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11.5 Energy equivalent of compressed air consumption in a machine tool environment

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Abstract

Compressed air has many applications in machine tools. Compared to the potential energy stored in the fluid tank its production requires a large amount of energy. In addition to the potential energy, heat loss does occur as by-products. Dependent on the amount of heat loss, energy consuming cooling is required. For life-cycle investigations of machine tools, the gray energy and environmental impacts of compressed air consumptions have to be known. This work presents a theoretical approach to quantify the energy equivalent of compressed air and its by-products. A model based approach is set up to describe the required physical relationships for the compressor and its peripheral components. Measurements obtained from a shop floor compressed air supply have been used to validate the results of the theoretical approach. Concluding from the analysis, a general approach for the theoretical energy equivalent calculation, including the compressor and treatment of heat loss, is possible.

Keywords:
Compressed air equivalent, machine tool modeling, energy monitoring

1 INTRODUCTION

Compressed air is a frequently used energy carrier in machine tools. Some examples of possible applications are pneumatic components, tool handling or protection of machine components by sealing air. Compressed air is a resource consumed by the machine and therefore related to the energy and resource efficiency. Energy efficient machine tool design is an emerging topic. In this context, the ISO standard 14’955-1 is under development and has reached the level of a drafted international standard (DIS) [1]. For an energy analysis according to the new standard, all energies supplied to the machine tool must be known. Further, the energy content of all the resources must be expressed in a common unit, in order to allow a synthetisation of the different energy flows to the total consumption but also for comparison of different energy carriers.

Compared to other energy carriers, compressed air is a cost intensive medium [2, 3]. The cost is mostly caused by the electricity needed for the compressor supplying compressed air [4], amplified by inefficiencies and thermal effects making up to 93% of the electric input power [5]. Machine tools generate heat loss during operation; therefore excess heat is an issue of great importance. Whereas the supply of compressed air causes excess heat in a remote part of the factory, the use of compressed air, i. e. the decompression in a machine tool, represents a heat sink, causing a direct impact on its thermal conditions.

Power measurements on machine tools have shown a substantial share of energy for the thermal conditioning of the manufacturing process and of the machine tool itself, including compressed air consumption [6]. For the evaluation of energy efficiency of machine tools as well as for modeling for analysis and optimization purposes, quantification of energy needs on component level by selective measurements are essential. This is required by ISO/DIS 14’955 [1] and is successfully demonstrated by Gontarz et.al. [7]. In case of multiple energy supplies, i. e. electricity and compressed air, the need of a common unit for consolidation and respective conversion equivalents is obvious.

Goal of this work is the derivation and validation of an electrical energy equivalent for compressed air used by a machine tool. As the machine tool environment is a thermal sensitive area, heat sources and heat sinks must be evaluated with care. The equivalent must further be adaptable for various conditions, e.g. different pressure levels, system characteristics, operational schedules and degrees of system integration.

2 STATE OF THE ART

Electric energy equivalents for compressed air can be obtained from measurements or estimated by models. Measured energy equivalents are specific for a certain compressed air system configuration, whereas models allow a generic parametrizable approach. Gauchel [5] introduced assessment approaches for compressed air, that are in line with assessment and improvement approaches from Energy Schweiz [8]. Joseph and D’Antonio [9, 10] showed different ways for the compressed air system assessment. The authors mentioned a production effort of 0.12 to 0.27 kWh/m^3 at nominal conditions for supply pressures in the range of 7 to 8 bar. Nominal conditions are defined in DIN 1343 [11]. For the sake of simplicity only the unit ‘m^3’ is further on used, referring to cubic meter at normal conditions. Modeling of compressed air supply from the view of energy consumption was performed by Schmidt et.al. [12], D’Antonio [10], and Hütter [4]. The theoretical values for adiabatic compression are mentioned by Harris et.al [13] to be in the range of 0.08 to 0.10 kWh/m^3, dependent on the pressure. The authors as well demonstrate the quantification of heat recovery influence to the system efficiency by an exgergy based approach.

The resulting compressed air energy equivalents of Schmidt, D’Antonio, Hütter and Harris are in the range of 0.08 to 0.14 kWh/m^3. Application and use of such equivalents were successfully shown in energy monitoring and assessments

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[14, 15]. All mentioned publications of compressed air energy equivalents consider the ratio between electrical power input to the compressor and volumetric flow per time. Energy consumptions due to secondary effects, like the treatment of heat loss, are not included. A more comprehensive energy equivalent for compressed air is required, which includes the thermal effects and distribution system pressure losses as well. Thermal effects and losses are in general very specific for each site. The new approach is thereto demanded to be adoptable to different systems, which leads to a model based equivalent. Promising modeling approaches of compressed air consuming devices by Harris et.al. [16-18] encourage a model based approach for the compressed air production electric power demand as well.

3 METHODOLOGY

In order to fulfill the above mentioned goals and requirements, a three-stage procedure is applied:

1. Identification and modeling of the relevant electric energy, pneumatic work and heat flows within a compressed air system.
2. Formulation of the theoretical energy equivalent for compressed air based on the results of the modeling.
3. Validation of the theoretical results and assumptions from the system modeling by measurements within a case study.

To derive the required energy equivalent for compressed air, a model based approach is presented. The used system model and its boundaries are shown in Figure 1. The model inputs consist of the demanded amount of compressed air V_n and inlet air conditions, represented by pressure p_{in} and temperature ϑ_{in}. The outputs of the model are the required electric energies W_{el} and thermal losses Q_{th}.

Given this dependency and the system boundary above, the energy equivalent in this approach is defined as:

$$C_{cair} = \frac{\sum_j W_{el,j} + \sum_i W_{th,i}\left(Q_{th,i}\right)}{V_n} \tag{1}$$

Including the electric components with consumptions $W_{el,j}$, and the function $W_{th,i}(Q_{th,i})$, describing the energy required to treat the i-th thermal loss $Q_{th,i}$. Objective of the following sections is to display the electric power demand and heat generation as a function of the consumed compressed air V_n at normal conditions and quantify the energy demand including thermal losses.

In order to identify the relevant energy flows – electric and thermal – a simple model is derived from the system described in Figure 2: Air enters the air system at certain conditions, is compressed by the compressor and cooled to the temperature ϑ_{out}. The compressed air is stored in a vessel until it is used by a machine tool. As shown in Figure 2, the inlet, production and consumption are locally separated, preventing heat exchange in between.

Production of the compressed air includes the compressor, the cooler and the vessel. Since the cooler is often integrated in the compressor, both components will be modeled as one. Objective of this system is the provision of the systems downstream with compressed air at pressure level p_{comp}. For this process, the following model is used: The pressure is raised adiabatically by $\delta p \ll p_{comp}$, resulting into a temperature raise. A subsequent isochoric cooling lowers the intermediate gas temperature to ϑ_{in}. Compression and cooling are repeated, until p_{comp} is reached. Using the first law of thermodynamics for a steady state system, the following energy demand for the compressor results:

$$W_{comp} = \frac{1}{\eta_{comp}}\left[V_n \cdot \rho_n \cdot R_s \cdot \vartheta_{in} \cdot \ln\left(\frac{p_{comp}}{p_{in}}\right)\right] \tag{2}$$

The term in square brackets from Equation (2) represents the fluid dynamical work required for the compression, where the first part includes the combined motor and shaft efficiency η_{comp} of the system. Gas properties are included in the specific gas constant R_s and the density ρ_n at normal conditions. Heat losses generated by the compressor due to friction and ohmic losses are described by the compressor efficiency η_{comp} as well:

$$Q_{comp} = W_{comp} \cdot \left(1 - \eta_{comp}\right) \tag{3}$$

The isochoric temperature drop of the compressed gas within the cooler requires a heat flux $\dot{Q}_{extr}$. Integrating this flux results in the total thermal energy Q_{extr}:

$$Q_{extr} = V_n \cdot \rho_n \cdot R_s \cdot \vartheta_{in} \cdot \ln\left(\frac{p_{comp}}{p_{in}}\right) \tag{4}$$

Equation (4), in comparison with the fluid dynamical work in Equation (2), reveals that all compression work is transformed into heat energy. In other words, isothermal compression does not increase the internal energy per unit of the gas, but makes it exploitable due to the pressure drop relative to the ambient conditions.

A compressor can operate in loading, during actual compression, or idle cycle, while the compressor is running, but no air is delivered. Reasons for idle cycle can be cooling functions or ready state of the system [3, 12]. During loading

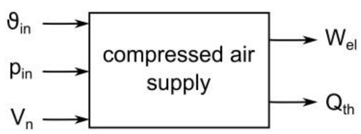

Figure 1: Boundary and interface of the discussed system.

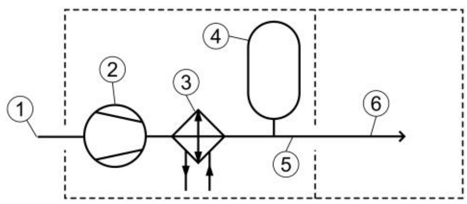

Figure 2: Simplified system model consisting of inlet (1), compressor (2), cooler (3), vessel (4), distribution system (5) and consumer (6).

cycle the system requires the electrical energy for the compressor and the cooler, while both components are generating heat loss:

$$W_{load} = W_{comp} + W_{cool} \quad (5)$$

$$Q_{load} = Q_{comp} + Q_{extr} \quad (6)$$

As mentioned by Hütter [4], the power consumptions during idle cycle are about 33% of the power demand during loading cycle. The duty cycle u represents here the ratio between the time of loading operation and the total time of operation, including idle time. With this information the total electrical energy demand and idle heat loss can be calculated according to Schmidt et. al [12] to:

$$W_{idle} = W_{cool} + W_{comp} \cdot \frac{1-u}{3u} \quad (7)$$

$$Q_{idle} = W_{comp} \cdot \frac{1-u}{3u} \quad (8)$$

The total Energy consumed and released per consumed volume V_n of compressed air by the system is now given as:

$$W_{el} = W_{load} + W_{idle} \quad (9)$$

$$Q_{th,1} = Q_{load} + Q_{idle} \quad (10)$$

respectively. Similar as for the efficiency dependent compressor power, the electric power required for the cooler is described by the coefficient of performance ε_{cool}:

$$W_{cool} = \frac{Q_{load} + Q_{idle}}{\varepsilon_{cool}} \quad (11)$$

The last element of the compressed air production system is the air vessel, an accumulator of potential energy. The compressor is operated such that the tank pressure is always between an upper and lower limit: $p_{tank} \in [p_{low}, p_{high}]$. Variation between the two pressure limits will cause a change of gas and vessel structure temperature, where the vessel structure is typically made of steel. Hence the structure is in contact with the ambient air, the vessel forms a heat sink or -source, dependent on its current temperature. In this context, it is assumed, that this heat flux can be neglected compared to compressor heat release. With this assumption and equations (9) and (10) the generative part of the system is fully described.

After the generation, compressed air enters the distribution system. In non-ideal distribution systems leakages occur. Leakage flows have to be compensated by the compressor. Given the area A_{leak} as the sum of all leak cross-sections and the total operational time t_{op}, during which the volume V_n of compressed air is consumed. Under the assumption of an isenthalpic process, the leakage loss is calculated as:

$$V_{leak} = t_{op} \cdot A_{leak} \cdot \frac{p_{comp}}{p_{in}} \cdot \sqrt{\frac{R_s \cdot \vartheta_{in}}{2}} \quad (12)$$

To model the fast expansion of the gas during the consumption, the process is separated into two parts: An isothermal expansion, followed by an isobaric expansion. The thermal energy required for the isobaric expansion is extracted from the surrounding air and can be estimated by the first law of thermodynamics as:

$$Q_{cnsp} = Q_{th,2} = c_p \cdot V_n \cdot \rho_n \cdot \vartheta_{in} \cdot \left[\left(\frac{p_{in}}{p_{comp}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (13)$$

Since $p_{in} < p_{comp}$ the consumption forms a heat sink. This is obvious, since the energy required for the expansion of the gas is drawn from the surrounding.

Summarizing the results from the identification and modeling of energy and heat flows, we have the compressor and the cooler as electrical energy consumers. Further, a heat source and sink are identified: The generative part of the system generates heat loss, where the use of the compressed air draws energy from its surrounding.

In industrial applications, heat sinks and sources raise the question about additional costs and compensation. Quantifying the costs of heat loss requires knowledge about the heat management strategy. The heat sink due to decompression causes no extra costs, because excess heat is generally available in the surrounding. But the important heat source, i.d. the compressor, has a significant impact on its surrounding. Further, for simplicity reasons a linear expression of the cost is assumed:

$$K_{th}(Q) = k_{th} \cdot Q \quad (14)$$

The energy demand of the treatment is characterized by the re-use factor k_{th}. Heat dissipation can cause extra costs or can support the heating of the building. Investigating modern heating, ventilation and air-conditioning systems (HVAC) for buildings, different values for the cost factors are possible, as shown in Table 2.

Using the results of the last section, Equation (1) can be evaluated as:

$$C_{cair} = R_s \cdot \rho_n \cdot \vartheta_{in} \cdot \ln\left(\frac{p_{comp}}{p_{in}} \right) \cdot C_{sys} \cdot C_{op} \cdot C_{leak} \quad (15)$$

With

$$C_{sys} = \frac{1 + \varepsilon_{cool} \cdot (1 + k_{th})}{\eta_{comp} \cdot \varepsilon_{cool}} \quad (16)$$

$$C_{op} = \frac{2 \cdot u + 1}{3 \cdot u} \quad (17)$$

$$C_{leak} = 1 + \frac{A_{leak}}{\dot{V}_{n,avg}} \cdot \frac{p_{comp}}{p_{in}} \cdot \sqrt{\frac{R_s \cdot \vartheta_{in}}{2}} \quad (18)$$

Analyzing the structure of equation (14), the specific energy for isothermal compression scaled by system, operation and leakage specific factors can be identified. The system specific factor C_{sys} includes the performance factors of the

Table 2: Examples for possible re-use factors on different scenarios [20-22].

Scenario	re-use factor [kWh/kWh]
Heat exchange with the ambient air. The ambient air is conditioned by the HVAC.	0.3
The building is heated by electricity only. Compressor heat loss is used to heat the building.	-0.9
The building is heated by geothermal energy. Compressor heat loss is used to heat the building.	-0.25

compressor and the cooler, as well as the costs of heat sources. This costs represent the degree of system integration: Positive values indicate low system integration; e.g. the thermal energy is not used and causes additional costs. Vice versa are negative values an indicator for advanced system integration; e.g. the heat energy is used for other processes. Leakage performance is expressed by C_{leak}. This factor depends on operational settings, gas properties, leakage size, as well as average flow rate $\dot{V}_{n,avg}$ over the whole time of operation.

Examples for numerical evaluations of the compressed air equivalent are shown in Table 1, where different efficiencies, pressure levels and degrees of system integration are compared. Resulting equivalent values are within the range of 0.08 to 0.90 kWh/m^3. For high efficient systems, the estimated equivalents between 0.08 and 0.18 kWh/m^3.

4 VALIDATION

Within the previous section, a method for the calculation of the energy equivalent of compressed air in a machine tool environment has been shown. The following points need to be validated comparing computed results to measurements of real systems:

- Consistency between the measured electric power demand per volume compressed air and the calculated ratio
- Insignificancy of the heat flux over the air vessel surface

Using a measurement system according to [7] installed on a test system, the required data is collected and used for the validation.

4.1 Measurement set-up

The test system consists of a state of the art screw compressor of a European manufacturer built in 2012; with a rated motor power of 15 kW and a throughput of 2.11 m^3/min. Excess compressor heat is extracted by a fan and emitted to the ambient air. Connected to the compressor is an air vessel with a volume of 250 l. Forced by the compressor control, the vessel pressure is always between 10 and 11 bar over ambient pressure. Instead of a machine tool, an orifice is used to simulate a consumer. This orifice leads to a constant flow rate of 3.75 l/s. Due to this configuration, the compressor operates at an average duty cycle of 20% during a constant consumption. This configuration represents a rather small installation for compressed air generation.

The measured variables are the power consumption of the compressor, the volumetric flow thought the orifice, as well as the temperatures of the ambient, the vessel surface and the

Table 1: Examples of estimated compressed air electric energy equivalents for different systems, pressure levels and system integration (low efficient: u=20%, ε=2, η=30%; high efficient: u=60%, ε=4, η=80% [19]). The units of the listed equivalents are kWh/m^3 at normal conditions.

	No secondary thermal treatment	Air-Air heat exchange	Connection to HVAC, moderate climate	Connection to HVAC, cold climate
Description	Only electrical power has to be taken into account, since no secondary treatments of heat sinks and sources are required.	Heat loss is exhausted to the ambient air, which is conditioned by the HVAC of the building.	The compressor is connected to the HVAC system of the building. During six months per year the excess heat is used for heating.	The compressor is connected to the HVAC system of the building. The excess heat is used to heat the building.
6 bar				
low efficient	0.55	0.65	0.50	0.45
high efficient	0.11	0.13	0.10	0.08
8 bar				
low efficient	0.63	0.75	0.58	0.53
high efficient	0.12	0.15	0.11	0.10
10 bar				
low efficient	0.70	0.84	0.64	0.58
high efficient	0.14	0.17	0.12	0.11
12 bar				
low efficient	0.75	0.90	0.69	0.63
high efficient	0.15	0.18	0.13	0.12

exhaust air of the compressor cooler. Expected relative errors are in the range of ±4.5% for power measurements and ±3.5 % for air flow capturing. For the temperature measurement a multi-probe system from *Hygrosense* with an absolute error of ±0.3 K [23] is installed. The measurement is performed over a sufficient amount of load cycles. Resulting are the time series of the electrical power demand of the compressor, volumetric flow through the orifice and the temperatures of the ambient air, exhaust air of the cooler and the vessel surface. A cut-out of this data is shown in Figure 3.

4.2 Measurement analysis

Based on the measurement results, the energy per volume compressed air at nominal conditions can be calculated:

$$\overline{C}_{cair} = \int_{t_1}^{t_2} \overline{P}\, dt \Big/ \int_{t_1}^{t_2} \dot{\overline{V}}\, dt \approx 0.55\,\text{kWh/m}^3 \tag{19}$$

In order to compare the measured value to the calculated value, the method described in Section 3 is applied for the present case. Based on the technical fact sheet of the compressor, a compressor efficiency of 40 % and cooler efficiency ratio of 2 can be calculated. Using the required conditions of DIN 1343 and the ideal gas properties of air all parameters for the calculation are known [10, 11, 24, 25]. With these parameters and equations (1), (15) and k_{th}=0 – since no heat recovery takes place, and the impact to the HVAC is not measurable – an estimated compressed air equivalent results:

$$C_{cair} \approx 0.57\,\text{kWh/m}^3 \tag{20}$$

Comparing the results of Equations (20) and (19), a relative error of 4% in the energy equivalent calculation can be identified.

During the measurement, also the temperatures of the vessel surface, as well as the temperature of the ambient air have been measured. From this measurement, a maximum temperature difference of 1 K is identified. For the vessel a surface of 4.5 m² can be estimated from its geometry. Under the assumption of free convection [24] with a convection constant of 5 W/m²/K, the maximum heat flux between the vessel and the ambient air can be estimated:

$$\left|Q_{vess}\right|_{\infty} = \alpha \cdot A_{vess} \cdot \Delta\vartheta \approx 23\text{W} \tag{21}$$

Compared to the rated power of 15 kW, the expected heat flux over the vessel surface is in fact negligible.

5 CONCLUSION

A model based approach for the evaluation of the energy equivalent of compressed air in a machine tool environment has been introduced. The application and validity of the approach have been demonstrated by measurements on a test system. The most sensitive parameters in Equation (15), are the compressor efficiency and the duty cycle, followed by the coolers energy efficiency ratio and the compression ratio. This observation is consistent with recommendations for compressed air system improvements [2, 4, 5, 8]. Hence, selection of the right energy equivalent is dependent on the available compressed air systems and degree of system integration. Selection of an equivalent should be done for each system individually. For high efficient systems in general, the equivalent is within the range of 0.11 and 0.15 kWh/m³, dependent on the compression level. For different levels of system integration, this value may change within ±20 % of the value above. For smaller systems, although state of the art, significant higher electrical equivalents must be assumed. Leakage is an important topic as well. Given the situation of the measurement setup, a leak of 0.5 mm² would cause a 30 % higher energy equivalent. The quantification of leakage in a shop floor environment is a challenging task, which is not discussed here. Interested readers are referred to publications such as [26] or [27].

In relation to measurement and monitoring applications the given approach and validation confirms the applied energy equivalent for energy efficient compressors without system integration. Compared to other compressed air energy equivalents, the given approach enables the quantification of heat loss treatments and savings through system integration. Furthermore it enables a more accurate energy equivalent for monitoring purposes on specific compressor types and configurations, without any measurement needed. This offers a significant advantage in the case of an assessment, where no direct measurement of the air compressor is possible.

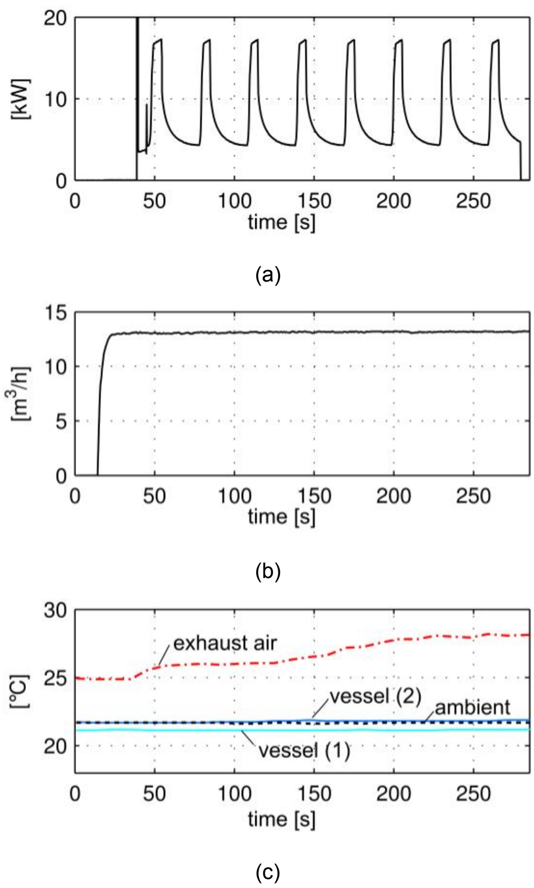

Figure 3: Cut out of the measurement results; with compressor power (a), compressed air consumption (b) and temperatures (c). The measured temperatures are ambient air (dashed), exhaust air of the fan (dash-dotted) and the vessel surface temperature at two locations (solid).

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