Industrial Compressed Air Usage – Two Case Studies

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ABSTRACT

Analyses of two industrial compressed air systems which are already installed and operating in manufacturing plants have been surveyed in the context of energy usage. The installations are quite different in compressed air needs: one is focused on actuation; while the other uses compressed air primarily for material handling. In both sites, the energy of the compressed air is evaluated at each key element of the system and the typical end use application profile is assessed. In this way, the energy balance of the system has been analyzed quantitatively, with the effect of distribution leaks accounted for directly. Combining the measured profile of the plants as currently used, scenarios for reduced compressed air usage are identified. Finally, similarities and contrasts between the two sites are drawn, and show that the basic challenges facing the process designer and plant manager in attempting to reduce the compressed air costs, and hence the cost of production, are similar across the manufacturing sector as a whole.

NOMENCLATURE

A	area	m^2
Ex	exergy	J
F	force	N
m	mass	kg
p	pressure	Pa
r	ideal gas constant	J/kg/K

T	temperature	K
V	volume	m^3, Nm^3
W	work	J
c_d	discharge coefficient	[-]
η	efficiency	[-]
κ	specific heat ratio	[-]

1 INTRODUCTION

In recent times, compressed air systems (CAS) are being rightfully criticized for poor energy efficiency. This has become an increasing important issue as the cost of energy has increased in all applications. The problem lies in the design of the CAS as plant design has implicitly assumed that compressed air is free at point of use and cheap to produce centrally, and so energy saving aspects are rarely considered at the process planning stage. The inadvisability of CAS in terms of energy usage has been well known for some time, but the demand for near term product yield has acted as a major obstacle to reduction of CA usage. As a result, compressed air users are looking for any way to achieve cost reduction. Typically energy savings measures are sought mainly on the generation and treatment side even though compressors and dryers have been considerably optimized during last decades (including more efficient compressor drive alternatives). In fact, it is actually the compressed air application that is defining the energy efficiency of the CAS, which also means that any further effort related to the energy savings must be focused on the application (end-use) side. While there are several best practice guides available (from, for example, the Carbon Trust in the UK or the DoE in the USA), the scale of the problem and the potential savings are difficult to quantify, even in terms of order of magnitude, for a given site. This is exacerbated by the fact that there appears to almost no baseline data in the open literature on industrial compressed air use. This paper presents a detailed assessment of the end-use profile of two representative companies operating large facilities in Ireland. In order to complete an overall picture of CA usage at each site, a model of the application side, the so called “end-use catalogue” has been compiled, including where necessary models of the various components of the system. Although results are limited in part by data availability/lack of detailed inputs for model, the study reveals inefficient usage, leak problems and also poor system control and monitoring, which is clearly not consistent with established best practice. While in an ideal world all these issues would be dealt with aggressively, the economics of the required investment is often not clear. However, the end-use catalogue can be used to target investment and activity to maximize the cost effectiveness of energy saving measures.

2 CAS ANALYSIS – EXERGY APPROACH

2.1 Exergy definition

In order to emphasize the inefficiency of the typical end-use a simplified system analysis based on work ability (exergy) of the compressed air is shown. The exergy represents the amount of useful energy, which can be theoretically converted to mechanical energy. It can be easily shown that in case of CAS only the pressure dependent term is crucial/Cai02/. Equation 1 shows the mathematical formulation of exergy referenced to the ambient conditions (subscript a) based on ideal gas approach. This is suitable for further investigation of the work ability change of CA over each of the key elements of the CAS characterized by relevant efficiency.

$$Ex = mrT_a ln\frac{p}{p_a} = p_a V_a ln\frac{p}{p_a} \tag{1}$$

2.2 Exergy analysis of the key components

2.2.1 Compressor

For sake of simplicity, consider that the process of transformation is related to the outlet parameters from compressor as an enclosed unit, thus any internal processes of the compressor, including control, are not detailed. The efficiency of conversion of the input electric power to the output compressed air power can be described by Equation 2.

$$\eta_{com} = \frac{Ex_{out}}{E_{el}} = \frac{\dot{E}x_{out}}{\dot{E}_{el}} = \frac{p_a \dot{V}_a ln\frac{p_{out}}{p_a}}{\dot{E}_{el}} \tag{2}$$

The long term averaged values of the existing oil free centrifugal compressor installed in Company A: p_{out} = 6.75 bar(a), V_a = 0.45 Nm^3/s, E_{el} = 220 kW. Thus, the efficiency of the process is 0.39. The rest of energy is lost in form of heat and losses (including bypass).

2.2.2 Dryer

The efficiency of the dryer is expressed by Equation 3. The outlet exergy of CA can be modified by the dryer pressure drop only or a combination of purge air loss and the pressure drop depending on the dryer type.

$$\eta_{dryer} = \frac{Ex_{out}}{Ex_{in}} = \frac{\dot{E}x_{out}}{\dot{E}x_{in}} = \frac{\dot{m}_{out} ln\frac{p_{out}}{p_a}}{\dot{m}_{in} ln\frac{p_{in}}{p_a}} \tag{3}$$

A typical long term purge air loss in Company A is approximately 5% of the inlet mass rate and a pressure drop of 2% of the inlet pressure. Under these conditions the value of dryer efficiency is 0.94. The installed dryer is a heated desiccant type, so inclusion of power consumption of the heating elements into dryer efficiency calculation is possible, but unlike the compressor, electrical energy does not directly change the exergy.

2.2.3 Pressure regulator

The next important element of the CAS is the pressure regulator stabilizing the pressure level before final application by reducing pressure of the incoming compressed air. The efficiency of the process can be also described by Equation 3 depending on the pressure regulator type. A non-ventilating pressure regulator adjusted to the pressure level of 6 bar(a) for pneumatic actuators, with data taken at Company A, yields an efficiency of 0.95. The pressure drop of the distribution system is not explicitly accounted for, hence efficiency of the pressure regulator is a lower bound. Please note that possible low efficiency of the regulation does not mean a poor quality regulator, but work ability destruction according to the need of the relevant application.

2.2.4 Application side

This is the most interesting part of the work ability analysis. Two different CA applications, pneumatic actuation and pure cooling/drying, are discussed from the point of the view of the usage of the useful energy. Pneumatic actuation, which is supposed to be a relatively efficient CA application, will be detailed for the example of the double acting rodless cylinder. Moreover, using work ability approach it is possible to understand compressed air power as similar to electrical power, which leads to the direct comparison of efficiencies between an electrical and pneumatic actuator. This has been previously done by Cai et al./Cai02/, who have reported that an electric actuator is always more efficient in wide range of situations especially under heavy duty cycle (non-stop operation). They also pointed out that pneumatic actuators with low duty cycle are more efficient than electric alternative, but these are rarely used in the industrial applications.

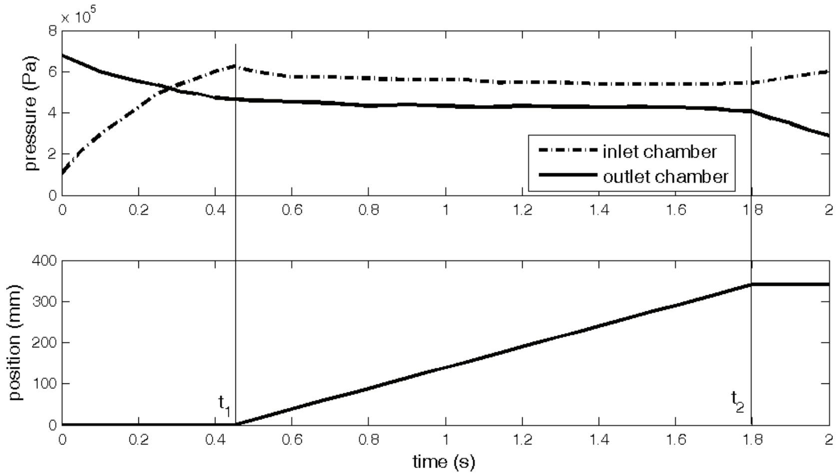

Figure 1: Rodless cylinder behaviour

The **Figure 1** shows dynamic behavior of the double acting rodless cylinder, trends have been slightly smoothed for sake of simplicity. The experimental data, test rig description and further system parameters can be found in Ning and Bone/Nin05/. The data has been used for estimation of the efficiency during the motion of the actuator. The efficiency of the process can be defined by Equation 4, where the numerator represents useful work W done by pneumatic forces F and denominator means total energy of incoming compressed air to the working chamber, which can be utilized for the useful work from the start of the motion at time t_1 till the end of motion at time t_2.

$$\eta_{act} = \frac{W}{\Delta Ex_{in}} = \frac{\int_0^L F\, ds}{\int_{t_1}^{t_2} \dot{E}x_{in}\, dt} \qquad (4)$$

Useful work W is performed by pneumatic forces F defined by Equation 5. The forces are acting on area A in the direction of the piston motion from start position 0 $(t=t_1)$ to final position L $(t=t_2)$.

$$F = A(p_{in} - p_{out}) \tag{5}$$

The efficiency this case is ~0.14. This is a maximum value as when total energy available for work is considered from time 0s until final time available of 2s, or when backstroke motion is included, will lead to lower efficiency. Although the efficiency of the actuation can vary depending on the situation under investigation, any results will make the total efficiency of the CAS (product of particular efficiencies) tend towards zero.

Cai et al./Cai02/ have shown that compressed air power (exergy rate) has two parts: (i) the transmission power used for pushing compressed air towards to the application side and (ii) the expansion power, which can be used when compressed air is expanding after closing inlet valve. Although the latter represents the larger part of all compressed air power above 5.2 bar, often only a small fraction of the expansion power is utilized. Furthermore, once the piston reaches the end of stroke position, the compressed air inlet remains open leading to accumulation of high pressure energy in the working chamber as shown in the Figure 1. This energy is wasted during retract motion of the cylinder as compressed air is ventilated to the atmosphere. Moreover, the same energy reduces the pneumatic force used for useful work as there is still a high level of decaying pressure in the non-working chamber, which is also clearly visible in Figure 1. Several approaches of energy efficiency improvement of pneumatic actuators can be found for example in Shen and Goldfarb/She07/.

In terms of energy efficiency, use of CA for pure cooling/drying is usually completely inappropriate. Since there is no useful work done, the efficiency is. The high pressure level energy is usually throttled down and work ability destroyed. For the sake of simplicity, the effect of cooling/drying can be related to the mass flow rate of the compressed air, which must run through compressor to the end use application according to conservation of mass. As a considerable part of electric energy is transferred into pressure energy of the medium and this high pressure energy is not utilized at the application side, compressed air usage leads to the most of the energy being wasted.

3 DETAILED INVESTIGATION OF THE SYSTEM

The application of the exergy analysis has shown that the critical point of the compressed air system is the application side, which is a limiting factor of the efficiency of the whole compressed air system. From the point of view of utilization of useful energy, the overall efficiency of the CAS generation is likely to be order of magnitude one. Although efficiency of particular CAS element may vary according to the specific operating conditions, the end-use side will be the most inefficient part of CAS. Thus, any future optimization of the compressed air system should be focused on the application side, where a high potential of energy savings is projected. This requires detailed knowledge of the application side.

In order to quantify air consumption of the CAS, different types of air consumers are modeled and the overall air consumption is compared to the real data available. The process includes collection of available information such as pressure level of the particular end application, geometric parameters, working time/cycling, shift patterns etc. which is necessary for a mathematical model. It is obvious that this kind of data is not always easy to obtain resulting in an imprecise model, but a better understanding of the process is still possible as shown in the results below.

In terms of the dominant physics, there are three main types of compressed air consumer found overall in the Companies A and B which have been modeled using a simplified ideal gas approach. The model in Equation 6a is used mainly for estimation of compressed air consumption of pneumatic actuators and valves, where volume V is a function of internal cylinder geometry and stroke (inlet tubing volume downstream of proportional valve included). Volumetric flow rate is defined by the number of cycles per time. Equation 6b represents all open blow applications (drying, cooling, cleaning, conveying etc.), where the pressure ratio is below critical pressure ratio (p_{crit} = 0.528). It is assumed that critical conditions occur at the nozzle outlet. Equation 6c is used for situations, where CA pressure is only slightly higher than ambient pressure. The temperature of the compressed air in the models is assumed to be 20°C. The discharge coefficient c_d is generally set to a value of 0.65 as a sharp rather than smooth edge is more appropriate. All results are recalculated to volumetric flow rate at the reference conditions of T_{ref} = 293 K, p_{ref} = 1 bar for comparison with the measured data.

$$\dot{m} = p\dot{V}/rT \tag{6a}$$

$$\dot{m} = c_d A \sqrt{\kappa r T \frac{2}{\kappa+1} \frac{p}{rT}} \left(\frac{2}{\kappa+1}\right)^{\frac{1}{\kappa-1}} \tag{6b}$$

$$\dot{m} = c_d A\, p \sqrt{\frac{2}{rT} \frac{\kappa}{\kappa-1}} \sqrt{\left(\frac{p_a}{p}\right)^{2/\kappa} - \left(\frac{p_a}{p}\right)^{\kappa+1/\kappa}} \tag{6c}$$

3.1 Company A

There is a multi-purpose usage of compressed air inclusive of pneumatic actuation, cleaning, panels and controllers sealing, fluidizing, filter pulsing, open blowing and use of some air accessories. The model does not take into account any dynamic behavior of the system, so all values of CA consumption are day average and total result is compared to long-term averaged daily measured data. The breakdown of air usage is:

- fluidizing (conveying/cleaning – at two different pressure levels) - 67.12%;
- open blow applications (e.g. bowl feeding, index sensor blow-off) - 12.76%
- filter pulsing - 5.92%;
- actuation - 3.82%;
- controllers sealing - 3.5%;
- hoisting - 3.46%;
- air whip cleaning - 1.77%;
- PLC sealing 1.53%;
- air gun 0.12%.

The difference between the model and measured values was initially almost 30%. However, based on the leak estimation technique described below, the difference has been reduced to 6.7%, as summarized in Table 1.

The company also implements a pressure decay measurement, which can be related to the leakage effect. At end of a production period, all machines are stopped and the system is pressurized up to a pressure of 6.6 bar(a). The generation side is separated from the distribution system and end-use side by closing a check valve behind the receivers. The time of 400 s of pressure decay from p_1 = 6.6 bar(a) down to p_2 = 2 bar(a) was recorded. Assuming an exponential decay, the pressure function can be described by Equation 7, where it is assumed that the final steady pressure is equalized to atmospheric (ambient) pressure i.e. p_a = 1 bar.

$$p(t) = Pe^{kt} + p_a \tag{7}$$

The unknown decay parameter k is solved using a matrix approach given by Equation 8,

$$\boldsymbol{X} = \boldsymbol{A} \backslash \boldsymbol{B} \tag{8}$$

where $\boldsymbol{X} = \begin{bmatrix} \ln(P) \\ k \end{bmatrix}$, $\boldsymbol{A} = \begin{bmatrix} 1 & 0 \\ 1 & 400 \end{bmatrix}$, $\boldsymbol{B} = \begin{bmatrix} \ln(p_1 - p_a) \\ \ln(p_2 - p_a) \end{bmatrix}$, yielding a value of k = -0,0043.

Under the conditions mentioned above, compressed air is escaping from the system at sonic flow velocity; leaking mass is defined using Equation 6b with generalized area A and discharge coefficient c_d. Moreover, system temperature is considered to be constant during system pressure decay, i.e. 20 °C. Thus the rate of pressure change is

$$-\dot{p} = \frac{rT}{V} \dot{m} \tag{9}$$

Combining this with 6b produces the completed differential equation of the process:

$$-\dot{p} = \frac{rT}{V} c_d A \sqrt{\kappa r T \frac{2}{\kappa+1} \frac{p(t)}{rT}} \left(\frac{2}{\kappa+1}\right)^{\frac{1}{\kappa-1}} \tag{10}$$

The Equation 10 can be integrated from zero initial time to final time τ

$$-\int_0^\tau \dot{p}\, dt = \frac{rT}{V} \int_0^\tau c_d A \sqrt{\kappa r T \frac{2}{\kappa+1} \frac{p(t)}{rT}} \left(\frac{2}{\kappa+1}\right)^{\frac{1}{\kappa-1}} dt \tag{11}$$

The leak flow rate can be expressed in general form as

$$\dot{V}_{leak} = -c_d A \sqrt{\kappa r T \frac{2}{\kappa+1}} = -\frac{V}{\left(\frac{2}{\kappa+1}\right)^{\frac{1}{\kappa-1}}} \frac{\int_0^\tau \dot{p}\, dt}{\int_0^\tau p(t) dt} \tag{12}$$

The leak flow rate for the particular case (i.e. k=-0,0043) is described by Equation 13.

$$\dot{V}_{leak} = -\frac{V}{\left(\frac{2}{\kappa+1}\right)^{\frac{1}{\kappa-1}}} \frac{P\left[e^{kt}\right]_0^\tau}{\frac{P}{k}\left[e^{kt}\right]_0^\tau + p_a \tau} = 0{,}0049 \cdot V \tag{13}$$

The volume V of distribution system and end use pipelines (without PLC cabinets) has been estimated as 5.1 m^3 from the system schematic drawings. The estimate of leak flow rate is directly proportional to V, and so has a sensitivity of 1, thus, there is a potential for improvement of the technique by detailed volume estimation. It is reasonable to assume that leaks appear largely on the application side, where operating pressure level is mostly 6 bar(a), thus a leak flow rate would be 342.2 Nm^3/hour. The technique assumes that the start pressure p_1 is the same everywhere in the system, which is obviously not true (similarly with the final pressure p_2), but as it is assumed that all leaks occur in a choked flow, this should not make much difference. An additional source of error is the assumption that ventilation of the end use machines during pressure decay process is equal to the internal machine leaks. This will tend to underestimate the leak flow rate. The result is also affected by CA consumption for PLC panels sealing, which is actually very low as PLC enclosure pressure is only slightly higher than ambient pressure, nonetheless air consumption of PLC panels is deducted from total leak estimation in the final calculations.

Based on the model, almost 80% of the fluidizing air is used at pressure slightly higher than atmospheric and this air has been generated with considerably higher pressure level than actually required, thus a more detailed study of the process and possible local blower implementation could result in a substantial energy saving opportunity.

Model (Nm^3/day)	35266.15	Air consumption (Nm^3/day)	27478.23
		Leakage (Nm^3/day)	7787.92
Measured data (Nm^3/day)	37800.00		
Difference (Nm^3/day)	2533.85	(%)	6.70

Table 1: Air consumption in company A

Model (Nm^3/hour)	22.01	Actuation (%)	15.65
		Open blow (%)	84.35
Measured data (Nm^3/hour)	31.00		
Difference (Nm^3/hour)	8.99	(%)	29.00

Table 2: Air consumption in company B

3.2 Company B

The usage of CA is primarily to operate automated assembly units with various pneumatic components. The model of compressed air consumption of only one assembly unit has executed due to the availability of real data. Further details of the measurement and system can be found in harris et al./Har09/. The data has been related to the reference conditions of T_{ref} = 293K, p_{ref} = 1 bar for direct comparison with the model. The consumers of compressed air are actuators for moving the products and air knives for drying in the form of open blow. Table 2 summarizes the results for the situation of a heavy load cycle (maximum number of products per minute = 24). Small variability associated with the data has been observed. The model has revealed that a small part of the utilized compressed air is used for actuation; most of the modeled process air is used for drying. Harris et al./Har09/ have also reported for a number of observations that air consumption at no production output was as high as that required for full production. This would support the result of a very small proportion of the compressed air used for actuation as well as poor control of pneumatic elements. The difference between the modeled situation and the measured values (29%) can be contributed to both imperfections of the model caused by lack of detailed input data and mostly compressed air losses in the form of leaks in the distribution system. The leak estimation can be also justified using available data/Har09/ for air consumption at zero machine output, where a box-plot containing the middle half of the scores in the distribution ranges from 2.6 – 14.5 Nm^3/hour (median 3.1). This would mean that the difference between the model and measurement of 8.99 Nm^3/hour associated mainly with leak estimation is a realistic value. However, it can be concluded that in this particular case the usage of compressed air for producing any useful work (actuation) is minimal as the machine uses most of the compressed air for drying. The inefficiency of this open blow process has been previously explained. The costs associated with compressed air usage required for drying are enormous. High pressure energy is given to the required mass flow rate at the generation side (system pressure level 8 bar(a)) and this energy is wastefully lost before final application as pressure is reduced down to 2.75 bar(a). Implementation of local blower with relevant air treatment and slightly bigger nozzle diameter could give a similar drying effect if flow is choked, but the initial pressure level can be considerably lower resulting in operational costs savings. The advantage of local configurations of CAS has been demonstrated by Yuan et al./Yua06/.

4 CONCLUSION

The compressed air systems are complex and any simplified approach will naturally lead to the differences between reality and the model. However, it is has been shown that it is possible to understand system behavior based on a suitable mathematical tool. The investigation revealed poor compressed air energy utilization on the application side resulting in low overall system efficiency and several energy savings opportunities are suggested for specific applications found in the investigated companies. The improvement of end-user performance, including reduction of inappropriate use and leaks, is a key step towards optimization of compressed air systems and any future work should be focused on this side.

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