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ABSTRACT: Compressed air is the most common energy supply alongside electrical energy in manufacturing. In addition to the positive properties such as clean application possibilities, safe usage or storability, the compressed air is an underestimated energetic factor. Compressed air supply on machine tools therefore requires particular quantification in machine tool energy assessment and further energy monitoring applications. These energy monitoring applications, which are able to quantify different and combined energy forms, e.g. electricity and compressed air, are not yet available or often based on assumptions.

The following paper evaluates the physical behavior of compressed air and defines the key efficiency parameters within a compressed air system. This knowledge enables the approximation of compressed air in relation to electrical energy and extended assessment possibilities for further measurement, monitoring and optimization purposes.

1 INTRODUCTION

The evaluation and assessment of energy and resources used in manufacturing is addressed in research, legislation and industry. In this regard energy assessments are currently performed by assumptions, modeling or measurement. For compressed air Saidur et al. [2] indicate that reliable information for energy efficiency is not yet present. In accordance with the importance [1, 3] and costs [4] of compressed air, consciously or unconsciously careless use of compressed air for multiple applications can still be observed.

In standardization the ISO/DIS 14955-1 [5] defines the energetic machine tool system boundary for the assessment of resources, including electricity, compressed air and other energy forms, which are used to perform manufacturing processes. A compressed air system is an assemblage of different components such as the compressor, filters, coolers, branched pipes, valves and nozzles. Each of these components represents an energy loss in the form of flow or pressure loss in the system, and these losses must be assessed and taken into consideration when comparing and using energy equivalents.

In machine tools energy in the form of compressed air is generated by integrated equipment or - more often - supplied by the shopfloor. In correspondence with the system boundary definition, not only measurement applications according to Avram et al. and Gontarz et al. [6, 7] but also monitoring applications [8] have to combine various energy forms and resources in order to assess, analyze, compare and optimize the total machine tool energy efficiency within the machine tool system boundary.

This paper reviews potential approaches towards energy assessment through a sensible combination of measurements with simulations based on detailed knowledge of the physical behavior of the components of compressed air systems.

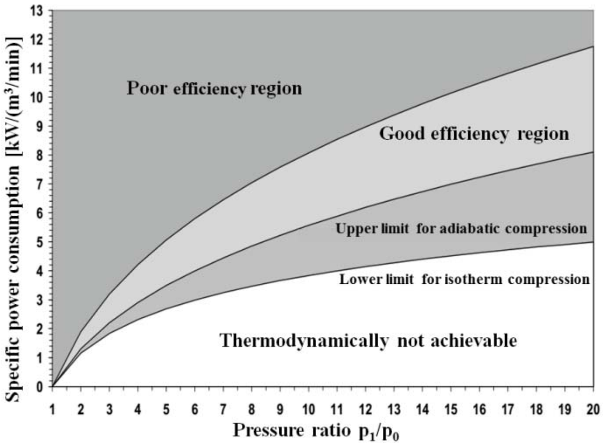

Figure 1. Specific power consumption for compressed air generation [1]

The target of the assessment is the design and quantification of energy saving measures, e.g. leakage prevention, the selection of optimal compressor type, the dimensioning and layout of the distribution network and the use of valves and nozzles. The added value of the approach is a deeper analysis of available or not yet available systems without excessive efforts for measuring on the component level, a step which is time consuming, costly and at times difficult to implement. Key findings are validated punctually against measurements in order to determine the correctness of the approach.

2 STATE OF THE ART

In literature different energy equivalents can be found depending on the compressor type, its capacity, pressure and temperature level. Hinsenkamp et al. [9] indicate the specific power consumption of a screw compressor for 6 bar supply of 0,1006 kWh/m^3 at a capacity of 90%; this value rises to 0,1286 kWh/m^3 at 50% capacity. Figure 1[1] shows an overview leading to < 0,1 kWh / m^3 for the most efficient compressors.

The given disparity depends on the physical behavior of the compressed air system as well as on cooling, heating and other energetic effects. In the following section narrower focus is given on compressed air subsystems and current assessment tools.

2.1 *Assignment of inefficiencies*

One of the main subsystems is represented by the compressor. Akbaba [10] states that savings of 8% are possible with high-efficiency motors made up of low-loss materials and constructive improvements in comparison to standard motors. Brunner et al. [11] from the Swiss initiative Topmotors indicate improvement potentials of 20-30%. The improvement of the compressor geometry, in example on a new profile of a screw rotary compressor, leads to 15% as shown by KAESER [12]. An important method for energy saving to be achieved is seen in the heat recovery; with a heat exchanger, the hot compressed gas is cooled by a cooling fluid, e.g. oil. The use of this resulting thermal energy is also discussed by Saidur et al. [2].

An essential element within the compressed air system is given by the filter, which, dependent on the application, results in a pressure drop. Wang et al. [13] and Del Fabbro et al.[14] introduced a model representing the pressure drop at filters; a simplified approach from pressure drop estimation is given by the Darcy equation [15]. Pressure drops with local compressed air supply are possible due to corresponding distribution, pipes and fittings. Coelho et al. [16] focused on energy dissipation by viscous flows. Brkic [17] analyzed friction factors for different flow regimes with approximation to the Colebrook relation. For pipe fittings, e.g. T-pieces or elbows, the empirical derivated values of Atlas Copco [18] are commonly used and represent an equivalent length of a straight pipe, for which the calculations explained above can be used. Major energy losses are caused by leakage in lenghty and widespread industrial distribution pipe systems for compressed air. An overview of the impact of leaks as well as the industrial procedure to evaluate the volumetric leakage flow are represented by Fraunhofer ISI [1]. Siekmann [19] introduces a model approach for the evaluation of the volumetric leakage flow.

2.2 *Simulation of compressed air*

Energy oriented simulations on the manufacturing level including the simulation of compressed air are introduced by Herrmann [20] and Thiede et al. [21] while simulation approaches on the machine tool level are given by Avram [6] and Gontarz et al.[22]. For the analysis and evaluation of compressed air systems the software tools AirMaster+ [23] and Air-Sim [24] are available. These approaches are either too detailed and thus require multiple parameters and detailed measurement values or valid for a particular system layout only, generally limited to the compressor and related components.

2.3 *Compressed air measurement*

The quantity of air is measured by the measurement of flow $\dot{V}$ at the given pressure or mass flow $\dot{m}$ with built in measurement equipment, e.g. calorimetric or coriolis sensor. Required measurement parameters are flow $\dot{V}$, pressure p and media temperature T. With the measurement based on reference conditions according to DIN ISO 1343 [25] and a standard 6 bar air supply pressure, media temperature can be neglected. Turbulences in the flow influence the measurement in calorimetric measurements, therefore an upstream pipe length of 20 times the pipe diameter, and a respective downstream length of five times is required. For the quantification of the required energy for compressed air generation and distribution the electric power consumption of the compressor in relation to the measured media flow has to be considered. If this measurement is not applicable, e.g. due to complex distribution systems, given power equivalent should be used.

3 METHODOLOGICAL MODELING APPROACH

For the evaluation of compressed air and the quantification of inefficiencies in the form of flow and pressure losses, the compressed air system is classified into six subsystems with respective influencing factors: generation, filtration, dehydration, distribution including leakage, valves and nozzles, and end use on a machine tool or for other purposes.

In a second step the thermodynamic and fluid dynamic behavior for each subsystem is assessed followed by the transcription into a modular modeling, based on known system parameters. Due to the specific system configuration, the design of infrastructure and therefore individual system parameters, inefficiencies cannot easily be identified by a rule-based procedure. In the following sections special focus and modeling examples are given for flow loss, caused by leakage, and pressure drops due to flow friction within the system distribution.

Based on this modeling approach and its validation, options for compressed air energy quantifications for monitoring applications are evaluated.

3.1 *Example: Modeling of leakage*

Leakage is a direct flow loss without any use. It is dependent on the system design, the choice of components, their actual state of wear or aging, and the usage; therefore special importance is given to leakage air over time, for instance, due to corrosion and aging of hoses and fittings, and through unsealed filters. Leakages often represent a significant cost factor, as shown with some sample values on the basis of a 6 bar supply with an electrical equivalent power of P_{eq} = 7 kW/(m^3/min), 365d/24h operation and electricity cost of 0.15€/kWh.

Table 1. Examples of the costs for an air leak through a hole with different parameters [12].

Hole diameter	Air consumption	Leakage loss	
mm	m^3/min	kW	€/year
1	0.065	0.46	604
2	0.257	1.80	2364
4	1.03	7.21	9474
6	2.31	16.17	21247

There are two possible ways for leak evaluation:

A) Determination of the leakage volumetric flow V_L. According to Fraunhofer [1] the time is measured while the air tank empties from a start pressure p_s to an end pressure p_e. This measurement can be used for the quanitfication of leakages.

$$\dot{V}_L = \left(\frac{\dot{V}_c \cdot (p_s - p_e)}{t} \right) \tag{1}$$

B) Evaluation of the leakage with an ultrasonic gauge or with infrared thermography. This measurement can be used for the detection of leakages.

Neither evaluation method allows a predictive estimation of leakage of a system before installation; therefore the physics of leakage are further evaluated. Figure 2 shows an idealized leak model. State 1 represents a fully developed flow without any influences of the leakage while state 2 represents the orifice point by the flow downstream the leakage. Ambient conditions are assumed outside of the pipe.

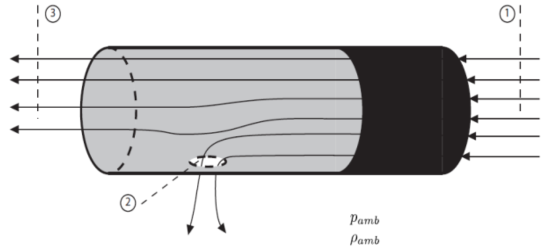

Figure 2. Model of an idealized discharge through a leak orifice.

For the discharge behavior the Bernoulli equation for compressible flow is used:

$$\frac{\gamma}{\gamma - 1} \cdot \frac{p_1}{\rho_1} + \frac{\omega^2{}_1}{2} = \frac{\gamma}{\gamma - 1} \cdot \frac{p_2}{\rho_2} + \frac{\omega^2{}_2}{2} \tag{2}$$

with the adiabatic coefficient γ, pressure p, media velocity w and density ρ.

This equation can be simplified according to Siekmann et al. [19] by assuming a free jet at orifice with $p_2 = p_{amb}$ and $\rho_2 = \rho_{amb}$, negligible velocity in the pipe with ω_1 = 0 m/s and an adiabatic process p/ρ^γ the velocity at the exit ω_2 to:

$$\omega_2 = \frac{\sqrt{2\rho_1 p_1}}{\rho_{amb}} \sqrt{\frac{\gamma}{\gamma - 1} \left[\left(\frac{p_{amb}}{p_1} \right)^{\frac{2}{\gamma}} - \left(\frac{p_{amb}}{p_1} \right)^{\frac{\gamma + 1}{\gamma}} \right]} \tag{3}$$

The discharge function remains constant for the critical pressure relation $(p_{amb}/p_1)_{crit} \leq 0.528$. This phenomena is called choked flow. The desired leakage volumetric flow is ultimately obtained by multiplying the outflow velocity w_2 with the orifice cross section A_2:

$$\dot{V}_L = \omega_2 \cdot A_2 \tag{4}$$

This model is used for the modeling of leakages and will be validated in Chapter 4.

3.2 *Example: Modeling of the distribution system*

In industrial environments the compressed air distribution represents a major pressure loss within a compressed air system; including all pipes and fittings in between the compressor and the end use. A widely used possibility to calculate the pressure drop between two points is given with:

$$\Delta p_{friction} = \frac{\rho}{2} \overline{w}^2 \cdot f\Big(\underbrace{\frac{L}{D}}_{\substack{Friction \\ part}} + \underbrace{\sum \zeta}_{\substack{Fittings \\ part}} \Big). \tag{5}$$

The friction is represented by the Darcy-Weisbach equation. L is the characteristic length of the straight pipe, D the inner pipe diameter and f represents the Darcy friction factor.

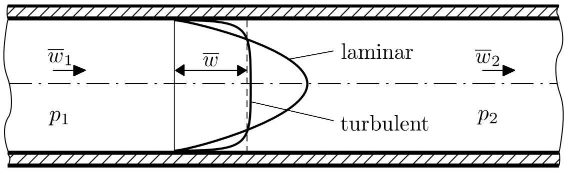

Figure 3. Model of a straight pipe with a laminar and turbulent velocity profile.

For the calculation of this dimensionless factor the flow behavior must be considered. Figure 3 shows the velocity distribution in a straight pipe for laminar and turbulent flow. The friction factor is determined by the Reynolds number which leads to different pipe surface friction.

$$Re_D = \frac{\overline{w} \cdot D}{\nu} \tag{6}$$

$\overline{w}$ is the mean flow velocity and υ is the kinematic viscosity. Up to a critical Reynolds number Re = 2300, a laminar flow is given and the friction factor results in:

$$f = \frac{64}{Re_D} \tag{7}$$

For turbulent flow, Re > 2300, f depends not only on the Reynolds number but also on the pipe surface roughness k_s. For smaller Reynolds numbers and smooth surface roughness $\omega^* k_s/\upsilon < 5$, where ω describes the shear stress velocity, the following formula is used:

$$\frac{1}{\sqrt{f}} = 2.0 \cdot \log\left(Re_D \cdot \sqrt{f}\right) - 0.8 \tag{8}$$

For the model implementation, various supporting points were calculated numerically and implemented into a lookup table. Based on this table, linear interpolation is used in the simulation. To obtain f for very rough pipes with $\omega^* k_s/\upsilon > 70$, von Karman's explicit formula is used:

$$\frac{1}{\sqrt{f}} = 1.14 - 2 \cdot \log\left(\frac{k_s}{D}\right) \tag{9}$$

As in the entire distribution system transitional roughness is often observed. Colebrook and White [26] developed a curve fit. A simplified alternative for the modeling is used with the explicit Chen approximation according to Brkic [17]. Together with the known pipe length and the gas properties, the pressure loss can be calculated for constant pipe diameter.

In order to quantify the pressure loss at fittings in this simulation a widely used empirical method is chosen. Depending on the fitting geometry an equivalent straight pipe length $L+$ can be obtained which represents the same pressure loss. Atlas Copco [18] and Miller [27] tabulated the values for different pipe diameters.

4 MEASUREMENT AND VALIDATION

For the validation of the findings and implemented models a defined leakage test was performed. As shown in Figure 4 a defined 2mm (8) and 4mm leakage was measured and compared against the defined model. The entire electrical power input on the compressor (1) against the flow (7) was measured. The tank (2) was filled up to a system pressure at the manometer (3) to 10.6 bar. A ball valve (4) was opened and released the stored compressed air through a high pressure hose (5) and filter (6) to a test pipe with the defined leakage (8).

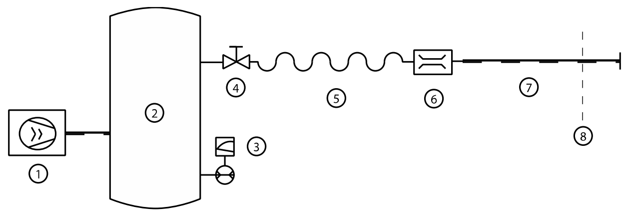

Figure 4. Piping and instrumentation diagram of the leakage measurement set-up.

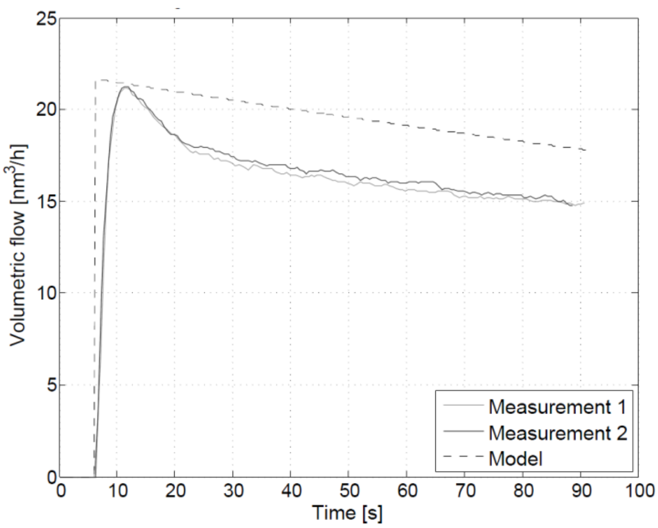

Figure 5. Comparison of measurement and model

The leakage model reflects correctly the volumetric flow over time (Figure). As friction and the dynamic accumulated pressure behavior are not modeled the model does not reflect an initial transient behavior and produces an offset in the long run. The offset is caused by the neglected friction and the measurement accuracy on the flow sensor.

5 CONCLUSION

The given approach and developed model is based on the fundamental physical behavior of the subsystems of the compressed air system and has illustrated that it is possible to estimate the needed compressor dimension and predict its power consumption according to defined parameters. Compared to available software tools it is modular and based on available unmeasured or assumed parameters. In relation to flow measurements, where only the actual flow use of the machine tool including leakage can be quantified, this approach enables the user to quantify energetic weaknesses. In the long term this set of subsystem models included into a software tool is seen as an extension to current measurement and monitoring approaches which enriches the evaluation of compressor efficiency. Depending on given monitoring and evaluation purposes this approach is suitable for compressed air system quantification and optimization.

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