices for air jet applications.

3.2 Air Leakage

Leakage is the largest single waste of energy associated with compressed air usage. Leakage rates exceeding 50% of site consumption are common. The sources of leakage are numerous, but the most frequent problems are:

- condensate drain valves left open;
- shut-off valves left open;
- leaking pipes and pipe joints;
- leaking hoses and couplings;
- leaking pressure regulators;
- air cooling lines left open permanently;
- air-using equipment left in operation when not needed.

It is estimated that UK industry wastes in excess of £20m each year on air leaks. Table 2 gives an example of how even very small holes can contribute to energy wastage.

Table 2 Power wastage through leaks

Hole diameter	Air leakage at 7 barg		Power required to compress air being wasted	Annual cost of leak	
				48 h/wk	120 h/wk
mm	l/s	cfm	kW	£	£
0.4 (pin head)	0.2	0.4	0.1	12	30
1.6 (match head)	3.1	6.2	1.0	120	300
3.0	11.0	22.0	3.5	420	1,050

Leakage is not only a direct source of wasted energy, but is also an indirect contributor to operating costs. As leaks increase, system pressure drops, air tools function less efficiently and production is affected. Often the only solution is to increase generation pressure to compensate for the losses.

The first step in tackling leaks is to recognise the costs involved and make a commitment to a plant-wide awareness programme. Regular, continuous attention to the compressed air system coupled with proper maintenance will lead to effective progress in minimising leaks.

3.3 No-load Testing

The best way to establish the amount of leakage in a system is by measurement (see Section 8 for details). However, in the absence of suitable measuring devices, a no-load test should be carried out to establish the percentage leakage from the system. Two possible methods that can be used are as follows.

- *Method 1:*
  This applies to compressors that are operated in on/off-load i.e. when the compressor is on-load it produces a known amount of air.
  - Close down all the air operated equipment.
  - Start the compressor and operate it to full line pressure, when it will off-load. Air leaks will cause the pressure to fall and the compressor will come on-load again;
  - Over a number of cycles make a note of average on-load time (T) and average off-load time (t).
  - Total leakage can then be calculated:

$$\text{leakage } (l/\text{sec}) = \frac{Q \times T}{T + t}$$

  where: Q = air capacity of the compressor (l/sec).

- *Method 2:*
  For modulating compressors the test is more difficult as the compressor output is unknown. The following method can be used if you have a pressure gauge downstream of the receiver.
  - Calculate the volume V (litres) of air mains downstream of the receiver isolating valve, including all the pipework (25 mm and above) and the receivers.
  - Pump up the system to operating pressure (P_1), and then close the isolating valve.
  - Record the time (T) for pressure to drop to P_2.

- Leakage can then be calculated as follows: [1]

$$\text{leakage } (\text{l/sec}) = \frac{V \text{ (litres)} \times [P_1 - P_2] \text{ (barg)}}{T \text{ (secs)}}$$

Having established the size of the problem, a realistic target for leakage rate (say 10%) should be set. No-load tests should then be carried out regularly, approximately every two or three months, with inspections when necessary during shutdown conditions.

3.4 Leak Detection

During shut-down of the whole factory, the detection of larger leaks within the compressed air system is simple, as they are audible. Once the compressor is started the exact leak positions should be marked with tags. In addition, checking joints and unions with soapy water is recommended to identify the smaller leaks which invariably develop with time. An organised approach is required to obtain good results. Once leaks have been detected they should be repaired as soon as possible, most can be repaired simply by tightening components and shutting off valves which have been left open. One method of motivating people to fix leaks is to offer a reward for the person who collects the most tags.

When plant is operating 24 hours per day it is more difficult to conduct no-load leakage tests and conditions make it very difficult to audibly detect leaks. Therefore, preventive maintenance (e.g. regular replacement of seals, gaskets, lubricating oil etc.) and operational methods (e.g. ensuring valves are closed) become critical in order to avoid energy wastage.

An Ultrasonic Leak Detector is a more effective leak detection method as it can detect leaks against a background of other equipment noise. The detectors work by picking up the very high frequency sound emitted by a leak, inaudible to the human ear. They are simple to use and do not pick up frequencies emitted by the mechanical actions of machines.

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1 **Note:**

System volume = V (m^3)
Initial pressure of system = P_1 (barg)
Final pressure of system = P_2 (barg)
Time for pressure decline = T (secs)
Atmospheric pressure = Pa = 1.013 bar

Initial quantity of <u>free</u> air $= \dfrac{(P_1 + P_a)}{P_a} \times V \quad (m^3)$

Final quantity of <u>free</u> air $= \dfrac{(P_2 + P_a)}{P_a} \times V \quad (m^3)$

Therefore, flowrate $= \dfrac{V}{T}\left(\dfrac{P_1 + P_a}{P_a} - \dfrac{P_2 + P_a}{P_a}\right) \quad (m^3/sec)$

$$= \frac{V}{T}\left(\frac{P_1 - P_2}{P_a}\right) \quad (m^3/sec)$$

$$= \frac{V}{T}\left(P_1 - P_2\right) \qquad \text{since } P_a = 1$$

The above equations assume constant temperature.

4. DISTRIBUTION NETWORK AND ANCILLARY EQUIPMENT

Having investigated possibilities for reducing the end usage of air and having determined the extent of leakages, the next step is to examine the distribution network. The main factors that affect energy consumption at this stage are:

- pressure losses due to inadequate pipe sizing;
- water condensing in the lines, causing damage to components and also reducing pipe cross-sectional areas which leads to additional pressure losses;
- air leakage from pipework and end-using equipment, due to poor maintenance and in many cases permanently open condensate drain valves.

A typical compressed air system is outlined in Fig 2. The following Sections refer to this diagram, describing the main design criteria and components.

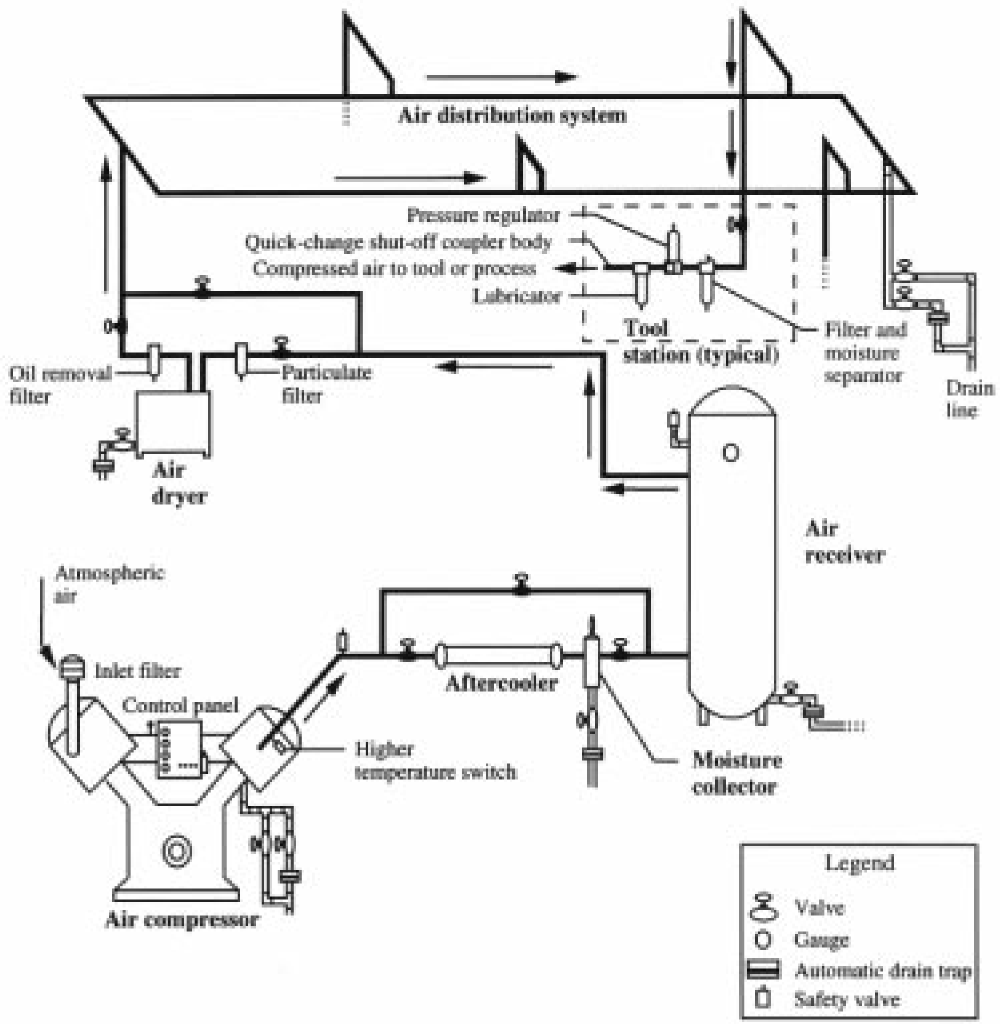


Fig 2 Compressed air system components

4.1 Distribution Main Sizing

Generally, a ring main is the most effective method of supplying compressed air to a point of use. The main advantages of this type of network are that:

- velocity to any one point is reduced, since air can converge from two directions, thereby reducing pressure drop over the system;
- automatic zone valves can be fitted to isolate areas operating different working patterns;
- alteration or extension of the distribution system is made easy.

Ideally a ring main should be placed around each building and a single branch point should feed the main, enabling each area of the network to be metered by a single meter (see Section 7). An example of such a network is given in Fig 3.

Air line diameters are usually based on calculations of throughput velocities. A figure of approximately 6.0 m/s is the accepted target value, because this is sufficiently low to prevent an excessive pressure drop. Table 3 outlines the maximum recommended flow in various line sizes based on a 7 barg output pressure.

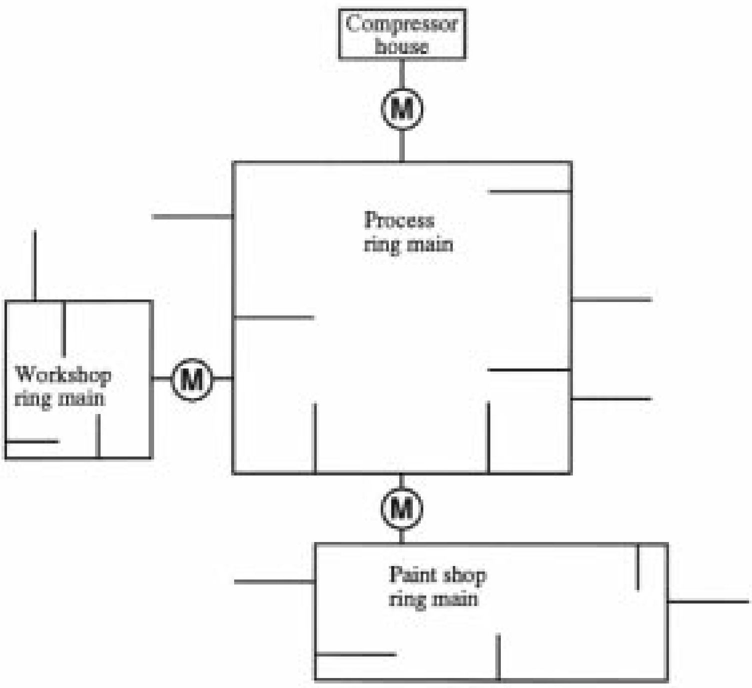


Fig 3 Preferred distribution line layout

Table 3 Maximum recommended flowrates

Pipe Bore	Max Flow	
mm	l/s	cfm
10	5	10
25	25	55
50	100	220
65	180	375
80	240	500
100	410	875
150	900	1,900

For a quick reference, the nomographs outlined in Figs 4 and 5 are a useful guide to sizing distribution lines. The distribution system should be designed to cause no more than 0.2 bar pressure drop at full demand at the point of use.

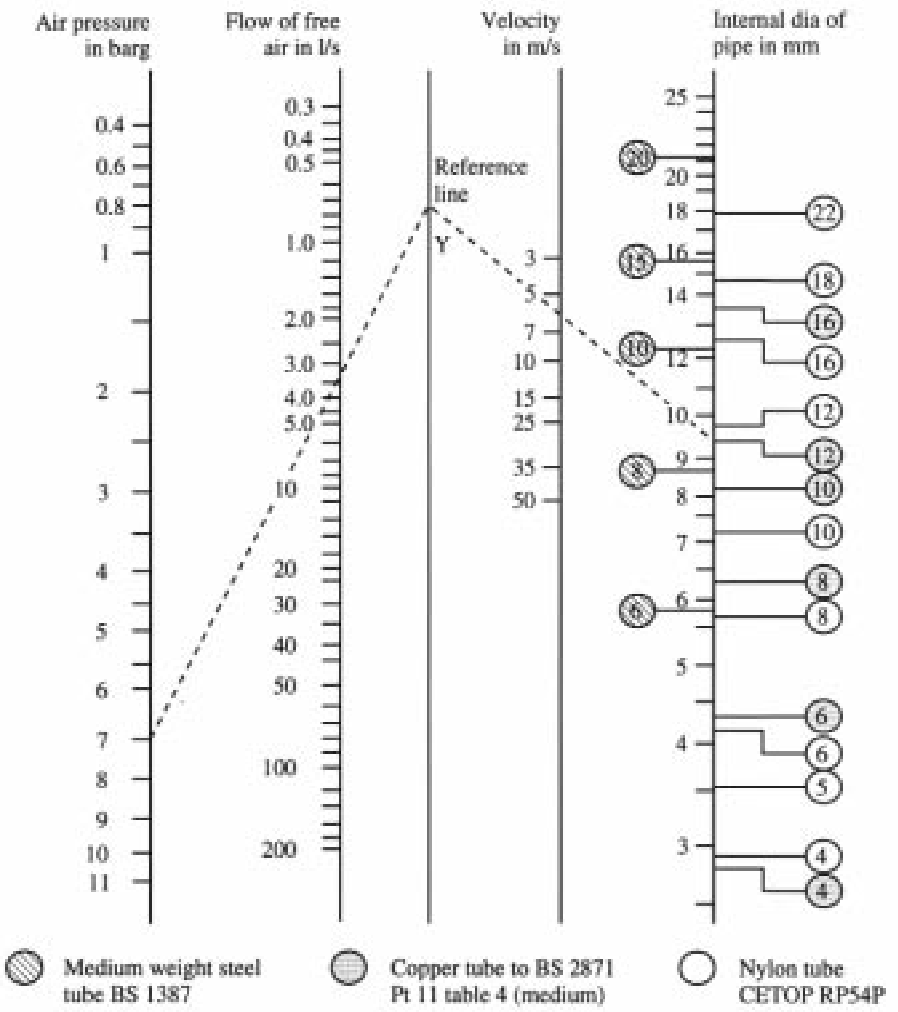


Fig 4 Pipe carrying capacities at varying velocities

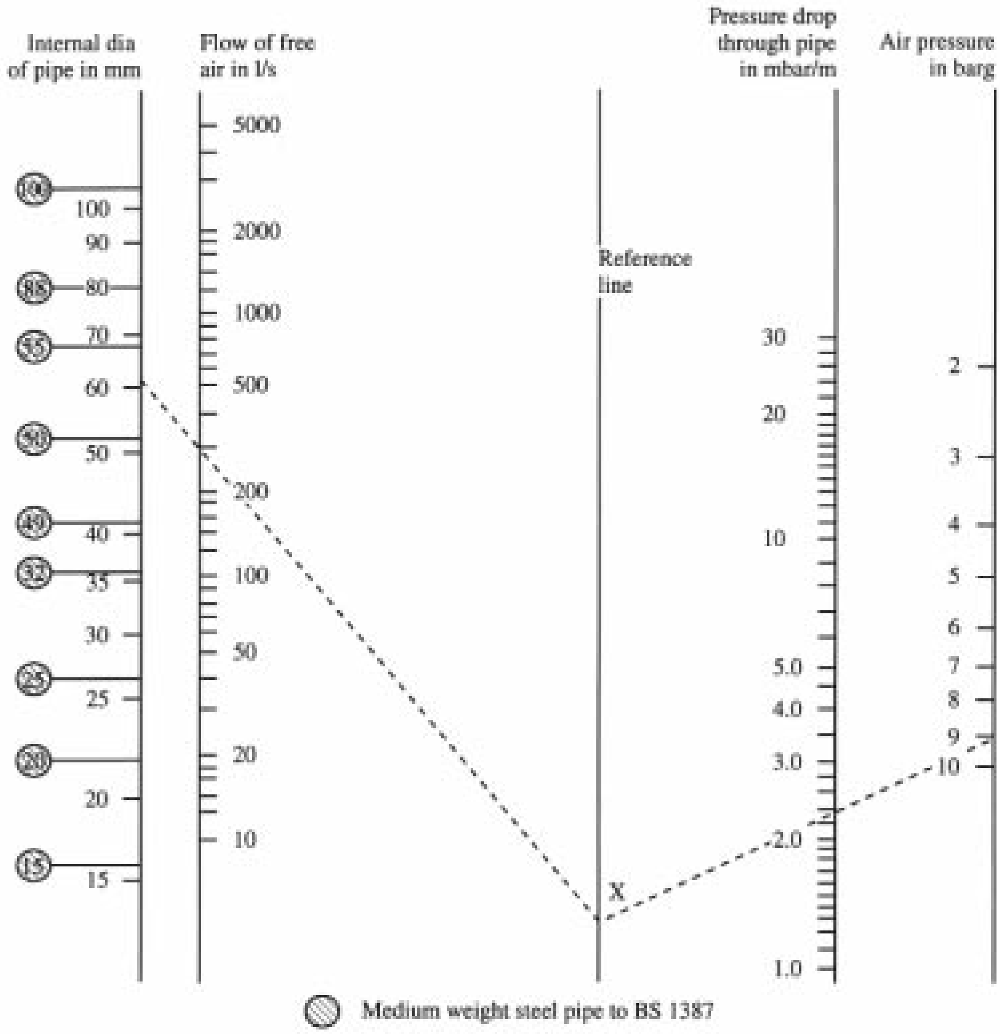


Fig 5 Pressure drop in steel pipes

When sizing ring mains it is advisable to err on the high side, as future demand may increase. The consequence of undersizing is very significant in energy efficiency terms, because air velocities will be excessive leading to high pressure drops and non-separation of condensed water. Table 4 shows the power wasted for different pipe diameters with a flow rate of 500 l/s (1,000 cfm) at 7 barg pressure. The extra energy cost for the 50 mm pipe over a more adequately-sized one for a system operating 48 h/wk would be £1,800 per annum/ 100 metres.

Table 4 Distribution line power wastage (@ 500 l/s, 7 barg)

Pipe Nominal Bore (mm)	Pressure Drop per 100 m (bar)	Equivalent Power Lost (kW)
50	2.6	18
65	0.9	5
80	0.2	0.8
100	0.1	0.4

Compressor house to ring main feeds should be oversized to avoid pressure drop problems. As a guide it is recommended that a pipe of twice the cross-sectional area of that used for the ring main is selected.

Higher air velocities (up to 20 m/s) are acceptable where the distribution pipework does not exceed 8 metres in length. This would be the case where dedicated compressors are installed near to an associated large end user.

4.2 Water Drainage

For undried compressed air, system problems may occur through water condensation in the distribution network. It is good practice to remove as much of this water as possible. Condensation typically occurs where the air main travels outside the buildings and is therefore subject to temperatures below those in the compressor house. With refrigerant-dried systems, if temperatures fall to below 2°C there will be further water condensation.

The accumulation of condensed water and scale in the system can lead to pressure drop. To prevent this occurring, the air mains should slope to strategically-located drain legs equipped with automatically operated drain valves (avoiding air wastage which frequently occurs with manually operated drain valves).

Building layout will dictate the best position for drain points, but in general the main should be installed with a fall of not less than 1 metre in 100 metres of ring main pipework in the direction of air flow. The recommended distance between drain points is approximately 30 metres.

Any branch line should be taken off the top of the main to prevent any water in the main pipe from running into it. In addition, the bottom of a falling branch line pipe should be drained, as shown in Fig 2.

4.3 Drain Traps

For the sake of energy efficiency, automatic drain traps must be fitted to the bottom of the drain line within a compressed air system. Reliable electronic condensate traps are available that ensure water is regularly drained away. Manual traps left open account for a substantial percentage of total wastage.

The ball float type of drain trap is the most common because it gives a positive shut off, opening only if water is present and closing immediately the water has cleared. If, however, there is a possibility of large quantities of water at the trap, then air binding can take place and a balance pipe must be fitted. With this arrangement, the water will flow freely into the trap, displacing air which then passes through the balance pipe and into the main system (see Fig 6).

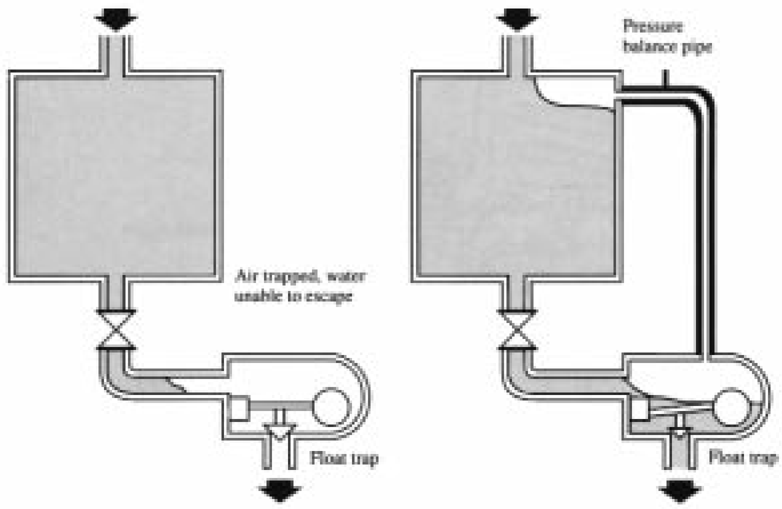


Fig 6 Air binding of water traps

All drain traps require occasional maintenance, to remove any build-up of oils or emulsions which may be present in the condensate. If oil/emulsion build-up is heavy, consideration should be given to using drain traps which have a blast action discharge.

4.4 Receivers

Receivers have three main functions:

- providing storage capacity;
- acting as a secondary cooler;
- creating more stable pressure conditions, effectively acting as a pulsation damper.

On most installations, the receiver is fed from the after-cooler and further cooling will take place in the receiver. On installations where the compressor plant is small, an after-cooler may not be fitted, making the receiver the point at which most condensed liquid will be found. If liquid is allowed to build up in the receiver, carry-over into the ring main is likely, with resulting efficiency implications. The receiver, therefore, needs to be fitted with both automatic and manual drain traps, in order to remove the condensate and any carry-over solids such as dust, scale, carbon and so on.

The function of the receiver as a pulsation damper is particularly important when installed in conjunction with reciprocating compressors and rotary screw compressors with on/off line control.

4.5 Regulators

Some applications, such as control and instrumentation, require air at a pressure lower than the main supply and in these circumstances, if a separate pressure system cannot be installed, then regulators should be fitted. Fig 7 illustrates a direct-acting regulator.

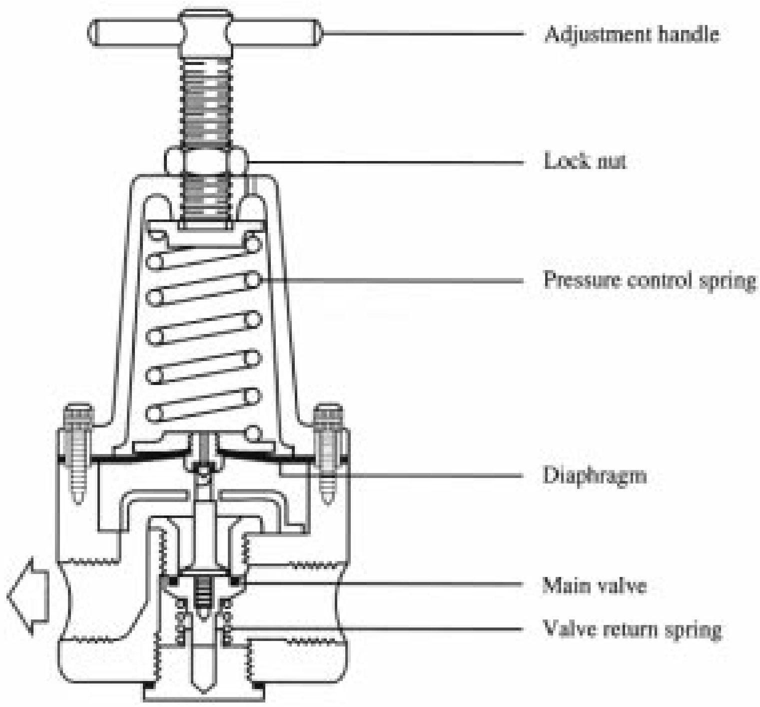


Fig 7 Compressed air regulator

Excellent savings can be achieved by reducing pressure at the usage point. The air consumption of most air-using devices, such as air tools, spray guns and air knives, increases in proportion to the operating pressure ratio. Reducing pressure to the minimum acceptable saves energy.

Example:

A 6 mm nozzle will consume 43 l/s at 7 barg and 38 l/s at 6 barg; a reduction in air consumption of 14.5%.

The annual savings achievable by this reduction in pressure would be £3,500 (based on the example demand of 500 l/s for 48 hours per week).

5. COMPRESSED AIR TREATMENT

Air treatment is normally required in order to provide compressed air of the quality required at the point of use. This avoids product contamination, product spoilage and poor control of, or damage to, air-using equipment. Treatment needs energy in terms of additional generation pressure, and possibly additional compressed air or electrical demand depending on the type of treatment used.

Compressing air concentrates the contaminants per unit of volume of air delivered at pressure. Compressed air can be contaminated by: water vapour; condensate; particulate matter (either air-borne or pipe-scale); oil in vapour or liquid state (either inhaled by the compressor from the atmosphere or added during compression); and microbes. The amount of treatment will depend on the users needs.

Table 5 gives the ISO/DIS recommendations on air quality classes.

Table 5 Air contamination classifications ISO 8573 - 1

QUALITY CLASS	DIRT Particle size in micron	WATER Pressure Dew-point °C (ppm vol) at 7 barg	OIL (including vapour) mg/m^3
1	0.1	-70 (0.3)	0.01
2	1	-40 (16)	0.1
3	5	-20 (128)	1
4	15	+3 (940)	5
5	40	+7 (1,240)	25
6	-	+10 (1,500)	-

The requirement for high quality compressed air is increasing as production methods become more sophisticated. A general breakdown of recommended standards for different manufacturing applications is included in Table 6. This table is intended for guidance only; in practice there are very many other combinations needed.

There is a very wide range of requirements for air quality. It is important to install the right equipment, and equally important to keep the energy requirements within reason. Every effort should be made to avoid unnecessary levels of treatment.

Table 6 Pneurop recommended standards

Application Classes	Typical Quality Classes		
	Oil	Dirt	Water
Air agitation	3	5	3
Air bearings	2	2	3
Air gauging	2	3	3
Air motors	4	4 - 1	5
Brick and glass machines	4	4	5
Cleaning of machine parts	4	4	4
Construction	4	5	5
Conveying, granular products	3	4	3
Conveying, powder products	2	3	2
Fluidics, power circuits	4	4	4
Fluidics, sensors	2	2 - 1	2
Foundry machines	4	4	5
Food and beverages	2	3	1
Hand operated air tools	4	5 - 4	5 - 4
Machine tools	4	3	5
Mining	4	5	5
Micro-electronics manufacture	1	1	1
Packaging and textile machines	4	3	3
Photographic film processing	1	1	1
Pneumatic cylinders	3	3	5
Pneumatic tools	4	4	4
Process control instruments	2	2	3
Paint spraying	3	3	3
Sand blasting	-	3	3
Welding machines	4	4	5
General workshop air	4	4	5

The type of compressor used is important. An oil-free machine could save one filtration stage over an oil-injected compressor, so for high quality air requirements an oil-free machine should be purchased wherever possible. In addition to the treatment savings, other benefits include increased efficiency and longevity, and in cases where a desiccant dryer is used there is no chance of contamination of the desiccant by compressor oil. However, when oil-free machines are used in heavily polluted atmospheres, it is still necessary to remove oil by filtration.

Many plants need only part of the air treated to very high quality. In these cases, excellent savings can be achieved by treating all the generated air to the minimum acceptable level and improving the quality to the desired level close to the usage points.

Example:

A car plant requires 70% of its compressed air at 4.4.2 quality, which can be supplied by a refrigeration dryer and an oil removal filter. 30% of the air is needed at 1.2.1 quality for the paint and engine assembly areas. The desiccant dryers and special filtration needed for this quality can be installed in those areas only, resulting in power cost savings of some £2,000 per annum for every 500 l/s (1,000 cfm) delivered. Savings will also result from reduced dryer maintenance and consumable parts, such as filter elements and desiccant replacement.

Ambient air typically contains 12.5 g of water for every 1 m^3 of free saturated air at 15°C. If a compressor produces air at 500 l/sec (1,000 cfm), the compressed air produced each hour will contain 22.5 l of water. Provided the free air is maintained at about 15°C, the water will remain as vapour; however, if the air is cooled or compressed the water will be condensed. The temperature at which water condenses is known as the dew-point.

The rise in air temperature in the compressor generally prevents condensation, but when the air passes though the after-cooler a large amount of water condenses. Typically the air temperature following the after-cooler will be around 35°C and water content will have been reduced. Water will, however, still be present as vapour and if the air temperature falls there will be further water condensation. Fig 8 shows the amount of water removal required for differing end temperatures.

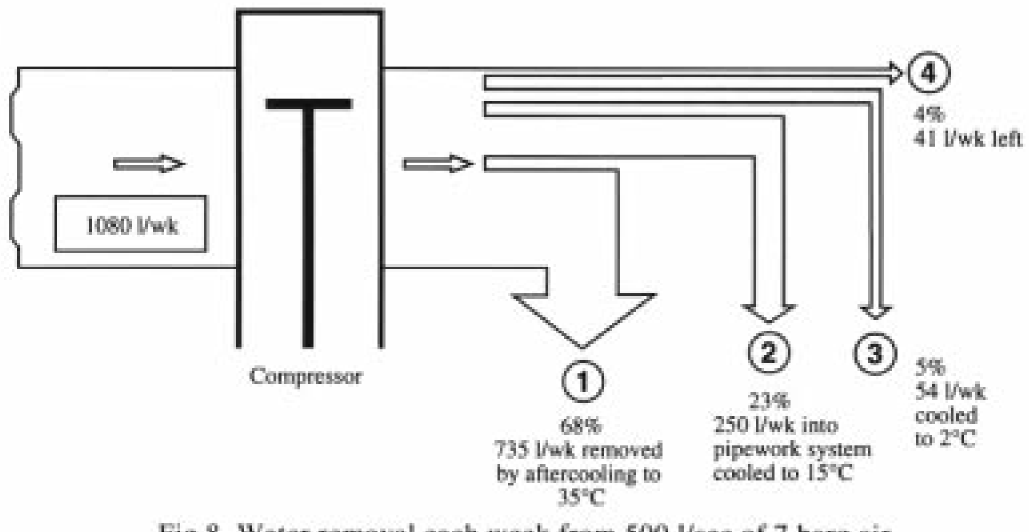


Fig 8 Water removal each week from 500 l/sec of 7 barg air

To remove significantly more water from the compressed air than can be achieved by after-cooling, a dryer is necessary. Air dryers typically take the air from the compressor after-cooler at a maximum temperature of 35°C. Dryer types and their relative merits are discussed in Section 5.1.

5.1 Dryers

5.1.1 *Desiccant Dryers*

For class 1.1.1 quality air, heated or heatless twin tower desiccant dryers with special desiccant and drying cycles are employed. Oil removal filters, water removal filters, dust removal filters and an activated carbon absorber unit are also needed. This type of system consumes a lot of energy, requiring up to 15% of the compressed air or electrical equivalent for desiccant regeneration, and there is a pressure drop across the filters, when in service, of up to 1.5 bar which will require additional generation pressure.

The example 500 l/s compressor, when supplying 1.1.1 quality air, would cost £4,200 (21%) per annum more due to the treatment devices, than if it were simply delivering after-cooled air.

As the requirement for air quality becomes less intense than 1.1.1, the energy requirement reduces.

The example compressor, when supplying air of 2.2.1 quality, where a lower specification desiccant dryer with pre- and after-filters performing at -40°C pressure dew-point could be used, would only cost 15% more than delivering after-cooled air.

5.1.2 *Sorption Dryers*

Air of 2.3.1 quality can be provided with a sorption dryer which can only be used with an oil-free compressor. A drum, impregnated with the drying medium, is slowly rotated by a very small motor. Compressed air is fed through a sealed segment of the drum and dried within a range of -15°C to -40°C, depending on the compressor load. The drying medium in the part of the drum not being used to dry the air is regenerated by hot air taken from before the machine after-cooler, i.e. by the waste heat of compression. The cost to provide 2.3.1 quality air by this method, with an oil free compressor and limited filtration, is around 3% more than delivering after-cooled air.

5.1.3 *Deliquescent (Absorption) Dryers*

Deliquescent dryers operate by passing the compressed air over soluble material, such as salt, which dissolves as it absorbs moisture. Their main advantages are that they do not consume any energy other than that required to overcome the pressure drop within the dryer (0.1 to 0.4 bar) and they do not lose any air volume. The dryers are not, however, regenerative and deliquescent material needs to be replaced periodically, incurring both labour and material costs. Deliquescent dryers are the least expensive dryer and are very energy efficient, but can only produce dew-points about 6°C below the inlet temperature.

5.1.4 *Refrigeration Dryers*

Compressed air of 4.4.5 quality can be provided from any type of compressor by use of a refrigerated dryer and, where necessary, oil removal filtration. The extra cost will typically be 5% over delivering standard after-cooled air.

This method of drying is very popular as it produces dew-points which are adequate for most duties in an energy efficient and reliable manner.

The dew-point produced is above winter ambient temperatures; therefore, if air distribution pipe work is taken outside buildings water will condense. It is recommended that a condensate trap is fitted to the system where it enters the next building to prevent problems.

5.2 After Filters

Filters cause pressure drops in compressed air systems. To save energy, it is recommended that only the minimum filtration requirement is met. Filters should be adequately sized for the duty; if the filter connections are considerably smaller than correctly-sized pipework they will cause pressure drops. It is better to pay more for filters with correctly-sized flanges, and avoid pressure drop and energy wastage.

5.3 Air Intakes

The location of the air compressors on site can have a bearing on the amount of energy used by the compressor. Cool, clean, dry intake air will lead to more efficient compression. Where possible, air should be taken from outside the building because its temperature will be lower. Lower temperature air is denser and the mass of air that can be compressed by the machine is increased. If the air inlet is on a north-facing wall, it should be protected to prevent rain and wind blown dust entering, clogging filters and thereby wasting energy. Ducting between air intake and compressor should be short, straight and of generous diameter. The condition of the air entering the compressor is extremely important, since fouling of inlet filters and high ambient air temperatures can result in significant energy wastage.

For every 4°C drop in intake temperature, there is a 1% increase in efficiency. For the example 500 l/s compressor operating 48 hours per week, this would save an extra £200 per annum.

For every 25 mbar pressure lost at the inlet, the compressor efficiency is reduced by 2%. This reduction is equivalent to £400 per annum for the example compressor.

5.3.1 *Air Inlet Cooling*

With air inlet cooling or pre-cooling, the air is cooled prior to the compression process resulting in energy savings and improved air quality. Air inlet cooling is carried out by twin two-stage coolers linked to a conventional refrigeration plant. Inlet air is cooled to between -20°C and -25°C, which reduces the pressure dew-point to about 0°C and, as an added benefit, removes any dust particles (these collect in the water/ice mixture on the cooler tubes). The density of the inlet air is increased by around 15%, thereby improving the volumetric efficiency of compression. Pre-filters and after-coolers are not required and can be removed from the compressors. As there is no after-cooler, the compressed air leaves the machine at 100°C to 120°C and then cools within the distribution system. Some of this air may be utilised at above ambient temperatures, further improving the system efficiency due to its increased volume. In these cases, oil must be filtered from the air to avoid the risk of explosion.

Air inlet cooling technology is particularly worthwhile on low pressure blowers and single stage compressors, but it is not so suitable for multi-stage units. It is not recommended for centrifugal compressors because, although volumetric flow increases, the machine will be functioning well away from its design condition and will therefore be inefficient. It is important to ensure that the inlet air temperature is not reduced below the minimum recommended by the compressor manufacturer, otherwise problems may occur with possible damage to the compressor and overloading of the main drive motor.

To justify air inlet cooling, high annual utilisation is needed, producing payback on the capital cost of between two and five years depending on local conditions.

5.4 Treatment Systems Maintenance

Fouling causes an increase in the pressure drop across all filter elements, leading to a higher generation pressure and additional energy use. It is important to keep pressure drops to a minimum. All filters should be fitted with differential pressure gauges which should be calibrated regularly.

With all dryers, particularly desiccant types, the dew-point should be checked regularly. Many will be operating well below specification, and yet the energy consumed will be continuing at the same rate as that needed for design dew-point performance.

With desiccant dryers, the purge cycle should be monitored to ensure that it does not become excessive and waste energy. During the purge cycle dried air is passed through the desiccant columns in order to regenerate them. Automatic purge cycle control, based on delivered dew-point, is available from several manufacturers and is worth installing to save energy.

6. COMPRESSED AIR GENERATION

Opportunities for more efficient energy use arise when new compressed air facilities are being designed, particularly when choosing the most appropriate compressor. The features of the main compressor types and their efficiencies are included in this Section.

6.1 Positive Displacement Compressors

Reciprocating piston, rotary vane, toothed rotor and rotary screw compressors are all positive displacement machines, but have very different configurations. In principle, a multi- stage compressor, with small clearance volumes, is more efficient than a single-stage machine.

6.1.1 *Reciprocating Piston Compressors*

Capacities of 25 to 250 l/s can be served by compressors of the single- or two-stage air-cooled piston type which are usually receiver mounted. It is possible to get non-lubricated piston compressors for duties such as food, air conditioning and pharmaceutical production; however, in these cases it is more common to use an oil-injected compressor with filtration to remove the oil carried over from the compressor.

From 250 to 1,000 l/s double-acting water-cooled piston compressors either in lubricated or non-lubricated forms are available. These machines are the most efficient available in terms of full-load and part-load power consumption.

A high number of piston compressors are in use today. Well-maintained piston machines are still the most energy-efficient compressors, although efficiency decreases significantly if they are poorly maintained. Over the last decade, however, the trend has been to purchase rotary vane, screw and centrifugal compressors, because they are quieter and simpler to maintain and install.

6.1.2 *Rotary Oil-Injected Compressors*

Rotary machines are not as efficient as well-maintained large piston compressors, but by using waste heat recovery (normally simple with rotary vane and screw machines) the efficiency deficit can be made up.

Capacities from 25 to 250 l/s can be served by single-stage oil-injected rotary vane or screw compressors. From 250 l/s up to 1,000 l/s duties are served by oil-injected screw machines.

Many users who need high quality air use filtration on oil-injected machines, because of the lower capital cost of the machinery. This method of providing high quality air is less energy efficient than using true oil-free compressors.

6.1.3 *Rotary Oil-Free Compressors*

For 100 l/s to 2,000 l/s oil-free applications the two-stage rotary toothed rotor compressor or non-lubricated screw compressor can be used. Their efficiency is high because compression takes place in two stages and the rotors have very close operating clearances. They are as efficient as oil-free piston machines and have a long lifetime, but have a high capital cost compared with oil-injected machines.

6.2 Dynamic Compressors

For efficient operation, it is important to state the site ambient temperatures and pressures, and the design flow and pressure when specifying dynamic compressors. The energy requirements and control range of these compressors are seriously affected by operation outside design conditions.

Generally for energy efficiency at full and part load, the more stages of compression the better.

Centrifugal machines are available from 250 l/s up to very large capacities, and are popular and most energy efficient for applications over 1,000 l/s. Capacities over 2,000 l/s can be met by multi-stage oil-free centrifugal compressors, until very large mass flow compressors of the axial flow configuration come into consideration. Centrifugal compressors are very reliable and efficient if properly applied, and usually have low maintenance costs.

6.3 Compressor Choice

In general, the choice of compressor and after-treatment system is dictated by:

- the capacity and pressure required;
- the capital available;
- the specified delivered air quality requirements.

The relative generation efficiencies of each different compressor configuration are summarised in Table 7.

Table 7 Summary of compressor configurations with relative efficiencies

Description	Capacity l/s	Specific Power J/l *	Part load efficiency
Lubricated piston	2-25	510	Good
	25-250	425	Good
	250-1000	361	Excellent
Oil-free piston	2-25	552	Good
	25-250	467	Good
	250-1000	404	Excellent
Oil-injected vane/screw	2-25	510	Poor
	25-250	446	Fair
	250-1000	404	Fair
Oil-free toothed rotor/screw	25-250	429	Good
	250-1000	382	Good
	1000-2000	382	Good
Oil-free centrifugal	250-1000	446	Good
	1000-2000	382	Excellent
	Above 2000	361	Excellent

* J/l ÷ 21 = kW/100 cfm
e.g. 510 J/l = 24.29 kW/100 cfm

Efficiencies are based on specific power consumption (joules/litre (J/l)).

6.4 Compressor Control

For energy efficiency, it is important to consider the control of individual machines and the way in which multiple installations meet the demand in terms of flow and pressure requirements.

For a relatively low capital outlay, a modern compressor control system can save between 5% and 20% of the total generation costs, and in some cases improved pressure control will also result in productivity gains.

6.4.1 *Individual Compressor Control*

Using variable speed drive (VSD) motors to drive piston and screw compressors offers many control and efficiency advantages. In the past costs have been prohibitive; however, new advances in electronics and control gear are making these systems more popular. Care should be taken not to reduce the compressor speed to the extent that it is inadequately lubricated. VSDs are unsuitable for centrifugal compressors unless specifically designed for the purpose.

Piston machines with two- or three-step inlet suction valve unloaders, on- or off-line inlet valves, or five-step clearance pocket unloading, give the best efficiencies at part loads. Piston, vane and screw machines with variable inlet throttle valves that modulate over a close pressure range are not efficient on low loads, because they are positive displacement machines and throttling causes an increase in compression ratio.

Rotary screw machines are often fitted with both two-step unloading and modulating control with a manual or automatic change over switch. Modulation should only be used if the load is over 75% of its design level: below this, two-step unloading is more efficient. Two-step systems, on any machine operate over a pressure differential of around 0.5 bar between full and no load. A correctly-sized air receiver should be fitted to avoid control hunting.

Centrifugal compressors are dynamic machines and behave efficiently on part load. Out put is normally reduced by modulation to 70% of the design flow. For installations where the demand is sometimes less than this, machines with automatic dual control systems should be installed to avoid wasting energy due to by-passing of pressurised air at part loads. Inlet guide vanes are preferable to inlet throttles, because they improve the part load efficiency and turn-down range, particularly at off-design inlet conditions.

Most compressors up to 1,000 l/s can be switched to automatic stop/start control; some machines change to this mode automatically. Automatic stop/start control stops the compressor after a period of no-load running, usually 10-15 minutes, and then automatically restarts the machine on a demand for air. The off-load running time is essential, unless a soft starter is fitted, to protect the drive motor from too many starts.

6.4.2 *Multiple Compressor Control*

Various forms of automatic sequencing control exist for optimising the operation of multiple installations and equalising the wear through rotation of the sequence.

Microprocessor-based systems have much more accurate pressure control than pressure switch or air governor controls, and avoid large pressure differentials and energy waste. They can take into account lower pressure requirements during non-productive hours and adjust accordingly, and can also control system isolation valves.

Some multiple machine control systems work with a combination of pressure and demand signals to ensure that only the correct number of machines are on-line at any one time.

Computer-based systems are available that save energy by reducing the period of time that machines in a multiple installation are running off-load. This is achieved by predictive switching which shuts a machine down immediately it goes off-load. When the demand increases, the next available machine in the rotational sequence will start, enabling the first to stay off, reducing the possibility of motor damage due to rapid switching and eliminating the need for no-load running. The system will also select the most suitable number and sizes of compressors in a multiple installation to meet the demand, thereby minimising energy costs.

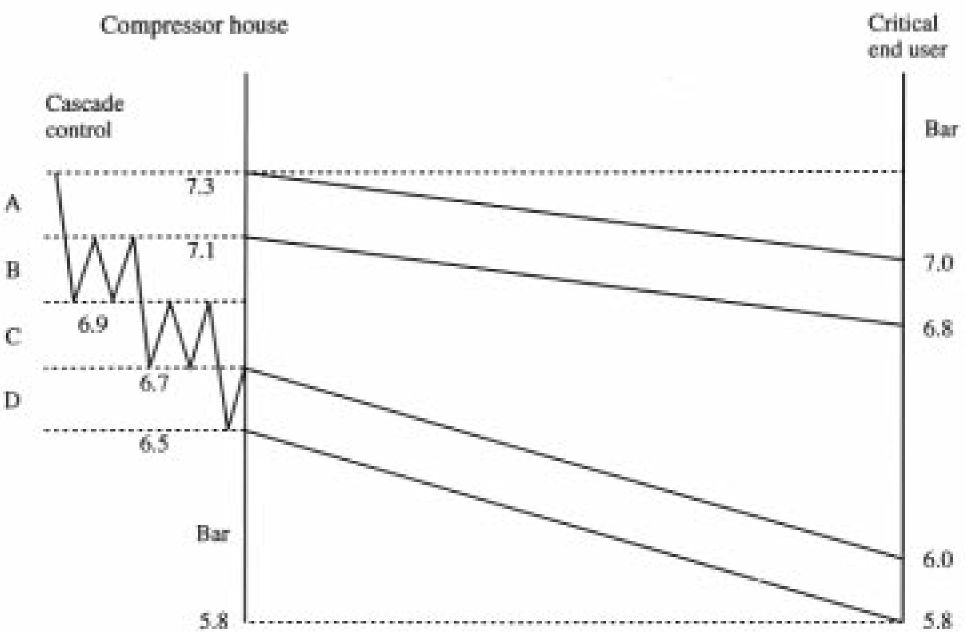
Control systems can be built into building management systems along with compressor condition monitoring, automatic operation of zone isolation valves, compressor electric motor input readings and departmental air demand metering from remote outstations.

6.4.3 *Pressure Control*

Energy can be saved in most multi-compressor installations by improving the pressure control system. The majority of systems have two basic shortcomings:

- pressures are maintained at a higher level than is needed by the end users;
- generation pressures are set too high and not varied according to demand.

The consequence of both of these failings is demonstrated in Fig 9, which represents a typical cascade control system.



A) No.1 Compressor ON/OFF load
B) No.1 Compressor ON load
No.2 Compressor ON/OFF load
C) No.1 & No.2 Compressor ON load
No.3 Compressor ON/OFF load
D) No.s 1,2, & 3 Compressors ON load
No. 4 Compressor ON/OFF load

Fig 9 Cascade pressure control typical situation

Usually the lowest acceptable pressure at the compressor house is set by the minimum pressure needed by the most critical piece of machinery. To ensure that this pressure is achieved at the critical point, the pressure at the compressor house (i.e. the control pressure) will need to be set to 0.7 bar (10 psi) above the value required, to overcome line pressure losses on the distribution network, including occasions when the air flow rate is highest and hence the distribution losses are greatest. For example, if 5.8 barg (85 psig) is required, the control pressure will need to be 6.5 barg (95 psig). This control pressure compensation is shown in Fig 9 by the lower two lines.

The left-hand axis in Fig 9 shows the cascade switching of the compressors and it represents a typical cascade pressure control system. At maximum flow rates all four compressors will be operating, as designated by region D. As the air usage falls, the pressure in the lines at the compressor house will rise and one compressor will unload. If usage falls further, the pressure moves into region C where another compressor unloads, and so on up to region A where only one compressor is providing the load. Due to the nature of the pressure switches, the minimum control bands are 0.2 bar for each compressor leading to a very wide overall control band (0.8 bar).

The consequence of a wide control band is that, except at maximum air usage, the end user and compressor control pressures will always be up to 10% higher than those actually required. This has two negative effects: firstly, it takes up to 5% more electricity to generate the air at a 10% greater pressure (see Table 8), and secondly the usage of air in most applications is directly proportional to its pressure. The worst case scenario, shown by the upper two lines in Fig 9 which may represent night time and weekend usage, could be costing at least 15% more than is necessary.

Table 8 Generation pressure savings

Pressure Barg (psig)	Energy Savings (%)	
	Single Stage	Two Stage
7 (100)	-	-
6 (90)	5	5
5.5 (80)	10	11

To keep generation costs to a minimum:

- pressure control should be based on the pressure at the most sensitive/critical pieces of machinery;
- compressor sequencing should be based on as narrow a pressure band as possible to achieve the minimum generation pressure at all times.

6.5 Sizing

Compressors should be sized as closely as possible to the duty. It is not economical to run any machine for long periods at low loads, due to electric motor inefficiencies. The off-load power can be 15% - 70% of the on-load power once motor inefficiencies have been taken into account.

For new installations with multiple compressors, it is worthwhile considering installation of a selection of unit sizes, so that the demand can be met by compressors operating close to full output. Care should be taken to ensure that the overall system efficiency is improved, taking into account the lower generating efficiencies of some smaller compressors.

6.6 Maintenance

Compressors run for many hours, often in appalling conditions. Equating the example 500 l/s compressor to a motor car, it would cover over 70,000 miles per annum at an average speed of 30 mph. Some compressors run the equivalent of 250,000 miles per annum on this basis. Good maintenance is therefore essential.

Piston compressors, particularly the oil-free type, suffer the most in efficiency terms from lack of maintenance.

If the example 500 l/s demand was served by a poorly maintained, oil-free piston compressor, compressor efficiency would deteriorate from 400 J/l to 450 J/l over a 12 month period, adding over £2,000 to the annual running costs.

The efficiency of rotary vane and screw machines does not deteriorate so rapidly; however, there is a finite life for such compressors. As a guide, these types of machine must receive major maintenance after 25,000 hours life to maintain good efficiency. Oil-free toothed rotor and screw machines perform well for periods of up to 40,000 hours, after which there is a slow fall off in efficiency due to gradually increasing internal clearances. These types of machines then need major refurbishment to maintain efficiency.

Centrifugal compressors, having few moving parts and comparatively large 'as built' clearances, will maintain their efficiency over longer periods. The inlet air filters, cooling water system and the inter-coolers must be rigorously maintained or efficiency will fall off rapidly.

It is a false economy to ignore maintenance on any type of compressor. It is recommended that manufacturers, or their accredited agents, are used for service work and that genuine spare parts, to the original design, are used. An apparently cheaper component, such as an incorrectly designed replacement discharge valve, costs more in the long term due to the detrimental effect that it has on the compressor efficiency.

6.7 Heat Recovery

Over 90% of the energy consumed by a compressor is turned into heat. This heat is low grade and is usually wasted, but in many cases it can be recovered. Uses for hot water or air should be considered together with compressor location, particularly when designing new installations.

The cost of linking a compressor to a suitable heat user will vary, but payback periods of between one and three years on the total marginal costs are typical. At the moment, however, compressors with integral heat recovery systems account for only 10 - 12% of total compressor sales within the UK.

6.7.1 *Inter-coolers / After-coolers*

Multi-stage compressors have inter-coolers that improve the overall efficiency of the compressor by reducing the inter-stage temperature, and in the case of positive displacement machines, by lowering the pressure of the air in the clearance spaces (improving the volumetric efficiency of the compressor).

After compression, the air is generally after-cooled, to reduce its temperature and remove condensed water. Compressors themselves can be air- or water-cooled. On water-cooled compressors heat is normally removed from the compression cylinders by water jackets. Hot water at around 50°C can be collected from a piston compressor and used for a variety of purposes, including increasing the temperature of boiler feed water, process water or domestic hot water. Warm air can be ducted from air-cooled compressors, particularly packaged rotary machines, and can be used for duties such as space heating and air curtains.

If the example 500 l/sec compressed air installation was served by an air-cooled compressor, the available heat would be typically 130 kW at between 30 to 40°C. For a site using direct gas-fired space heating, operating 48 hours/week, the use of this heat would produce annual savings of around £2,000.

During the summer months, any hot air should be ducted to atmosphere, otherwise it will be dissipated to the surrounding area and could subsequently be drawn back into the compressor preventing adequate cooling and increasing generation costs.

6.7.2 *Use of Hot Compressed Air for Process Duties*

Some process applications, such as spool valves in a glass factory or drop forging hammers, benefit from hot compressed air. Hot compressed air is not often used because of safety concerns, since there is a risk of compressor oil carry-over spontaneously igniting if the discharge temperature is too high.

If hot compressed air is to be used, all the air pipework should be lagged to prevent cooling and an over-sized condensate recovery system should be fitted to take care of the additional condensate which will form. An after-cooler will not be required. Using hot air is especially worth considering if the air is compressed near to the point of usage and the pipe runs (and hence the heat and pressure loss) are therefore small. The volumetric increase achieved by using hot air will save up to 25% of the energy used on the duty, provided the air is kept hot up to the usage point.

6.8 Site Integration

It is important that site integration of air compressors is considered at the planning and design stages to get the maximum benefit of using packaged compressors with integrated heat recovery systems. These packaged systems generally use rotary screw compressors, which have the additional benefit of operating with low noise.

When there is a choice between using a central installation or smaller compressors nearer to the points of end use, the choice will largely depend on:

- the physical layout of the factory;
- the disposition of off-takes;
- the compressor operating hours.

Smaller compressors are generally less efficient than larger ones, and this must be taken into account when justifying decentralisation. Where practicable, air compressors should be sited near to points of large air demand and, if possible, near to heat demands to reduce pipe runs and also capital and running costs.

7. MONITORING OF COMPRESSED AIR SYSTEMS

7.1 Monitoring

Without sufficient instrumentation on compressed air systems it is impossible to determine whether or not they are operating efficiently. Throughout this Guide there are many examples of where energy is wasted; without a mechanism for detecting wastage, it is likely to continue and minimum energy costs will not be achieved. This Section sets guidelines for the minimum amount of metering to be installed on compressed air systems.

7.1.1 *Compressors*

New compressors, particularly those within an integrated package, are now supplied with all the required instrumentation. For existing plant, however, the following list gives the recommended minimum instrumentation:

- pressure gauge on the receiver;
- water temperature gauges in the compressor cooling jacket and the after-cooler, to detect any blockages that may be occurring;
- air temperature gauges at the outlet of the compressor and in the receiver, to help detect any fouling of the after-cooler heat exchanger;
- inter-cooler pressure and temperature gauges where applicable;
- pressure gauges at selected points along the distribution system, to obtain a pressure profile across the site and hence identify high pressure loss areas;
- hours-run meters that differentiate between on-load and off-load running times;
- kWh meters (on compressors rated above 50 kW (150 l/sec)).

Readings from the first four instruments should be recorded daily; hours run and kWh meters should be read weekly to find usage trends. Any sudden increases should be investigated immediately in line with the Monitoring and Targeting activities outlined in Section 7.2.

7.1.2 *Air Flow Meters*

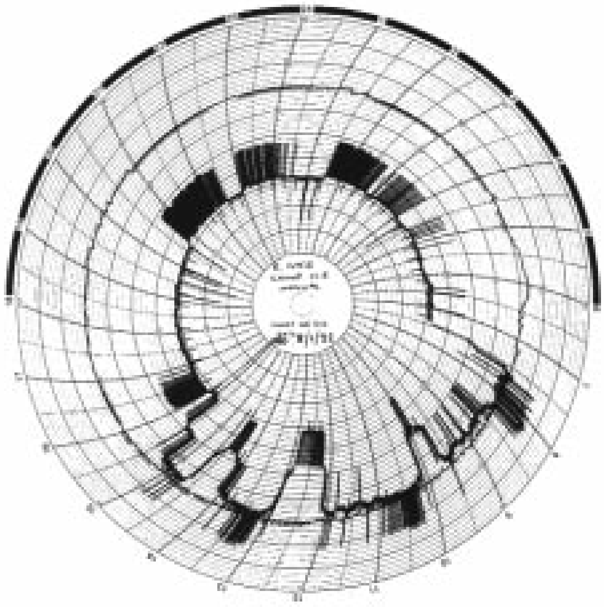
Many different forms of air flow meter are available, each with its strengths and weaknesses. The four most common are:

- pitot tube (or Curnon meter);
- turbine meter;
- orifice plate;
- vortex shedding meter.

All of the above meters measure the actual air velocity, and pressure and temperature compensation is necessary to get an accurate measure of the standard air flow rate (at 0°C, 760 mm Hg).

Details of all these meters are described in greater detail in Appendix 1.

A chart recording facility attached to any of the meters will give very valuable information about the flow rate demand pattern. Recorders can be purchased to give a 24-hour (as shown in Fig 10) or seven-day circular chart recording. Fig 10 also shows static pressure which helps detect the critical pressure demand periods. It also gives a clear indication of how the compressors are coping with the demand and whether control pressures can be reduced during certain times of the day or week (see Section 6.2.2).



24 hour recording showing:

(i) SCFM - Internal line, scale x 100 SCFM; Range: 1600 - 3500 SCFM

(ii) Compressor House pressure, scale x 3 psi; Range: 102 - 110 psi

Fig 10 Daily chart recording

7.1.3 *Compressor House Air Flow Metering*

The installation of an air flow meter on the main air line from the compressor house will give two very valuable pieces of information:

- a true usage demand profile, and base leakage or minimum use information of the plant. This is preferable to estimating usage from the on/off-load running times of the compressors, as compressor output capacity must be assumed as design level, although it could be a long way from it.
- the overall generation efficiency coupled with electricity consumption readings, i.e. input energy to output air for a given demand, can be calculated and used for analysis. Care must be taken when trying to establish the efficiency of an individual compressor, because the meter will be reading plant demand, not compressor output.

It is possible to estimate the compressor output if the pressure is carefully monitored. If the system pressure is slowly falling with a compressor on load, then the whole output of the compressor, less any condensate or seal losses, will pass the flow meter and an estimate of the compressor output can be made.

It is worth installing a permanent air flow meter on generation systems rated above 200 kW (600 l/s or 1,200 cfm) operating on a single-shift system. For full-time working, systems around half this capacity would warrant a meter. This limit is based on the meter costing around 10% of the annual air costs and assumes that the information given will lead to a 10% saving and hence a 12 month simple payback period.

7.1.4 *Distribution Line Air Flow Metering*

Permanent metering of air flows around a large distribution network can be very costly. This is mainly because good practice has led to the installation of ring main systems, so many branches have to be metered to account for a reasonable percentage of the total air used.

As a guideline, a plant or area using in excess of 500 l/sec (1,000 cfm) warrants the installation of a permanent meter.

An alternative to permanent metering is to fit some means of measuring air flow in all main lines and to use a portable meter. This could mean fitting orifice plates in each line with a portable differential pressure transducer and chart recorder, or the addition of insertion points for turbine flow meters or pitot tubes. A single meter can then be attached to each measuring point in turn and initiatives taken to reduce air consumption where appropriate.

7.2 Monitoring and Targeting

Monitoring and Targeting (M&T) has a much wider application than compressed air systems and it is assumed that most readers have a basic understanding of the term. When applied to compressed air systems, M&T could be defined as:

comparing the weekly usage of compressed air (as measured by kWh or air flow meters) against a pre-determined target which reflects good practice.

In the majority of applications, compressed air usage does not vary greatly from week to week, and so the traditional method of establishing an energy index such as kWh (compressed air) per tonne of output is not relevant. However, since there is only a small weekly variance, it is sufficient to set a fixed target (a constant) and to compare the effect of subsequent measures against this target. For instance, if only kWh meters are installed, an average weekly usage over 4 weeks can be found and any subsequent variation from this average can then be reported weekly, to show any change in either air generation efficiency or end usage. An example of a compressed air usage trend showing the effect of air leakage repairs is shown in Fig 11.

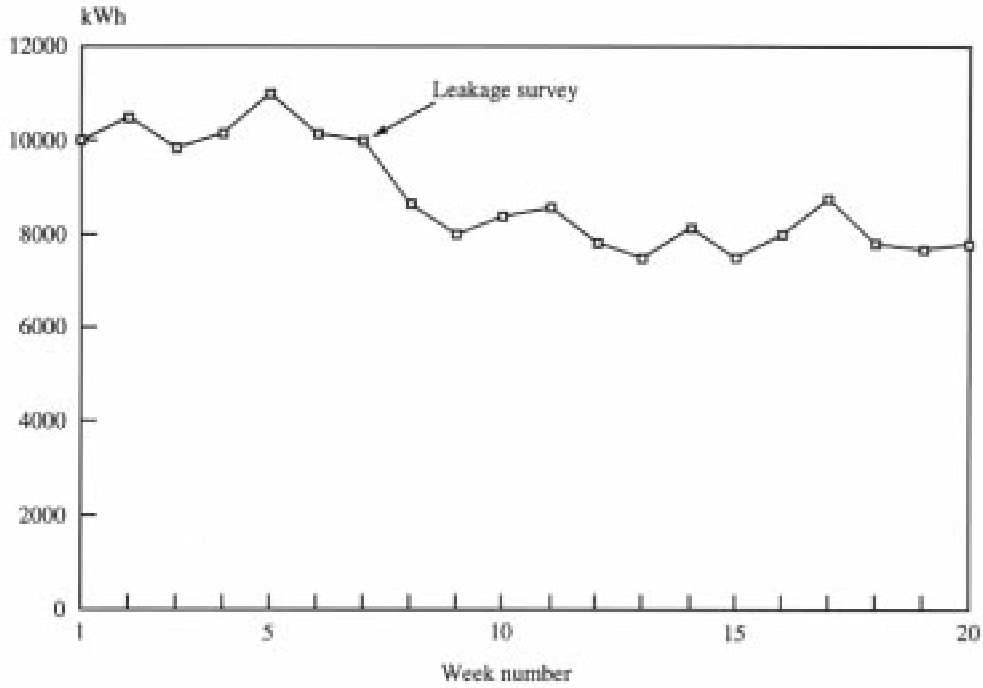


Fig 11 Compressed air trend

If an air flow meter is placed on the outlet line from the compressor house and all the compressors have kWh meters, a compressor house efficiency plot can be made, as shown in Fig 12.

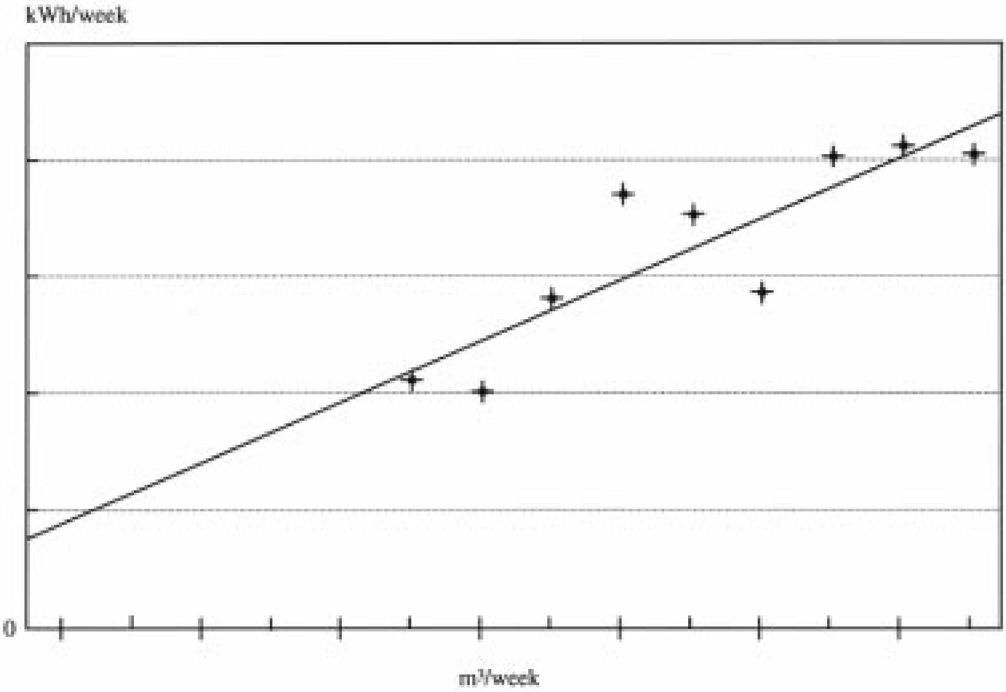


Fig 12 Targeting graph

The average efficiency (target) is shown by the regression line, which is the line of 'best fit' through the points. Points above the line show poorer efficiency and may indicate a deterioration in compressor performance which can be corrected. By altering the sequencing of the compressors to favour the use of the more efficient machines and then redrawing the graph, points below the line should be obtained, and the target can then be re-set. Reducing the generation pressure will also improve the efficiency and produce further points below the average line. Constant feedback of this nature along with the daily monitoring as outlined in Section 7.1 ensures that faults or problems are spotted quickly and put right.

If air flow meters are installed on certain plant items or in departments, individuals can be made responsible for the usage in their own area. Air usage can then be related to a measure of output from the plant, and a similar policy to that outlined in the previous paragraph can be adopted, i.e. a performance index, such as m^3/tonne, can be set and deviations from this figure can be measured. If compressed air usage is constant and cannot be related to an output parameter, an average consumption can be set and deviations from this can be measured. Measurement of air usage outside the main occupancy hours in a department will reveal leakage rates which can be targeted for improvement.

7.3 The Pressure Systems and Transportable Gas Containers Regulations

The most important clauses of the Pressure Systems and Transportable Gas Containers Regulations have been included in Appendix 2. These regulations apply to all compressed air systems where air is used at a pressure greater than 0.5 bar gauge and an air receiver is fitted. Compliance will be mandatory from 1 July 1994.

8. COMPRESSED AIR AUDITS

This Guide gives advice on where energy savings can be made in compressed air systems. To maximise these savings it is vital to adopt an approach that ensures all possibilities are investigated. This Section outlines a strategy for identifying all the opportunities, including some auditing techniques that can easily be implemented.

Ideally, before any actions are taken to improve a compressed air system, an audit should be carried out to determine the annual costs of the current system. If permanent metering is installed (see Section 7.1), this will provide an enormous amount of information to help find the best solutions. Without permanent meters, the energy consumption of each compressor will have to be estimated from the size of the motor, its average load (or its on/off times) and the number of hours it operates. The total energy costs can be calculated by adding the information for all of the compressors. It is estimated that 30% of the annual costs could be saved, providing good impetus for action.

Having calculated the annual costs and established a baseline against which improvements can be measured, an audit of the complete compressed air system should be carried out. The audit methodology is outlined below. It is best to start with the end users, because any improvements here may well have an effect on the air distribution network (i.e. redundant pipework and reduction in pressure losses) and compressor demand. It is also normally the area where the greatest savings can be achieved.

8.1 End Usage Audit

8.1.1 *Leaks*

The first priority is to assess the site leakage rate, as this is often the greatest wastage. To do this, a no-load leakage test must be carried out (see Section 3.3). From the results, the total percentage air lost and the annual costs of leakage can be calculated. Following this a leakage survey should be carried out and all leaks identified on a site drawing, as well as by tagging.

8.1.2 *End Users*

After the leakage appraisal, it is essential to look at each compressed air use in detail. The major tasks that need to be carried out are to:

- estimate the volume of major plant items, from either plant ratings or calculations, and note the number of hours they are worked, to help produce a breakdown of air usage and assist in deciding whether distribution lines are adequately sized;
- compare the actual operating pressure with the design pressure and, if appropriate, fit a reducing valve (if the overall distribution line pressure cannot be reduced);
- investigate other methods of operation not involving the use of compressed air.

8.2 Distribution Network Audit

The distribution network should be surveyed and drawings obtained in line with the Pressure Systems and Transportable Gas Containers Regulations summarised in Appendix 2. The main things to look out for are:

- zoning arrangements, to isolate areas on different working patterns;
- adequate pipe sizing and drainage: a pressure profile across the mains would be useful to identify large pressure losses;
- elimination of redundant pipework or shortening of supply lines.

8.3 Air Treatment Audit

The following programme should be carried out:

- The total air drying capacity required should be calculated during the end user audit. If more air than necessary is being dried, the possibility of having two distribution systems, a wet and a dry, should be considered. Consideration should also be given to treating the higher quality air at the point of use.
- All drainage traps should be checked to ensure that they are neither leaking nor air binding.
- The location of the air intakes into the compressors should be checked to ensure that they are not supplying warm, wet or dusty air.

8.4 Compressor House Audit

Having established the lowest possible demand profile for compressed air, it is necessary to ensure that the demand is serviced in the most efficient way possible. To do this it is necessary to carry out the following steps.

- Record the electricity consumption of the compressors over a week by installing portable recording ammeters or demand recorders on the supply cables. Over the same period also record the hours run and hours 'on-load', if meters are available;
- For systems supplying a demand greater than 500 l/s, which should have an air flow meter installed, record the actual air demand over the week. If this is not possible, estimate the air demand profile by assuming an output capacity of the compressors and combining this with the hours 'on-load' data (this is not possible if the compressors are on modulation control).
- If an air flow meter is installed, record the static air pressure over the week to establish times when the control pressure can be reduced to reflect lower air usage, as discussed in Section 6.4.3.
- From the electricity and air flow recordings, calculate an air generation efficiency. It is worthwhile running each machine individually to monitor actual output capacity and to determine the J/l (or kWh/100 cfm) performance of each machine. These figures can then be compared with those in Table 7 to assess whether each machine is good, average or poor. A simple calculation will then identify how much energy can be saved by either maintaining the poor machines or preferentially using the more efficient ones to service the demand.
- Investigate the load profiles of each compressor with a view to deciding whether the optimum size machines are running at any one time (see Section 6.5).
- Consider better methods of compressor control, such as predictive switching or rotational sequencing depending on the compressor load profiles.

9. CASE HISTORIES

The following case histories show how some industrial sites have saved money by improving the efficiency of their compressed air systems.

Case History 1 - Site: United Glass, Alloa

The United Glass factory has 16 reciprocating and two centrifugal compressors, controlled by conventional on/off loading. The plant operates 24 hours a day continuously throughout the year.

The factory management decided to investigate methods of reducing energy costs associated with the production of compressed air. This investigation was commissioned on condition that production was maintained at all times, as compressed air pressure is critical to product quality. Furthermore large sums of money were not available for capital plant.

As a result of the study of the compressed air system, the following actions were taken:

- The generation pressure was reduced. The study revealed that the air pressure at the furthest usage point from the compressor house was 46 psi, whereas the machine actually only required an operating pressure of 42 psi. Conventional on/off loading control was retained, but the operating compressors were selected according to size to give an actual generation pressure as close as possible to the new target figure. As a result, three compressors were not required.
- Automatic drain valves were installed in a phased programme, to overcome the significant leakage detected due to manual drain valves being left cracked open.
- A formalised maintenance regime was introduced for the compressors.
- A weekly monitoring programme was set up in conjunction with the new maintenance regime, covering the volumetric output and electrical usage of each compressor. Any reduction in generation efficiency could then be pin-pointed immediately and the most efficient machines used preferentially.

The introduction of these measures resulted in a reduction of £55,000 in the annual cost associated with the generation of compressed air. This represents a saving of almost 15% from the original total of £375,000.

Case Study 2 - Site: British Aerospace (Military Aircraft) Ltd, Brough

BAe (MA) at Brough have an extensive compressed air network and compressed air costs account for approximately 15% of the total electricity costs. Originally there were four independent compressed air mains, each having its own supply compressors. An attempt was made to operate two of the systems in combination using a connection pipe, but results were unsatisfactory due to the insensitive nature of the original compressor control systems in use.

BAe aimed to combine the distribution systems by investing in a microprocessor-based control system and operating the minimum number of compressors necessary to satisfy the pressure demand across the site. Once the new control system was in use BAe were able to combine three of the distribution systems into one and, by improved pressure control, reduce the demand on the compressors significantly. The new control system also allowed pressure reduction outside normal production hours and automatic selection of the most appropriately sized compressors to service the demand.

The new control system produced annual electricity savings of around £50,000 - almost 25% of the compressed air costs. Furthermore, because the control system included some condition monitoring and a remote terminal, it was possible to de-man the compressor houses, giving an additional annual saving of £50,000 on manpower costs.

Case Study 3 - Site: Jones & Campbell, Larbert

Jones & Campbell's Larbert iron foundry has seven compressors on site: four feeding a common distribution network and three individually dedicated to specific end users. The three separate compressors operated efficiently, as they had been selected to service a set duty. With the common distribution system, however, problems had increasingly occurred with meeting peak demands, particularly in the moulding section of the factory where production quality was often affected. Also, because the four compressors were running at full load for long periods, their maintenance costs were rapidly escalating. The total annual energy cost was £90,000 with an additional maintenance cost of £30,000.

A new screw-type compressor was installed which was dedicated to the moulding plant, where operating pressure was higher than the rest of the existing network, solving the production problems associated with air pressure in that area. As a result of removing the moulding plant from the network, the target output pressure for the original four compressors was reduced from 115 psig to 90 psig . A packaged control unit incorporating sequencing was installed to optimise the efficiency of the four compressors by minimising the time they were idling.

The implementation of these measures involved an initial capital outlay of around £40,000. This resulted in direct savings of approximately £20,000 per annum from both reduced energy usage and less maintenance. In addition, major savings were made in that the moulding plant no longer incurred any downtime due to low air pressure.

Case Study 4 - Site: ICI, Grangemouth

The ICI site at Grangemouth is extensive and had several compressed air networks. Over the years, however, changes both in production methods and actual processes resulted in large tracts of the systems either becoming redundant or inefficient. Management therefore decided to optimise the systems which could easily be overhauled without disrupting production. The following measures are currently being implemented.

On the instrument air distribution network, the operating pressure switches were reset to a lower value, as a result of the reduced demand around the network, and several air leaks were detected and fixed. **Estimated savings of approximately £20,000 per annum are projected from these measures.** Leakage detection and investigating the effects of removing end users from the compressed air system are now regarded as important management priorities.

On the north site compressed air system, it was noted that machines stayed in idle mode for long periods with conventional off-load system controls. A new software package incorporating rotational sequencing is now on order, so that the future system will have just one compressor running fully with the others operating in a sequencing arrangement. **The installed cost of the new control package is £25,000 and it is estimated that it will produce energy savings of £10,000 per annum and reduce annual maintenance costs by £5,000.**

Factory demand has reduced, so the existing low pressure system will be shut down. The remaining low pressure users will be fitted with the necessary reducers and then connected to the high pressure system. Removing the low pressure system and fitting reducers will cost approximately £20,000 with projected annual energy savings of £10,000.

Case Study 5 - Site: Kilmeaden Creamery, Waterford, Ireland

Kilmeaden Creamery is compact and uses compressed air for process valves in its milk processing and cheese making plants. The air is generated at 100 psi and is needed 24 hours a day, seven days a week due to complex shift patterns. The Creamery has two compressors on site; normally one 55 kW compressor is required to maintain the necessary pressures to operate the air valves.

A site-wide Monitoring and Targeting (M&T) system was installed at the Creamery and the two compressors were separately metered for electricity use. Compressed air costs were monitored on a weekly basis via the M&T software and trend graphs produced for production management and site engineers.

The costs for compressed air were running at between £200 and £220 per week during the first few weeks of the programme; however, one week it was noted that the costs suddenly rose to over £300, an increase of 50%. Investigation showed that the second compressor was coming on-load and causing a rapid rise in compressed air costs. Further investigation revealed that the inlet air filter of the first compressor was severely blocked, so it was unable to deliver the required volume at the right pressure.

After the air filter was replaced, the first compressor was able to deliver the required load and compressed air costs dropped to under £200 per week. **If the M&T system had not been in operation the second compressor would have been in operation for a number of weeks before the change was noticed, costing an extra £100 each week. If the second compressor had operated for five weeks, the additional energy costs would have exceeded the cost of installing the meters!**

APPENDIX 1

METER DETAILS

A1.1 Pitot Tube

A pitot tube is relatively easy to install. It consists of two pressure lines that measure the static pressure and the static plus dynamic (or total) pressure of the compressed air. By measuring the differential pressure between the two lines, the velocity of the air can be obtained. Pitot tubes can, however, be inaccurate if the dynamic pressure profile across the pipe diameter is not taken into account. Manufacturers' instructions should be followed to avoid this problem. A typical layout for a pitot tube is shown in Fig 13.

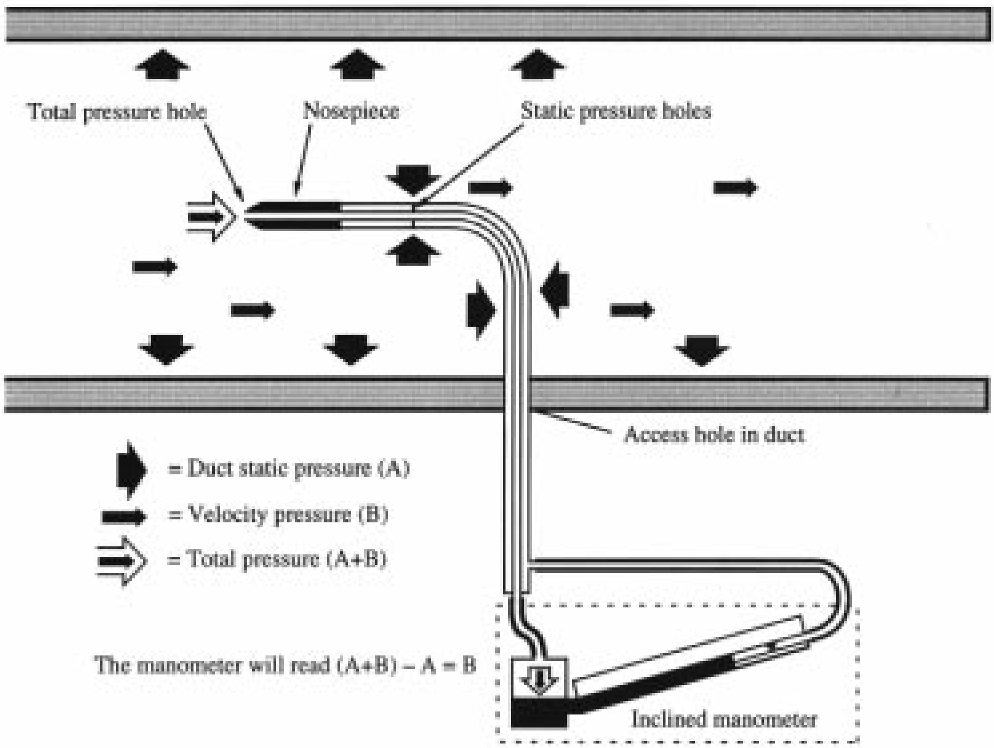


Fig 13 Pitot tube operation

Pitot tubes are often used for permanent meter installation. They give an excellent indication of flow rate and are relatively cheap. A pitot tube plus a pressure transducer costs around £800.

A1.2 Orifice Plate

Orifice plate meters consist of a drilled and machined disc inserted into the flow stream and clamped between two flanges (Fig 14). The orifice plate continues to be the most commonly used device for measuring flow rate.

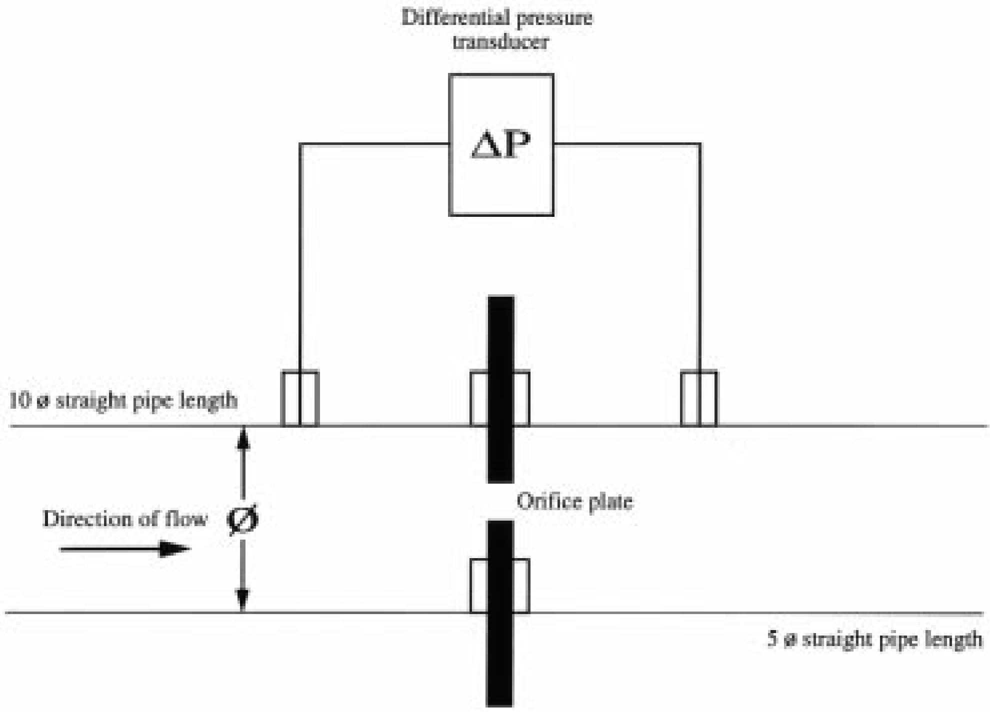


Fig 14 Orifice plate

The orifice plate itself is the main element of the meter. The resulting differential pressure is measured via impulse lines connected from the pressure tappings of the orifice plate to a differential pressure transducer.

The pressure tappings can be located in the pipeline up and downstream of the orifice plate. Alternatively, orifice plates can be supplied with corner or flange pressure tappings as part of a plate carrier ring assembly. Variations in the performance of the various tapping point locations can be used in formulae used to determine the coefficient of discharge.

The location of the orifice plate in the pipe run is also important. The differential pressure measurement is sensitive to swirl and other fluid effects, so the orifice plate should be located a certain distance away, upstream and downstream, from any pipe fitting. British Standard BS1042 and International Standard ISO 5167 provide details of the dimensions required. Orifice plate manufacturers should, however, advise on standard requirements, and reference to the published Standards is usually unnecessary. A typical requirement would be that there must be at least ten pipe diameters of straight pipe upstream of the meter and five pipe diameters downstream of the meter.

An orifice plate plus a pressure transducer would cost in the order of £800.

A1.3 Turbine Meters

Turbine meters consist of a freely-rotating propeller or screw located in the air pipe (Fig 15). Provided that bearing drag is minimised and the blades are well-designed, the process stream will exert a torque on the turbine causing it to rotate at a velocity proportional to the air flow rate. If a magnetic coil or optical device is placed in the meter housing, a voltage pulse can be induced each time a turbine blade passes it. The pulse rate will be proportional to the rate of flow and the total number of pulses can be integrated to give the air volume which has passed the meter.

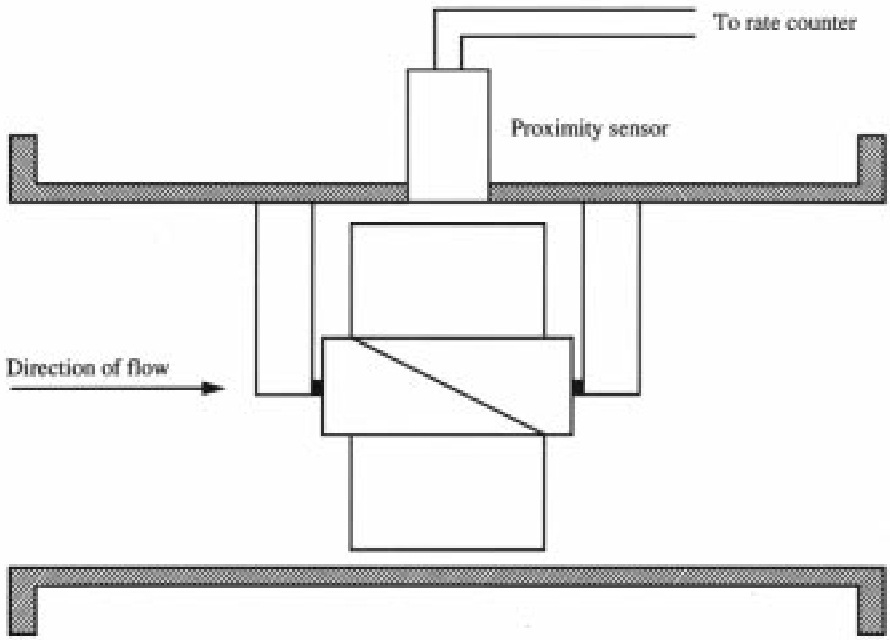


Fig 15 Turbine meter

The response of this type of meter is approximately linear, except at low flow rates when the drag effects of the bearings become significant and may affect the linearity of the response. Any non-linearity can be overcome by incorporating a calibration curve into the system which is used to convert the signal pulse into a flow rate.

The cost of a turbine-type meter is between £500 and £1,000.

A1.4 Vortex Shedding Meters

The vortex shedding meter operates on the principle that when a fluid stream flows around a bluff body (the vortex 'shedder'), viscosity-related effects produce vortices downstream (Fig 16). The most common body shapes used in such meters are rectangular or triangular. The vortices are shed sequentially from either side of the bluff body at a frequency proportional to the flow velocity.

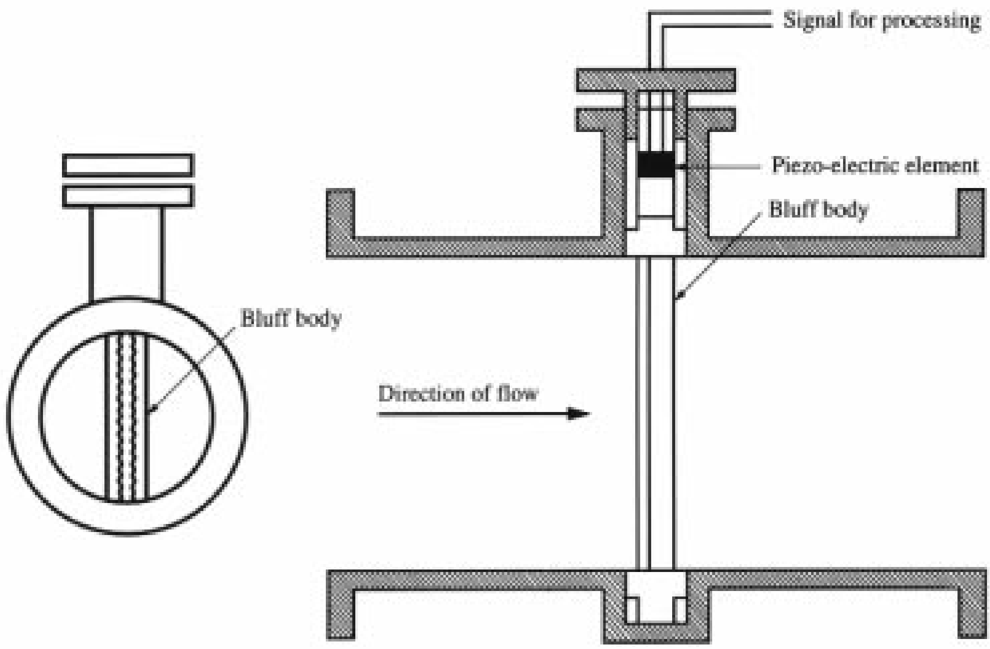


Fig 16 Vortex meter operation

Several methods of sensing the vortices exist.

A common method is to use a piezo-electrical cell located in the bluff body support spindle. Shedding of the vortices creates lift in the bluff body which in turn causes small movements to the spindle. Each movement compresses the cell, thereby generating a small electric current.

Other methods use ultrasonics which are modulated by the vortices, or involve the detection of the small pressure waves that accompany the shedding of the vortices.

The cost of a vortex shedding meter is in the region of £1,000 to £1,500.

APPENDIX 2

THE PRESSURE SYSTEMS AND TRANSPORTABLE GAS CONTAINERS REGULATIONS

The Pressure Systems and Transportable Gas Containers Regulations will be mandatory for all compressed air systems from 1 July 1994.

The major requirements are as follows:

- The preparation of complete system diagrams showing each item of equipment such as compressor, receiver, dryer, isolating valve, pressure gauge and filter. In addition, all changes, additions, modifications etc. must be recorded on these diagrams.
- The preparation of a 'Written Scheme of Approval' to be certified by a 'competent person', which identifies how the system shall, in future, be examined.
- Regular examination of the system by a 'competent examiner' to ensure that it continues to comply with the Regulations.
- The written records should be updated to include the examination and any subsequent rectification work needed and carried out.
- The responsibility for defining the scope of examination lies with the user/owner. After initially identifying the protective devices, they should then consult a 'competent person' on what other parts of the system should be included.
- In general, all pressure vessels and protective devices should be included together with any pipework which is in such service and location that failure, with sudden release of stored energy, would be dangerous.
- Comprehensive operating instructions including start-up and shutdown procedures should be supplied to all operators.
- A manufacturer's literature list, including details of spare parts, service requirements and lubricants, should be prepared.
- A full no-load leakage test (see Section 3.3) should be carried out at regular intervals.
- Pressure vessels and in particular air receivers should be correctly marked.

These requirements are not significantly different to those previously imposed by the Health and Safety at Work Act 1974, the main difference being that documented proof of examination and the resulting compliance is required.

These regulations are welcome from an energy saving viewpoint, because they force companies to review distribution systems thereby directing them towards redundant pipework and zoning possibilities.

Advice on these Regulations is available from the British Compressed Air Society (BCAS).

APPENDIX 3

FURTHER INFORMATION

A3.1 Department of the Environment Publications and Videos

Good Practice Case Study 136:	Cost & Energy Savings Achieved by Improvements to a Compressed Air System
Good Practice Case Study 137:	Compressed Air Costs Reduced by Automatic Control System
Fuel Efficiency Booklet 4:	Compressed Air and Energy Use
Energy Consumption Guide 40:	Compressing Air Costs - Generation
Energy Consumption Guide 41:	Compressing Air Costs - Leakage
Energy Consumption Guide 42:	Compressing Air Costs - Treatment
Best Practice Video 6:	Compressing Air Costs

Copies of the above publications, further details on the video and information on energy efficiency in industry are available from:

Environment and Energy Helpline, Tel no: 0800 585794 or from
EEBPP
Harwell
Didcot
Oxfordshire
OX11 0QJ
Helpline E-mail: helpline@eebpp.org
Website: www.energy-efficiency.gov.uk

Information on energy efficiency in buildings is available from:

BRECSU
Building Research Establishment
Garston
Watford
WD2 7JR
Tel No: 01923 664258
Fax No: 01923 664787

A3.2 The British Compressed Air Society (BCAS)

Advice and copies of 'Guide to the Selection and Installation of Compressed Air Services' can be obtained from:

BCAS
33 - 34 Devonshire Street
London
W1N 1RF

Tel No: 0171 935 2464
Fax No: 0171 935 3077

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General Information: describes concepts and approaches yet to be fully established as good practice.

Fuel Efficiency Booklets: give detailed information on specific technologies and techniques.

Energy Efficiency in Buildings: helps new energy managers understand the use and costs of heating, lighting etc.

FIRST PUBLISHED 1994
REPRINTED 6/94, 4/95, 6/96, 3/98, 12/98, 6/99, 12/00, 9/01