VOLUME FOUR

Compressed Air System -Energy Auditing Guidebook

Introduction to compressed air systems auditing

Previously in the introduction to energy auditing volume, we discussed types of energy systems where compressed air systems were categorised as part of power transmission system. Compressed air system is a pneumatic system and is used to transmit electrical or mechanical energy using compressed air to a point of application. The system is made up of a compressor, receiver tank, cooler, dries, distribution network and points of use as illustrated in figure 1.

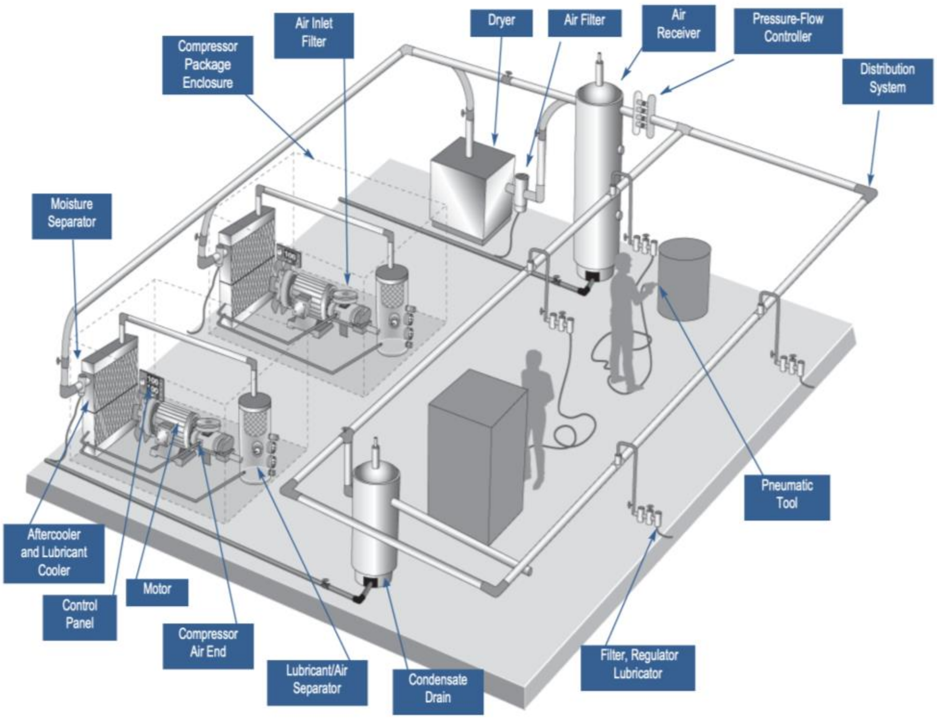

Figure 1: Parts of compressed air systems (Mckane, A. T. (2003)

It is not possible to conduct a comprehensive system audit without understanding the specific applications of the compressed air in a facility. Compressed air has applications ranging from construction, manufacturing, mining, and recreation services. Even though the operating principles are the same, the use characteristics may differ from system to system and facility to facility. The auditor should therefore purpose to independently understand each application in every audit they conduct. Some of the applications are:

Construction and Mining	Used for drilling holes in concrete; steel and other materials; tightening and loosening bolts; smoothing surfaces or cutting through materials; transportation of construction materials; moving heavy materials in a controlled manner; spray

	painting; powering vibrators; hydraulic system support; air blow guns and power washing; pneumatic safety valves.
Manufacturing	Driving of power tools like drilling; grinding; sanders; impact wrenches; riveters; nail guns; pneumatic controls for packing; filling; labelling; sealing and capping; painting and coating; cooling; drying; pneumatic pumps, electric switches and robotic systems.
Recreation services	Inflating equipment like sport balls, tires for recreational vehicles; scuba diving and water sports for breathing air supply and water parks; adventure and outdoor activities for paintball and airsoft, zipline braking systems and pneumatic launchers.

In case you encounter a compressed air system in a facility, do not panic. There are many ways to understand these applications. First, one can consult the machine operators and technicians in the facility. Second, consult the relevant literature on the specific machines and systems.

These systems significantly contribute to energy consumption in facilities. In the manufacturing industry compressed air systems consume about 10% to 15% of the facility's total energy bill. Generally, to produce 1 kWh of compressed air energy, it requires a minimum of 8 kWh of electricity**.** It is therefore imperative that energy efficiency be considered an important parameter during design, installation, commissioning and operating of compressed air systems.

Energy audit of compressed air systems has three fundamental steps as illustrated in figure 1:

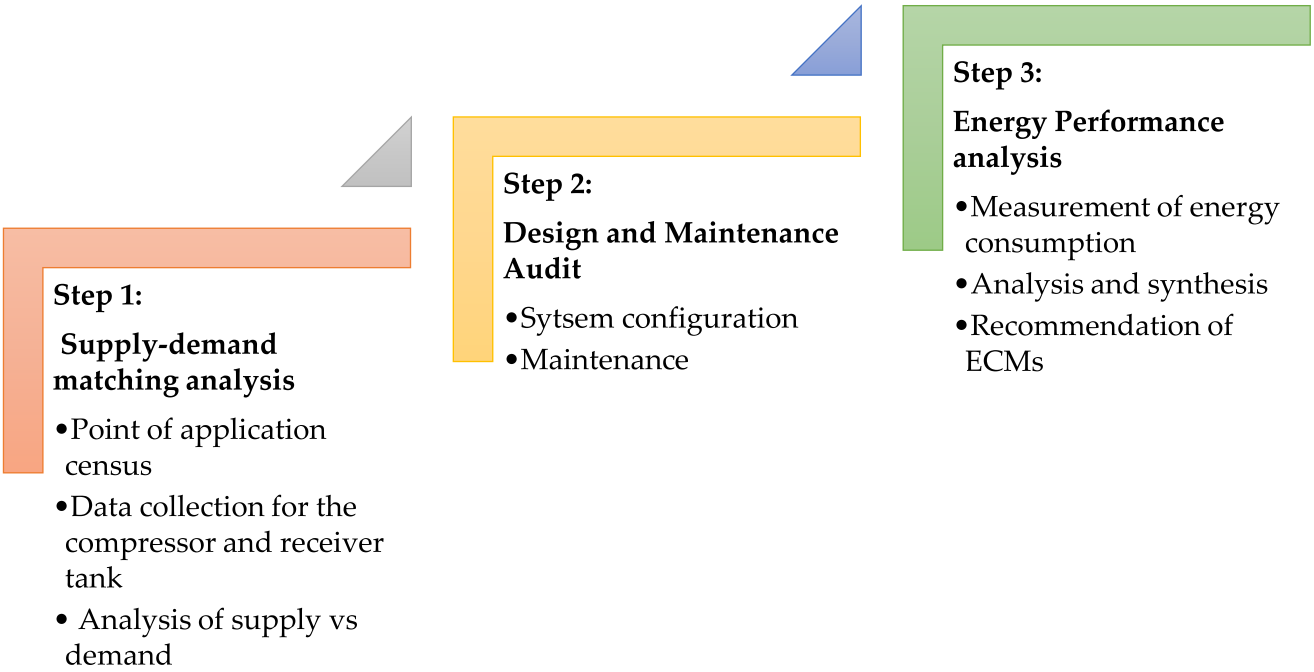

Figure 1: Steps of Auditing a Compressed Air System

The processes illustrated in figure 1 are sequential, with the results from the preceding step informing the analysis of the succeeding step.

Supply-demand matching analysis

As in any other energy system, the facility should design a supply utility that meets the demand. Instances of under sizing lead to poor quality of production. For example, the point of application might have less than the required process, pressure and flow rate. Under sizing can also lead to poor efficiency due to many starts and stops. Similarly, oversizing leads to inefficient operations due to poor loading. The first point of focus during audits should therefore be comparing the load requirements to the installed system size.

To understand the load requirement a census should be conducted for all points of compressed air applications. The load requirements are defined using the designed pressure and flow rates. This information can be obtained from the machine manuals and name plates. Where the manuals cannot be found in the facility, the auditor can search for the same online or contact the manufacturers. Take a bottling process plant for example, as illustrated in figure 2.

Figure 2: Dossing and capping processes

In each of the dossing and capping processes, there are air pressure and volumetric flow requirements. During conducting of census, all these requirements should be noted and summed up. These can be obtained from the name plate of the header (central point of distribution on the shop floor). Figure 3 shows an example of a name plate for a pneumatic cylinder operating the dossing and capping machine.

Bore size (mm)	25	32	40	50	63
Action	Double acting, Double rod				
Fluid	Air				
Proof pressure	1.05 MPa				
Maximum operating pressure	0.7 MPa				
Minimum operating pressure	0.05 MPa				
Ambient and fluid temperature	–10 to 60°C				
Lubrication	Not required (Non-lube)				
Piston speed	50 to 500 mm/s				
Stroke length tolerance	$^{+1.4}_{0}$				
Cushion	Rubber bumper				
Mounting	Foot, Rod flange				
Allowable rotational torque	0.25 N·m		0.55 N·m	1.25 N·m	2.0 N·m
Rod non-rotating accuracy	±1°	±0.8°	±0.5°		

Figure 3: Pneumatic cylinder name plate

Sometimes the numbers on the name plate could be blurry or even deleted due to ageing. Such numbers can be obtained from manuals or from the operators.

Compressed air is supplied by the compressor part of the system and stored in the receiver tank. The receiver tank takes care of demand intermittency to prevent instances of the frequent starting and stopping of the compressor. This can be equated to a storage tank in a water pumping system and battery energy storage in a solar photovoltaic system. An energy auditor should familiarize themselves with the operating principles and the specifications of the compressor and receiver tank.

The compressor and receiver tank specifications are important for supply demand matching analysis. These specifications include:

Maximum pressure	This is the highest pressure the compressor can achieve. The units can either be in MPa, bars or psi.
Minimum pressure	This is the lowest permissible operation pressure of a compressor. The units can either be in MPa, bars or psi.
Rated free flow	The volumetric flow rate that can be achieved by the compressor when unconnected to any load
Mechanical power input of the motor	This is the shaft power output of the motor. The units can be in kW or hp.
Motor efficiency	This is the conversion ratio of input electrical power to output mechanical.
Speed of the motor	The maximum speed that can be achieved by the motor.
Receiver tank volume	The maximum volumetric capacity of the receiver tank in cubic meters.

The specifications of the compressor can be obtained from the nameplate as illustrated in figure 4:

Figure 4: Compressor nameplate

After recording the specification, the loading factor can be determined using equation 1

$$\%\,Loading = \frac{\sum_i^n p_i\ q_i}{PQ} * 100 \qquad (1)$$

Where

p	Pressure requirement of each machine
q	Volumetric flow rate requirement of each machine
P	Maximum pressure of compressor
Q	Free flow rate

The preferred minimum loading factor is 80% for optimum overall energy efficiency.

Design and Maintenance Audit

The design, operation and maintenance of the compressed air system determines its energy performance. The energy auditor should assess the operating conditions of the compressor and the receiver tank, and the efficiency of delivery of the compressed air from the receiver tank to the point of use. The delivery efficiency is informed by the configuration of the distribution network and the associated air leaks. Evaluation of all these parameters is necessary for a quality compressed air system audit report.

The initial conditions of the compressed air determine the energy performance of a compressed air system. To achieve optimum compression efficiency, the air temperature in the compressor room should be kept as low as possible. The room temperature should be as near as possible to the ambient atmospheric conditions. Take for example two identical compressors, A and B, working in conditions with high and low ambient temperatures respectively. For both compressors to achieve the same pressure output, compressor A will be required to do more work. This is because hot air is less dense meaning it has less mass per unit volume. This relationship is illustrated using equation 2

$$W = nRT\ln\left(\frac{P_2}{P_1}\right) \tag{2}$$

Where

W	compression power at isothermal temperature
n	number of moles of air
R	ideal gas constant for air
T	temperature (in Kelvin)
P_1	inlet air pressure (mainly atmospheric pressure)
P_2	air pressure after compressor (read at the gauge)
ln	natural logarithm (i.e. $\log_e$)

From equation one, it is clear that an increase in temperature *T* will lead to an increase in compression power W. The auditor should therefore establish conditions that might lead to increase in air temperature in the compressor room. Some of the conditions would include poor ventilation, locating the compressor room close to source of heat like boiler, furnace or generator exhaust duct.

Apart from temperature, air composition can also affect performance of a compressed air system. The auditor should check the dust levels in the compressor room. Dust embedded in air reduce the amount of air entering the compression chamber. This could be as a result of clogging of the intake air filters. This forces the compressor to do more work to achieve pressure requirements.

The performance of the compressed air system partly depends on the delivery path ways between the receiver and the point of application. The delivery pipes should therefore be optimally designed. The optimization variables include:

i. Pipe diameters
ii. Pipe length
iii. Number of bends
iv. Number of valves

The friction losses in a pipe are inversely proportional to the diameter of the pipe and directly proportional to the length. This relationship is illustrated in the Darcy-Weisbach equation.

$$H_f = f \frac{l}{d} \frac{v^2}{2g} \tag{3}$$

Where

H_f	Friction loss along a pipe in meters of air
f	Darcy friction factor. It depends with the type of the pipe material
l	Length of the pipe in meters
d	Diameter of the pipe in meters
v	Velocity of air inside the pipe in meters per second
g	Acceleration due to earth's gravity

Equation 3 can assist the auditor to determine if the pressure loss is within acceptable limits. The rule of the thumb is that the pressure drop along the compressed air distribution line should not exceed 0.15 bars, notwithstanding the size of the system. Further, the air speed in the compressor should not exceed 5m/s, while that of the main header and the distribution blocks should not exceed 10m/s and 15m/s respectively (Mckane, A. T. (2003). With these limitations, the auditor should manipulate the other variables in equation 3 (the length, the diameter and pipe characteristic) to reduce the head loss.

Due to the inherent inefficiency of compressed air systems, care should be taken not to lose any of the compressed air via pipe leaks. The auditor should be acquainted with methods of detecting leaks within the system. Two methods can be used to accomplish this:

i. Human senses (audio and visual)
ii. Ultrasonic leak detector

Audio technique involves listening to the hissing sound on the shop floor. This can be done when the machines are running. The most accurate method however, would be when the machines have been switched off and the compressed air system is switched on. An auditor can visually inspect leaks in pipe joints either with or without use of soap. In cases where soap is not used, the auditor relies on appearance of abnormal fluids at the joints. This fluid is caused by condensation of water in the air around the leaking point. For accurate detection of leaks, the auditor can use ultrasonic leak detector which can show the intensity the leaks. Once the leak points have been detected, the auditor can recommend for maintenance of the pipes.

It is possible to estimate total air leaks in a facility without necessarily investigating the specific leak points. This can be achieved using the receiver method. The following steps should be used in this method:

i. Record the receiver tank volume, V
ii. Switch off the compressed air dependent operations in the facility
iii. Shut off supply from the receiver tank to the facility
iv. Drain liquid from the receiver tank
v. Record the atmospheric pressure of the area, P_0
vi. Record the receiver tank initial pressure, P_1
vii. Switch on the compressor to fill the receiver tank to operation pressure, P_2
viii. Switch off the compressor and isolate it from the receiver tank
ix. Open the supply valve to the facility
x. Record the time t_d taken to reduce the pressure from P_2 to P_1

Equation 4 can be used to calculate the rate of compressed air leakages in the system.

$$Q = \frac{(P_2 - P_1) x\, V}{P_o x\, t_d} \qquad (4)$$

Where:

Q	Rate of compressed air leakage (m^3/minute)
P_1	Receiver tank initial pressure
P_2	Receiver tank to operation pressure
V	Receiver tank volume, m^3
P_o	Atmospheric pressure
t_d	Time (minutes) taken for pressure to fall from P_2 to P_i.

Points to note

i. In case the auditor notices no significant change in the pressure after about 1 hour, this could indicate that the system has no leaks.

ii. The value Q can be multiplied with the compressor operating pressure to determine the fluid power loss due to the leaks.

To convince the facility on arresting leaks, the energy lost through the leaks should be quantified.

Energy Performance analysis

The results of supply and demand, design and maintenance analyses inform the assessment of energy performance of the compressed air systems. In most observed cases, energy auditors start by logging energy supply to the compressor and using the result as the sole source for recommendation of ECMs. This approach obviates the importance of thermal fluid characteristics of a compressed air system. In this section, we discuss how to link the findings in the previous section to energy performance of the system.

As previously observed, the supply load should not exceed the demand load by more than 20%. Where supply exceeds this value, there are instances of compressor underloading and this affects the efficiency. The relationship between compressor loading and efficiency is illustrated in figure 4.

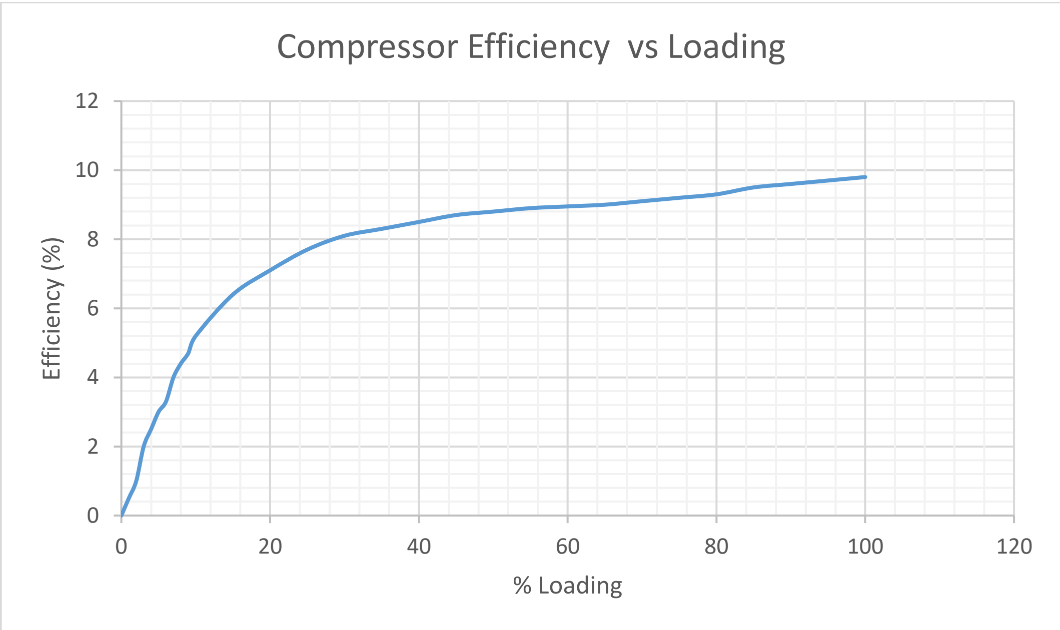

Figure 4: Compressor loading vs efficiency

The auditor can use figure 4 and the results of supply and demand analysis to quantify the overall system efficiency. Let us look at hypothetical data for a compressed air system in a facility with 5 points of application. The data is presented in table 1.

Table 1: Compressed air system data

Compressor details		**Machine details**		
			Pressure (Bars)	Volume (l/s)
Maximum pressure	13 bars	Machine 1	6	25
Minimum pressure	7 bars	Machine 2	5.5	15
Rated free flow	226l/sec	Machine 3	6	40
Mechanical power input of the motor	75kW	Machine 4	4.5	40
Motor efficiency	93%	Machine 5	6	30
Speed of the motor	2785rpms			
Receiver tank volume	6000 L	**Total**		150

Substituting the details in table 2 in equation 1, the percentage loading is determined to be 28.3%. Read off from figure 4, this loading indicates an overall efficiency of 8%. This compressor is oversized and the energy auditor should recommend resizing of the compressor which can achieve 80% loading. At this loading the overall efficiency would improve to 9.3%. The computed differences in overall efficiency can be used to determine the actual energy to be saved should the recommendation be implemented.

Instances of compressor underloading are real and have been identified when conducting energy audits in several facilities. In instances where it is practically impossible to conduct supply demand analysis, basic data logging can be an indicator of loading performance. Examine figure 5 extracted from an energy audit report from a packing materials processing facility in Kenya.

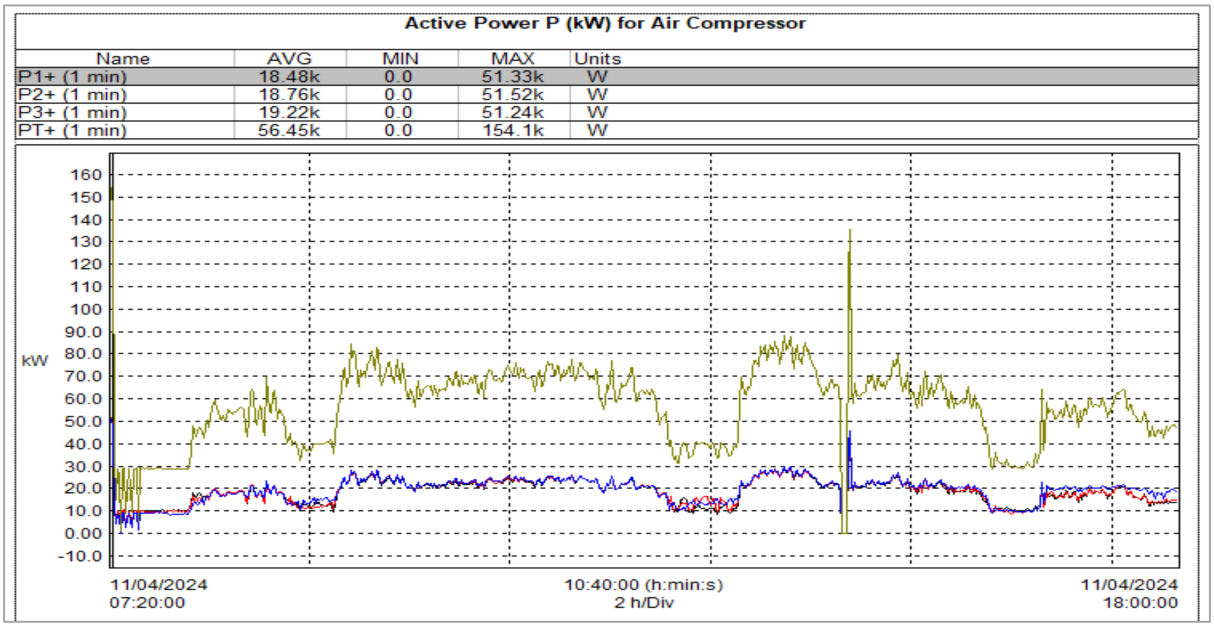

Figure 5: Compressor Logging

The logs presented in figure 5 were obtained from a compressor in the facility. The maximum power observed during current inrush is around 154 kW. In the absence of a name plate, we can conservatively assume this to be the rating of the compressor. The consumption trend shows the total average power consumption to be 56.4 kW. From this, we can compute the loading factor as a ratio of the average power to the maximum power. The loading factor is 36.7%. This is below the recommended loading factor of 80%. For the case of this facility, some lines had been decommissioned from the shop floor leading to a reduction in compressed air requirements. This was not accompanied by corresponding resizing of the compressor.

Point to note:
Compressor loading factor can also be estimated using the motor loading factor. This can be achieved by logging the actual electric power drawn from the system and comparing it with the power rating of the motor.

Should the auditors find that the temperature in the compressor room is more than the ambient, they need to recommend corrective actions. These can be achieved by improving the ventilation and positioning of the compressor room as far as possible from any heat source, as applicable. This recommendation should be followed by computation of energy savings. Example 1 illustrates such a computation.

Example 1

Assuming the auditor found out that the compressor room temperature was 29°C and the ambient temperature was 24°C with the following compressor operating parameters:

Operating pressure of 7 bars

Motor rating 75 kW

Rated motor efficiency 93%

Running 300 days per year

Average operating time of 18 hours per day

The auditor can use equation 1 to estimate the potential savings should the compressor room temperature be lowered to the ambient level.

At 24°C (297K)

$$W_{24^oC} = nR \times 297K \times \mathbf{ln}\left(\frac{8}{0.8154}\right) = 678.2\ nR$$

At 29°C (302K)

$$W_{29^oC} = nR \times 302 \times \mathbf{ln}\left(\frac{8}{0.8154}\right) = 689.2nR$$

$$\text{Percentage energy savings} = \left(\frac{689.6 - 678.2}{678.2}\right) \times 100 = 1.68\%$$

$$\text{Annual energy savings} = \left(\frac{75\ kW}{0.93}\right) \times 300\ \text{days} \times 18\ \text{hours} = 7317.2\ \text{kWh / yr}$$

Points to note:

i. The operating hours of a compressed air system depend on the size of the receiver tank and the demand requirements. The hours can be determined by logging the compressor over the operating cycle.

ii. The temperatures should be obtained by logging using a thermometer.

Upon detecting the leaks using methods and techniques mentioned in the design and maintenance section of this guide book, the auditor should quantify the energy losses. This can be achieved by approximating the diameter of the leak hole and using it to determine the compressed air and associated annual energy losses. There are tables published by Compressed Air and Gas Institute (CAGI) that can assist in estimating the volumetric rate of air loss. Table 2 is an extract of such tables from CAGI.

Table 2: Leakage Orifice Chart: Loss of air flow in m^3/min

Gauge pressure before orifice, bar	Diameter of Orifice, mm (note: calculated flow rate assumes orifice coefficient of 0.61)											
	1	2	3	4	5	6	7	8	9	10	15	20
4	0.03	0.11	0.25	0.45	0.70	1.01	1.38	1.80	2.28	2.82	6.34	11.28
4.5	0.03	0.12	0.28	0.50	0.78	1.12	1.52	1.98	2.51	3.10	6.98	12.40
5	0.03	0.14	0.30	0.54	0.85	1.22	1.66	2.16	2.74	3.38	7.61	13.53
5.5	0.04	0.15	0.33	0.59	0.92	1.32	1.79	2.34	2.97	3.66	8.24	14.65
6	0.04	0.16	0.35	0.63	0.99	1.42	1.93	2.52	3.19	3.94	8.87	15.78
6.5	0.04	0.17	0.38	0.68	1.06	1.52	2.07	2.70	3.42	4.23	9.51	16.90
7	0.05	0.18	0.41	0.72	1.13	1.62	2.21	2.88	3.65	4.51	10.14	18.03
7.5	0.05	0.19	0.43	0.77	1.20	1.72	2.35	3.06	3.88	4.79	10.77	19.15
8	0.05	0.20	0.46	0.81	1.27	1.82	2.48	3.24	4.11	5.07	11.40	20.27
8.5	0.05	0.21	0.48	0.86	1.34	1.93	2.62	3.42	4.33	5.35	12.04	21.40
9	0.06	0.23	0.51	0.90	1.41	2.03	2.76	3.60	4.56	5.63	12.67	22.52
9.5	0.06	0.24	0.53	0.95	1.48	2.13	2.90	3.78	4.79	5.91	13.30	23.65
10	0.06	0.25	0.56	0.99	1.55	2.23	3.03	3.96	5.02	6.19	13.94	24.77

Assuming the auditor detects a leak and records the following measurements:

i. Gauge pressure before the leak, 7 bars

ii. Diameter of the orifice, 3mm

The associated air loss can be read off from table 2 as 0.41 m^3/min. This value can be used to compute the fluid power loss at the orifice. However, there is a direct method of using the orifice diameter to compute annual energy losses. Table 3 can guide the auditor in estimating the losses.

Table 3: Annual energy losses associated with air leaks

	Losses From Compressed Air Leaks (kWh/year) via SI and Imperial Units @ 8760 hours/year					
Approximate Hole Diameter mm (in)	*Air Pressure in bar (psig)*					
	4 (58)	*5 (72)*	*6 (87)*	*7 (102)*	*8 (116)*	*9 (131)*
1 (5/128)	900	1,300	1,600	2,000	2,400	2,900
2 (5/64)	3,700	5,100	6,500	8,100	9,700	11,500
3 (1/8)	8,400	11,400	14,600	18,200	21,900	25,900
6 (1/4)	33,700	45,500	58,600	72,700	87,700	103,500
9 (3/8)	75,800	102,400	131,800	163,600	197,400	232,800

From table 3, the associated energy loss will be 18200 kWh/yr.

Points to note:

i. The leak orifice diameter can be estimated using a traveling microscope or using a soft paper to estimate the diameter of the jet stream.

ii. In lieu of using tables, the losses can be determined using online software or first principles.

All the previous examples have focused on system audit. However, instances of inefficiency in the system are caused by performance problems of individual system components. One of the methods of detecting inefficiency in the compressor is to conduct a free air delivery test on the compressor using the following method:

i. Record the receiver tank volume, V
ii. Record the compressor's design free air delivery
iii. Switch off the compressed air dependent operations in the facility
iv. Shut off supply from the receiver tank to the facility
v. Drain liquid from the receiver tank
vi. Record the atmospheric pressure of the area, P_0
vii. Record the receiver tank initial pressure, P_1
viii. Switch on the compressor to fill the receiver tank to operation pressure, P_2
ix. Record the time t_d taken to fill the compressor to the operational pressure.

Using equation 4, determine the free air delivery rate of the compressor and compare it with the name plate parameter. The best industry practice recommends that the measured free air delivery should be at least 85% of the rated one. This guideline can be found in CAGI.