Exergy Flow Diagrams as Novel Approach to Discuss the Efficiency of Compressed Air Systems

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Abstract

Compressed air systems are among the major consumers of electrical energy in industry. As the importance of energy-efficiency grows in general, so does the need for valid and reliable metrics for discussing efficiency. Today, energy flow diagrams are a common tool to illustrate energy efficiency in compressed air systems. They are however subject to various shortcomings which are mainly related to their lack of transparency and reproducibility. Therefore, a novel approach for the assessment of efficiency is presented which is based on the exergy concept. This approach allows for a transparent calculation of flow diagrams for compressed air systems, including the possibility to illustrate the effects of a heat recovery system. The concept is illustrated at the example of an industrial set-up starting at the compressor inlet and ending at the application. The resulting diagram allows a more transparent and objective view on efficiency evaluations and thus contributes to a better understanding of energy-efficiency in compressed air systems and related applications.

KEYWORDS: energy efficiency, compressed air system, flow diagram, exergy concept

1. Introduction

The efficient usage of energy in production processes is of growing concern to modern industrial companies. Typical energy sources for drive applications are electricity, hydraulics and compressed air. The latter is said to be simple in use and its applications have comparatively low investments. Approximately 10% (80 TWh) of the

total European electric energy consumption in industry is spent for powering compressed air applications. 75% of the total cost of ownership in a typical compressed air system is spent for allocating and distributing the energy which offers a high cost saving factor /1/. Pneumatic drive applications in automation processes such as pneumatic positioning drives account for approximately 30% of the industrially processed compressed air.

In the discussion about energy-efficiency, so-called energy flow diagrams are often used to judge compressed air infrastructure and to quantify and illustrate losses within the infrastructure. Typical representatives given in /2/ and /3/ state an efficiency of about 6-9% for compressed air applications including generation, air treatment and distribution. Based on these figures, pneumatic applications are considered as having a low energy-efficiency, especially if assuming efficiencies of about 80% in electrical spindle or belt positioning systems. However, a direct comparison of electric and pneumatic drive systems is challenging, not distinct and often not adequate. It often lacks of a valid definition of system boundaries, dynamic behavior and application areas. Both technologies are complementary and each one offers advantages as well as disadvantages. In order to compare different systems and different technologies, energy flow diagrams are often used to visualize the energy losses and the efficiency at certain observation points in the system. Their usage as a metric suffers from various shortcomings such as a lack of transparency and reproducibility and thus the interpretation of results is difficult.

To overcome these shortcomings, the aim of this paper is to propose a novel approach to create flow diagrams based on the thermodynamic concept of exergy. The benefit of this approach is to improve the quality of judgments on energy-efficiency of compressed air as it makes flow diagrams more transparent to the reader. The paper is structured as follows: In the first part, energy flow diagrams are introduced and their shortcomings for quantifying the efficiency of compressed air systems are discussed (section 2). Based on the results of this discussion, a novel approach is proposed that allows reproducible calculations of losses using the thermodynamic concept of exergy. This includes an illustrative case study for the overall compressed air infrastructure with compressor unit, air treatment, storages, distribution, ending at the service unit of an application (section 3). Then an exergy-based analysis of a pneumatic application is provided (section 4). The paper ends with a conclusion of the presented results (section 5).

2. Exergy flow diagrams representing compressed air systems

2.1 Energy flow diagrams

Energy flow diagrams represent in general the energetic flows from an energy source across several observation points to an energy sink. The observation points are usually chosen to separate single processes in a system or parts of it. For example, a typical observation point in compressed air systems is chosen before and after the air treatment equipment. Depending on the scope and level of detail of the flow diagram, branched and unbranched flow diagrams are used. The unbranched display format is typically used for representing energetic flows in compressed air systems. The total amount of inflowing energy is normalized to 100%. The width of the arrows is proportional to the absolute value of the energy flow or losses.



Figure 1: Energy flow diagram, taken out of previous research /6/

By balancing incoming and outgoing energy flows between two observation points, the efficiency of the concerned part of the system is determined; if the first and last points are chosen, then the overall system efficiency can be calculated.

2.2 Shortcomings of energy flow diagrams for compressed air

Figure 1 shows a typical example of an energy flow diagrams similarly found in various publications. The diagram states a mechanic volume expansion work of app. 7%. For

this and similar energy diagrams for compressed air systems, it is difficult to determine how the "energy" flow has been determined. In a technically correct way, the calculations have to be based on the thermodynamic concept of energy. The amount of stored energy in a system is always related to a reference value, e.g. the zero level for the potential energy definition. For calculating the energy in a observation point, the thermodynamic energy towards the absolute zero-point has to be computed which is normally not done. Most energy flow diagrams neglect the energy of the intake air which is available for free in the environment. Losses are only calculated based on the electrical energy input. Energy flow diagrams which do not include the energy of intake air are incomplete and erroneous. At each observation point *a*, the temperature T_a , the standardized volume flow rate \dot{Q}_a , the density ρ_N and the specific heat ratio c_v need to be known. With this information at each point *a*, it is possible to calculate the inner energy U_a of closed systems with Eq. (1) /5/.

$$U_a = Q_a \cdot \rho_N \cdot c_v \cdot T_a \tag{1}$$

Eq. (1) shows that the energy content is mainly a function of the temperature. The current pressure level does not influence the energy content of the system. Since the pressure is the driving force in pneumatic systems to carry out work, the recurrence to the stored energy content is unsuitable for setting up a flow diagram for compressed air usage. Even with a complete balancing of inner energy U, the usability or the benefit of the available energy cannot be expressed. It depends in pneumatic drive systems mainly on the current pressure level towards ambient conditions. The inner energy U does not account for that. Regarding a "compressor" more closely further underlines the problem of energy considerations: an energetically ideal compression is isothermal as it requires the least amount of work for producing compressed air at ambient temperature. By definition, isothermal means that there is no temperature change during the compression. Thus, the same amount of energy required for the compression has to be removed from the process, e.g. in terms of a heat flow. The compressed air does not heat up during the compression process. Hence, the compressor "exchanges" electrical energy for heat. Based on these considerations, the process is energy neutral with an efficiency of 0%. By changing the system boundaries and including the produced heat in the energy balance, the efficiency is 100%. Both efficiency values are correct, but based on different definitions of the system boundary. This leads to the conclusion that the concept of thermodynamic energy is not sufficient for describing the users benefit at certain observation points. The values provided in Figure 1 are necessarily based on a different method of calculation since the

compressors efficiency is neither stated at 0% nor 100%, but which is not further specified. The shown diagram from /6/ defines losses as efficiency numbers which are approximately calculated based on measurements and related to the input energy. A consistent balancing of all energy parts in each observation point is thus not possible anymore, especially when considering heat recovery system for compressors as well. Hence the energy-flow approach suffers from a lack of transparency and clearness and diverts the thermodynamic concept of energy from its intended use.

3. Exergy flow diagram for the supply side

The discussion of existing energy flow diagrams illustrates that their coherent and reliable use is not possible. In this section, a novel approach using the concept of thermodynamic exergy is proposed. It overcomes main drawbacks of energy flow diagrams and leads to a straight-forward and reproducible approach for an exergetic analysis of compressed air systems. (Please note that exergy is defined in this paper as a physical power with the unit [kW].)

3.1 The thermodynamic concept of exergy adapted to flow diagrams

In thermodynamics, the exergy that is available at a certain observation point is the share of the energy in the system that can be transformed into work if the system is brought to equilibrium with ambient conditions. The remaining share is called anergy. Exergy is a function of state with no conservative properties, i.e. exergy can be transformed to anergy and is therefore lost. For the calculation of exergy in an open system according to /7/, four variables of the survey point ^{*a*} are required: the amount of input electric energy P_{e_a} (pure exergy), the absolute pressure p_a , temperature T_a and the related volume flow rate Q_a . With those variables, the exergy is computed to:

$$E_a = P_{e_a} + Q_a \rho_N c_p (T_a - T_{atm}) + Q_a \rho_N T_{atm} \left(R \cdot ln \left(\frac{p_a}{p_{atm}} \right) - c_p \cdot ln \left(\frac{T_a}{T_{atm}} \right) \right)$$
(2)

The concept of the exergy allows the consideration of four main events: consumption of electrical energy in a process, pressure changes (mainly pressure drops), temperature changes and changes in volume flow rate, caused by leakages. Those events happen separately or at the same time. By comparing the exergy content of two observation points, exergy losses can be derived. The percentage of losses can be calculated by relating the exergy loss between two observation points to the overall exergy input.



Figure 2: Compressed air infrastructure with generation, air treatment and distribution up to the service unit at the consumers.

3.2 Exergy flow diagrams without heat recovery

In this section, the exergy for an exemplary compressed air system is computed for an illustrative case study with several observation points as defined in **Figure 2** and illustrated in an unbranched flow diagram. **Table 1** shows eight observation points, each with information on the absolute pressure, temperature and volume flow rate. The leakage can be computed from the changes in the volume flow rate. In the example in Table 1, the difference between "Pipe network" and "Service unit" is the amount of leakage under the assumption that the main part of the leakage appears close to the consumer. Based on Eq. (2) the exergy, lost exergy and efficiency (expressed as the percentage of remaining exergy) are calculated. For the calculation of exergy in point 1 (starting point), the overall input of electric energy into the whole system (here for the compressor, after cooler and dryer) is computed. The system uses this exergy during the following process stages. The exergy of the intake air at ambient conditions is zero.

| Description | Point | p bar abs | т °С | Q m³/min | P_{ε} elect. power kW | Exergy kW | lost Exergy % | remng. Exergy % |
|--------------------|-------|--------------|---------|-------------|--------------------------------------|--------------|---------------------|-----------------------|
| Start (intake air) | 1 | 1.0 | 20 | 10.15 | 0 | 63.6 | 0.0% | 100% |
| Compression | 2 | 8.3 | 80 | 10.15 | -61.2 | 39.3 | -38.2% | 61.8% |
| After cooler | 3 | 8.0 | 25 | 10.15 | -1.2 | 36.4 | -4.6% | 57.2% |
| Dryer | 4 | 7.8 | 20 | 10.15 | -1.15 | 34.8 | -2.5% | 54.7% |
| Filter | 5 | 7.6 | 20 | 10.15 | 0 | 34.4 | -0.6% | 54.1% |
| Storage | 6 | 7.6 | 20 | 10.15 | 0 | 34.4 | -0.0% | 54.1% |
| Pipe network | 7 | 7.5 | 20 | 9.74 | 0 | 32.8 | -2.6% | 51.5% |
| Service unit | 8 | 7.2 | 20 | 8.12 | 0 | 26.8 | -9.4% | 42.1% |

Tab. 1: Exemplary calculation for a typical industrial set-up

Based on the results for the exergy loss, the exergy flow diagram is illustrated in **Figure 3** (left diagram). It shows a final exergy efficiency of 42.1% at the service unit of the consumer. The shown case study represents a typical compressor unit with an energy consumption of 63.55 kW. It can be considered as a representative industrial set-up. Depending on the size of the compressor station, the dimensioning and the quality of the system, the given numbers can vary in a certain range. Especially the efficiency of small compressor unit is lower than for machines with > 30 kW.

A direct comparison with the results from Fig. 1 is difficult since system boundaries therein are not well-defined. Assuming that the consumption starts after the observation point "leakage losses" with a remaining energy of 7.8%, a big difference can be seen from the exergy flow diagram with a left energy of 42.1%.



Figure 3: Exergy flow diagram with (right) and without (left) heat recovery for a small compressor unit

3.3 Exergy flow diagrams with heat recovery

A considerable amount of heat is available from the thermodynamic process of compressing air. An important efficiency improvement of compressed air systems is the integration of a heat recovery system at the compressor to make this heat accessible for heating up water or heating up air. Heat recovery systems promise a recovery rate

corresponding up to 90% of the used electrical energy. Note that the phrase "recovery" is often misleading. The heat is not mainly extracted from the electrical input energy but it originates from the intake air (heat pump principle).

For an inclusion of the effects of the heat recovery in the exergy flow diagram, the exergy content of the heat can be calculated based on the Carnot efficiency (Eq. 3).

$$\eta = 1 - \frac{T_{atm}}{T_2} \tag{3}$$

This resulting exergy flow can be directly included in the exergy flow diagram. In the considered case study, it is assumed that 90% of the electrical energy input can be recovered as heat (here: 90% of 61.2 kW translated into 55.1 kW of heat). The exergy content of this heat amount can be transformed into work, e.g. by a thermal engine. With a temperature of 80 °C in point 2, the Carnot efficiency is 17%. Thus, the amount of exergy in the recovered heat is comparatively small (9.4 kW).

While the value of 9.4 kW depicts the correct amount of exergy in the recovered heat, is does not reflect the real benefit for the user, e.g. in terms of providing hot water. For this reason, it seems helpful not only to provide information on exergy content, but to provide information on the heat content which is 55.1kW. However, a common representation of exergy and heat within one flow diagram is not possible, since this would mix up state and process variables. Therefore, the exergy flow diagram is modified according to Figure 3 (right diagram) by adding an exergy boundary. The exergy of the heat recovery systems is exited by the exergy flow diagram to the left side. It is used to lift the temperature level in the compressed air to T_2 which makes the heat in the compressed air usable. The resulting arrow representing the heat flow is a graphical representation of the benefits from the heat recovery system, i.e. it represents potential energy savings if used to substitute for example oil or gas-fired water boilers.

4. Detailed study of exergy losses in pneumatic drive applications

So far, the flow diagram has ended before the application. The leaving exergy arrows in Figure 3 show the maximum power that is available to the application. For providing a uniform and closed exergy concept for the entire compressed air system, it is possible to further analyse the exergy flows in the application, because the behaviour of common pneumatic actuators is well-known and independent of a particular application. Exergy shows the entity of all energy forms that can be used within a system. Not only pressure differences but also temperature differences in reference to the environment are included in the exergy concept. In pneumatic applications, the temperature differences cannot be used. Only pressure is used as driving force for the execution of work. The amount of power that can be used in a pneumatic application is specified by the technical power:

$$P_{tech_a} = c_p \cdot \rho_N \cdot Q_a \cdot T_a \left(\mathbf{1} - \left(\frac{p_{atm}}{p_a} \right)^{\frac{\kappa - 1}{\kappa}} \right)$$
(4)

As expected, the technical power in point 8 (Table 1) is smaller than the exergy. Its value is 20.4kW, which is 76% of the incoming exergy or app. 32.1% of the exergy in point 1.

The pneumatic power includes a hydraulic and an expansion part. Some applications can use both parts to generate mechanic work, e.g. modern pneumatic turbine tools. On the contrary, commonly used pneumatic drives have a similar behaviour as hydraulic actuators. With the help of intelligent switching strategies, also pneumatic drives can use a certain amount of the expansion power, but when they are used in a conventional way, the expansion part of the compressed air's exergy remains unused. If only the hydraulic part of the pneumatic power is used, it can be calculated as follows:

$$P_{\mathbf{k}ydr_a} = \Delta p_a \cdot \dot{V}_a = (p_a - p_{atm}) \frac{\rho_N \cdot Q_a \cdot T_a \cdot R}{p_a}$$
(5)

In point 8, the hydraulic power is 11.6kW or app. 18.2% of the exergy in point 1.

When looking at the mechanical work available from a pneumatic cylinder, this value is in reality just a part of the ideal hydraulic power, because the friction in the cylinder needs to be overcome and a certain amount of power is additionally necessary for the acceleration of the load. Experience shows that in a well dimensioned system, up to 75% of the theoretic hydraulic power can be used for a mechanical movement.

$$P_{real_a} = 0.75 \cdot P_{hydr_a} \tag{6}$$

Thus in point 8, the real mechanical power in the pneumatic application is 8.7kW, which is 13.7% of the exergy from the beginning. This part of the exergy can finally be used in a commonly used pneumatic drive and represents the actual effective output.



Figure 4: Exergy losses in regular pneumatic applications

Figure 4 shows, which parts of the incoming exergy from the supply side cannot be used in a common pneumatic application and which part finally is available at the end of the system as mechanical energy. In the presented case study, it is possible to use 13.7% of the original input exergy, i.e. electrical power. Depending on the quality of the installation, the design and the degree of optimisation, this value can fluctuate strongly. Therefore it should not be considered as a generally valid figure. It rather represents one possible result in a defined case. Note further that the actual benefit for the user can often not be suitably quantified based on the energy concepts. In most applications, the benefit for the user is not a high energy output at the end of the functional chain, but the aim is to perform functions like positioning, holding and moving an object. With regard to a horizontal movement, the movement does not produce any "useful" physical work. Thus the energetic efficiency in this case would be 0%. Another example is the function "holding" which does not require any compressed air if the system has no leakage. This underlines that an application cannot be only judged reasonably by relying on the exergy concept. Therefore it is advisable not to include the pneumatic application in the exergy flow diagram and end it already before the application as shown in Figure 3.

5. Conclusion

The presented paper studies the use of energy flow diagrams for the discussion of efficiency of compressed air systems. An alternative approach using the thermodynamic concept of exergy is presented. This approach offers a straight-forward and unambiguous method for determining the efficiency of the system. An extension of this diagram allows for the integration of heat recovery systems. A study of pneumatic

applications shows that though it is in principle possible to include them in the flow diagram, there are various issues that argue of letting the flow diagrams end behind the service unit.



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6. References

- /1/ P. Radgen & E. Blaustein. 2001. Compressed air Systems in the European Union: Energy emission, saving potential and poly actions, L. V. GmbH.
- W. Gauchel. 2006. Energiesparende Pneumatik: Konstruktive sowie schaltungs- und regelungstechnische Ansätze, O+P 1/2006, pp. 33-39.
- /3/ S. Berchten & C. Ritz. 2006. Ersatz von pneumatischen und hydraulischen Antrieben durch Elektroantriebe – Potentialanalyse, Schlussbericht.
- /4/ R. Gloor. 2000. Energieeinsparung bei Druckluftanlagen in der Schweiz, Forschungsbericht (Projektnummer 33 564).
- /5/ E. Hahne. 2004. Technische Thermodynamik Einführung und Anwendung, Oldenbourg Wissenschaftsverlag GmbH.
- /6/ F. Ilmberger & F. Seyfried. 1994. Druckluftversorgungskonzepte für Industriebetriebe, BWK 46 (9), pp. 398-401.
- /7/ W. Bader & J.K. Kissock. 2000. Exergy analysis of industrial air compression, Proceedings of the Twenty-second National Industrial Energy Technology Conference, Houston, TX, pp. 89-98.

7. Nomenclature

| variable | description | dimension |
|---------------------------|---|-------------------|
| $c_v; c_p$ | heat capacity of air | J/(kg*K) |
| η | efficiency | |
| Ea | exergy (power) at point ^a | kW |
| κ | adiabatic coefficient | |
| Pea | electric energy (power) at point ^a | kW |
| P _{techa} | technical power at point ^a | kW |
| P _{hydra} | hydraulic power at point a | kW |
| P _{reala} | real mechanical power at point ^a | kW |
| p_a | absolute pressure at point a | Pa (Pascal) |
| p _{atm} | absolute atmospheric pressure | Pa (Pascal) |
| Qa | standard volume flow rate at point a | m³/s |
| Q | heat quantity | kW |
| R | air gas constant | J/(kg*K) |
| ρ_N | standard density of air | kg/m ³ |
| T _a | temperature at point ^a | K (Kelvin) |
| T_{atm} | atmospheric temperature | K (Kelvin) |
| Ua | inner energy (power) at point ^a | kW |
| ν _α | volume flow at point <i>a</i> | m³/s |

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