Mathematical model for a compressed air system that couples demand and supply

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Abstract: This paper presents a mathematical model for a compressed air system (CAS) that couples system supply and demand. The supply side contains components responsible for production, treatment and storage of compressed air, while the demand side contains components that deliver and consume compressed air. Components considered include compressor, cooler, storage tank, linear actuators and an air blower. Simulations were performed to study the impact of pressure regulation and storage tank size on system energy consumption. Results showed that pressure regulation reduced air and energy consumption and a properly sized tank volume reduced energy consumption while maintaining good system pressure stability.

Keywords: compressed; air; systems; modelling; simulation.

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1 Introduction

Compressed air has been considered safe, easy to use, store and transport (Nehler, 2018). Because of these and other favourable characteristics, compressed air has been widely used in industrial plants (Saidur et al., 2010). Significant amounts of energy have been consumed by compressed air systems (CAS) globally. Previous studies indicated that CAS consumed 10% of the total annual industrial electrical energy in the UK (Thabet et al., 2020). CAS has often been criticised for their low energy efficiency since 81% or more of energy supplied to CAS was wasted (Benedetti et al., 2018). These statistical figures suggest that improving energy efficiency of CAS may lead to considerable energy saving.

A modern CAS is formed of several sub-components (Lawrence Berkeley National Laboratory, 2003), including a compressor, cooler, filters, dryers, tank, pipes and end-user tools. It has been common to divide a compressed air system into a supply and a demand side. The supply includes components where compressed air is produced, treated and stored (compressor, cooler, filter, dryers, tank, etc.) while the demand side has consisted of the distribution network and air consuming tools, such as pneumatic devices (Lawrence Berkeley National Laboratory, 2003).

Experimental evaluation of CAS performance under different operational conditions is time consuming, challenging and sometimes unfeasible. An alternative, is to use computer based simulations. Computer simulations allow the evaluation of changes to a system and operating conditions at a minimal cost. Researchers and engineers often rely on models and computer simulations to evaluate and optimise component design, control strategy and overall system performance. More recently, models have been used to simulate system faults and develop monitoring and diagnostic technologies (Watton, 2007).

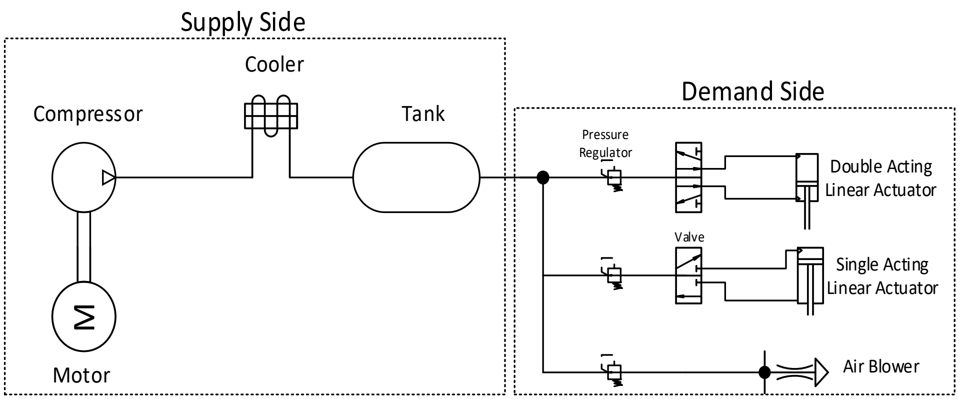
Past models have either focused on modelling the supply side, demand side or individual components. These models were crucial for studying and optimising generation and consumption of compressed air, however, they did not capture the dynamic interaction between the demand and supply side. The model presented in this paper considered both demand and supply side. The model is mathematical and based on first principles relation, ideal gas equations and equation of air flow through a nozzle. The objective of the model is to obtain a better understanding of the interaction between the demand of compressed air and the energy consumed in its generation. Moreover, the model could be used as a tool for evaluating changes to system components and for studying new control strategies.

The CAS model in this paper includes system supply and demand sides. In the first section of the paper, an overview of the CAS configuration is discussed. After that, the second section presents a literature review of previous research on CAS modelling. The third section focuses on the generation side of a system and mathematical models of individual components. The fourth section considers the demand side. Section 5 presents the combined model along with simulations and results. Sections 6 discusses limitations of the model and some conclusions.

2 Overview of compressed air system

A basic CAS, shown in Figure 1, was considered in this study. CAS have often been divided into two major sections labelled the supply side and demand side (Nehler, 2018). The supply side is responsible for production, treatment and storage of compressed air. The demand side includes distribution, pressure regulation end user consumption. Although in some other studies, storage was included in the demand rather than supply side (Lawrence Berkeley National Laboratory, 2003).

Figure 1 Schematic diagram of a typical CAS



The supply side often included a compressor, air cooler, filter, water separator and a storage tank. Air intake into the compressor was filtered to prevent solid particles from entering into the compressor. A compressor, which is typically driven by an electric motor, increases air pressure and consequently its energy content. The compression process also leads to an increase in air temperature, which is undesirable in most applications. A cooler is installed to reduce air temperature. Cooling leads to moisture condensation and water particles are removed with a dryer/water separator. Finally, the compressed and cooled air is stored in a storage tank ready for supply to the demand side.

The demand side of a system includes pressure control valves, pipes and pneumatic tools. Different tools require air supplied at different pressures. Pressure regulators are installed upstream of tools to stabilise network pressure at required levels. Pneumatic tools, which are the main air consumers, transform energy in the compressed air into mechanical work.

3 Background

Attempts to develop CAS models have been reported in the literature but most models only considered either the supply or the demand side.

In Maxwell and Rivera (2003) a basic dynamic model for the supply side of a CAS was developed. The components modelled were a compressor, cooler, piping and tank. To test the model, an arbitrary air demand profile was assumed. The study included two different simulations. The first one studied the effect of varying the pressure control settings of the lead compressor when multiple compressors were operated. Results showed that setting the pressure too low caused operational problems, whereas setting it too high led to energy waste. The second simulation studied energy consumption as storage tank size was varied. Results showed that increasing storage volume decreased energy consumption.

In Anglani et al. (2012), a new tool to simulate generation, treatment and distribution of compressed air was presented. The tool was called Modsca and it was designed in a modular way, making it useful for studying system efficiency, retrofits and for sizing distribution networks. Models for a compressor, air cooler and piping network were suggested. Similar to the study in Maxwell and Rivera (2003), a simulation of energy consumption as size of storage tank changed was performed. Simulation results showed that increasing tank volume decreased energy consumption and stabilised power utilisation profile.

Improvements to the tool presented in Anglani et al. (2012) were suggested in Anglani et al. (2015). The improvements included new modules to model filters, linear and circular distribution networks and different compressor control strategies. Moreover, the tool allowed modelling of the distribution network using an electrical network equivalence approach. Three different simulations were reported. The first two compared linear and circular distribution networks using physics based equations and an electrical equivalent network. Results showed that losses in circular distribution networks were less than linear ones and therefore they were more efficient. In the third simulation, PI and model predictive control (MPC) strategies for the compressor were compared. Results showed that energy savings from MPC compared to PI were low (2.2%) and might not justify the associated complexities of MPC control.

A study reported in Hyvarinen and Lappalainen (1995) considered both the supply and demand side. Mathematical models for air production, distribution and consumption were presented, however the overall consumption of all pneumatic tools was modelled with one lumped parameter equation. The system was made up of a distribution network with multiple pressurised air centres containing compressor(s), storage tank, valves and air consumers. The model was useful for optimal dimensioning of distribution pipes, evaluating system improvement and for general network analysis. Models developed in Hyvarinen and Lappalainen (1995) were used to create a computer program (simulator) to simulate pneumatic networks Hyvarinen and Lappalainen (1996).

In Parkkinen and Lappalainen (1991), a model for the demand side was developed. The model estimated pressurised air consumption in a pneumatic system. Pneumatic tools were classified into two main types: active and passive consumers. Tools with a constant air consumption profile were classified as passive consumers, while tools with periodic and short lived consumption were classified as active consumers. A study reported in Harris et al. (2013) presented a similar model to estimate air consumption. The consumption of air in a linear cylinder during expansion and retraction strokes was calculated separately, while in Parkkinen and Lappalainen (1991) the consumption was assumed equal to the average consumption per double stroke (one expanding stroke and one retracting stroke). The model in Harris et al. (2013) was validated by comparing its results to experiments and the error margin obtained was reported as acceptable.

The research described in this paper combined models based on those in previous papers to build a coupled supply-demand model. Unlike the study in Hyvarinen and Lappalainen (1995) which modelled air demand of all consumers with one lumped parameter equation, the model in this paper considered each air consuming tool individually. Moreover, to account for dynamic variation in air consumption, the approach presented in Harris et al. (2013) to estimate air flow in a pneumatic linear cylinder was followed. Filters and losses in piping were ignored.

4 Modelling individual components

4.1 Supply side

4.1.1 Compressor

Compressors increase the pressure of a fluid or a gas. Depending on their mode of operation, compressors are categorised into one of two broad types: Positive displacement compressors and dynamic compressors (Lawrence Berkeley National Laboratory, 2003). The pressure of air is increased via a compressor in one of two ways: either by decreasing the volume enclosing the gas (positive displacement compressors) or by increasing the number of air molecules within a given space (dynamic compressors). Even though these types of compressors differed significantly in their build and mode of compression, they both performed three common tasks (Kent, 1974):

- Suction: Allowing air into the compressor.
- Compression: Increasing pressure to discharge pressure.
- Discharge: Releasing compressed air into the discharge line.

Assuming air behaved like an ideal gas, the work required (W_{comp}) to compress a volume (V_i) of air from air inlet pressure (P_i) to discharge pressure (P_o) was calculated using Equation (1):

$$W_{com} = P_i \times V_i \times \frac{n}{n-1} \times \left[\left(\frac{P_o}{P_i} \right)^{\frac{n-1}{n}} - 1 \right] \tag{1}$$

where (n) is the polytropic compression exponent.

The process was assumed to be isentropic and therefore n=1.4. To calculate the power, volume flow rate per unit time was used instead of volume. To estimate the electric power supplied to the compressor (W_{sup}), Equation (2) was used.

$$W_{\text{sup}} = \frac{W_{com}}{\eta_{ds} \eta_c} \tag{2}$$

where (η_{ds}) and (η_c) represent the efficiency of the drive system and of the compressor respectively. In this study, both efficiencies were assumed to be constant and equal to 90% and 80% respectively. In reality, the compressor efficiency would vary with discharge pressure, however, for simplicity, compressor efficiency was assumed constant.

Another important factor affecting compressor performance was the assigned compressor control. In this study, a load/unload control was assumed. In load/unload control, the compressed air flow was set to zero when maximum pressure was reached. After that, the machine switched to unload mode, where the compressor operated at part load even though it was not delivering any compressed air. After a period of time unloading, if the pressure in the system remained above the minimum allowable limit, the compressor switched off. In this control mode, when the compressor was on, it operated at its full rated capacity.

4.1.2 Air cooler

The mechanical compression of air causes an increase in its temperature, often reaching an outlet temperature in the range of 70–200 ^{o}C. Coolers are typically installed after the final stage of compression to reduce air temperature. Ignoring water vapour in the air, the temperature of air discharged (T_2) from the compressor was obtained using equation (3):

$$T_2 = T_{in}\left(\frac{P_o}{P_{in}}\right)^{\frac{n-1}{n}} \tag{3}$$

where T_{in} and P_{in} are the temperature and pressure of air at compressor inlet.

Heat transfer between the hot air in the heat exchanger and cooling air in the surrounding was estimated using the effectiveness-NTU method. Assuming a cross flow heat exchanger with a constant effectiveness (ε), the temperature of air leaving the cooler (T_3) was obtained using equation (4) (Bergman et al., 2011):

$$T_3 = \varepsilon\left(T_{amb} - T_2\right) + T_2 \tag{4}$$

Ambient air was assumed to be the cooling fluid. In equation (4), it was assumed that the mass flow rate of cooling fluid was bigger than mass flow rate of compressed air.

4.1.3 Storage tank

The purpose of a storage tank in a compressed air system is to store compressed air for when it is needed. The storage tank pressure depends on the mass of air it stores, its temperature and the overall tank volume. The change of mass in the storage tank was obtained by assuming the tank content to be a control volume and applying a mass balance, leading to equation (5):

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_{out} \tag{5}$$

where m_{in} and m_{out} are the mass flow rate of air entering and exiting the tank, respectively. The mass of air entering the tank was obtained from the compressor capacity, and it was assumed constant while the compressor was running, or zero when the compressor was not running or unloaded. On the other hands, the mass of air leaving the storage tank depended on the demand of air by end user equipment. Calculating mass of air leaving the tank is discussed later in modelling demand side of the system.

The mass of air in the tank at a specific time instant was obtained using equation (6)

$$m(t) = \int_o^t (\dot{m}_{in} - \dot{m}_{out})\, dt + m_0 \tag{6}$$

where m_0 is the mass of air in the tank at time t=0.

Assuming air to behave as an ideal gas, and that the temperature of air in the tank was equal to temperature of air leaving the cooler (T_3), the pressure of air in the tank (P_{tank}) of volume (V_{tank}) was obtained with equation (7):

$$P_{tank}(t) = \frac{m(t) \times R \times T_3}{V_{tank}} \tag{7}$$

In equation (7), R is the specific gas constant for air.

4.2 Demand side

Energy consumption of a compressed air system is highly influenced by end users compressed air consumption. Attempts to model air flow through pneumatic tools were reported in the literature Parkkinen and Lappalainen (1991), Harris et al. (2013) and Beater (2007).

Understanding the flow characteristics of pneumatic components was important to evaluate their air consumption. In Beater (2007), what is known as the ISO 6538 flow model was recommended for estimating mass flow rate through all pneumatic components. The ISO 6538 model is given by equations (8) and (9).

$$\dot{m} = P_1 C \rho_o \sqrt{\frac{T_o}{T_1}} \sqrt{1 - \left(\frac{\frac{P_2}{P_1} - b}{1 - b}\right)^2} \qquad \frac{P_2}{P_1} > b \tag{8}$$

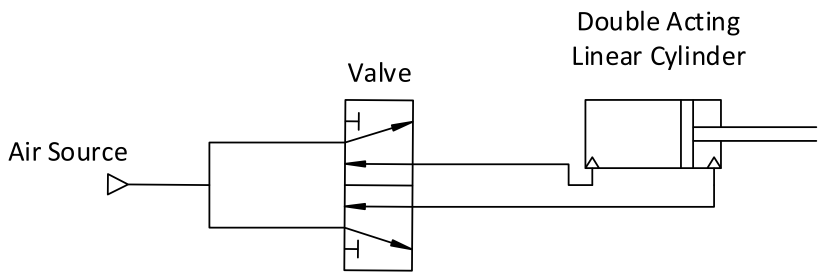
$$\dot{m} = P_1 C \rho_o \sqrt{\frac{T_o}{T_1}} \qquad \frac{P_2}{P_1} \le b \tag{9}$$

where ρ_0 is the air density at atmospheric pressure and the subscripts 1 and 2 indicate upstream and downstream respectively. The parameters 'C' and 'b' are the sonic conductance and critical pressure ratio, respectively. Their value depends on the particular design of the component and typically they were determined experimentally or given in a manufacturer data sheet (Beater, 2007).

Equations (8) and (9) could be directly used to estimate flow rate through tools with constant downstream pressure, such as open pipes, blowers and nozzles. However, for tools with variations in downstream pressure, such as linear cylinders, determining the instantaneous mass flow rate required modelling pressure dynamics, which in turn required modelling forces acting on cylinder bores (Harris et al., 2013; 2012).

A simplified approach for modelling mass flow rate through linear cylinders was suggested in Parkkinen and Lappalainen (1991). The suggested approach calculated the average mass flow rate per unit time by considering the mass of air required to fill the cylinder bore and then multiplying it by the number of cycles per unit of time.

Figure 2 Schematic representation of double acting pneumatic cylinder



A schematic representation of a double acting linear pneumatic cylinder is shown in Figure 2. The motion of the rod is controlled by a switching valve that inflates and deflates the cylinder chamber. Single acting cylinders have a similar build; however, they are equipped with a spring that returns the piston to its initial position after an extension. Single acting linear actuators only consume air on extension stroke, while for double acting, air is consumed on both strokes. Based on the approach suggested in Parkkinen and Lappalainen (1991), equations (10) and (11) were used to estimate mass flow rate for a single acting and double acting linear cylinders, respectively.

$$\dot{m}_{sa} = a_1 (s\frac{\pi D^2}{4} + l\frac{\pi d_t^2}{4})\frac{P_1}{P_2}\rho_0 \tag{10}$$

$$\dot{m}_{da} = a_2 (s\frac{\pi D^2}{4} + s(\frac{\pi D^2}{4} - \frac{\pi d_r^2}{4} + l\frac{\pi d_t^2}{4})\frac{P_1}{P_2}\rho_0 \tag{11}$$

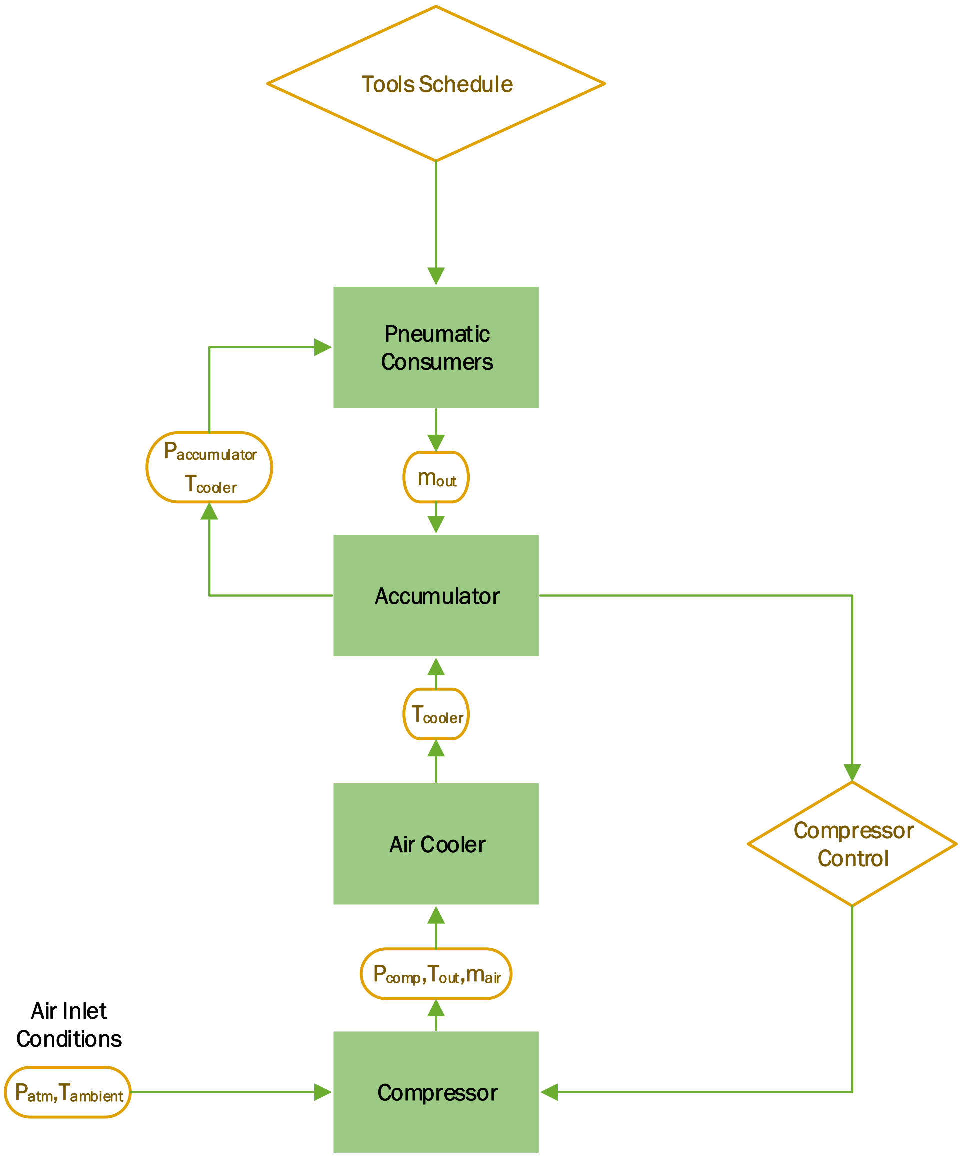
where m_{sa} is the mass flow rate for a single acting cylinder, m_{da} is mass flow rate for a double acting cylinder, s is the stroke length, D is bore diameter, l is tubing length, dt is tubing length, dr is rod diameter, a1 is the number of strokes per unit time, a_2 is the number of double strokes per unit time, P_1 is the upstream pressure and P_2 is the downstream pressure.

5 Model structure

The supply and demand side models presented in the previous sections were implemented in MATLAB. The model was made of separate functions that estimated the required variables, such as compressor power consumption, temperature of air leaving the heat exchanger and properties of air in the tank. Equation (5), which is a first order differential equation, was solved numerically using Euler method. The remaining equations were algebraic and their solution was straightforward. A flowchart showing the structure and logic of the model are shown in Figure 3 and further explained in the next paragraph.

To run the model, a tool schedule defining the periods of operation for each pneumatic tool was defined and supplied as initial input to the model. After that, compressed air consumed (m_{out}) by the pneumatic tools was estimated based on the defined tools schedule. Temperature of air at pneumatic consumer outlets was assumed equal to temperature exiting the air cooler. Compressor control was determined based on air pressure in the tank. Compressor power consumption (P_{comp}), mass and temperature of air supplied by the compressor (m_{air} and T_{out}) were calculated and their value depended on air pressure and temperature at compressor inlet. Temperature of air at air cooler inlet was assumed equal to temperature of air at compressor outlet. Finally, temperature of air in the tank was assumed equal to temperature of air leaving the air cooler.

Figure 3 Model structure and flowchart (see online version for colours)



6 Simulation and results

A tool activation profile was assumed, as shown in Figure 4, and was used to test the model. A value of 0 in the activation profile indicated that the tool was inactive, while a value of 1 indicated the tool was active. A system with one compressor, cooler, storage tank, a double acting linear cylinder, a single acting linear cylinder and a blower was modelled. A schematic diagram of the system was shown in Figure 2.

Figure 4 Schedule of activation for the three pneumatic tools

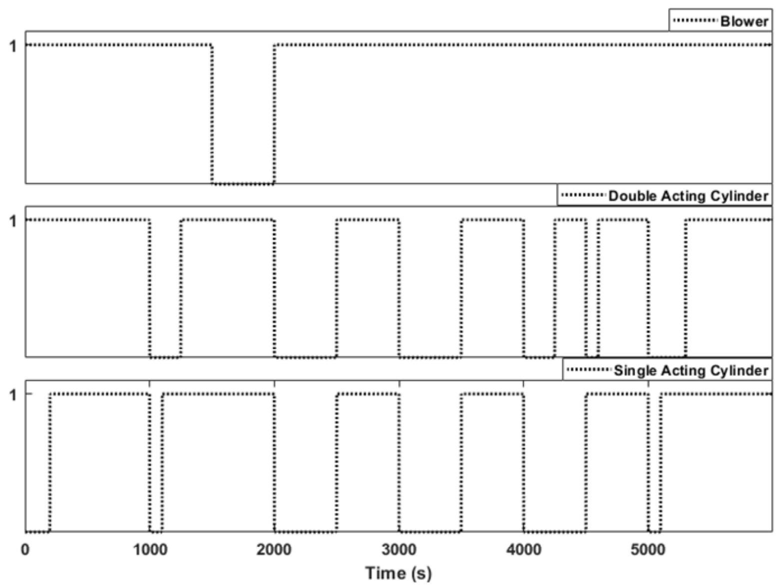
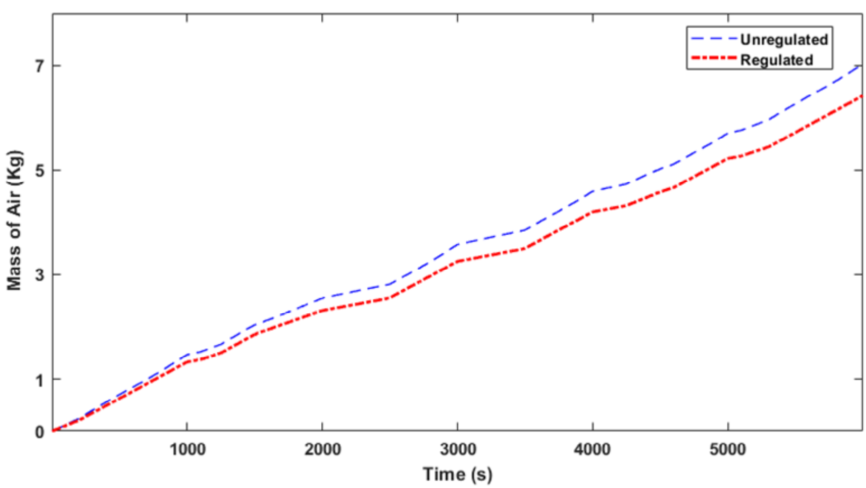


Figure 5 Total mass of air consumed (all tools) for regulated and unregulated air supply (see online version for colours)



The system performance with compressed air consumption was simulated. The role of pressure regulating valves in reducing pneumatic tools air consumption and consequently energy consumption by the compressor was studied. In addition to that, the impact of the tank volume on system performance was analysed. The simulation parameters are summarised in Table 1.

6.1 Pressure regulation

Different tools operate at different pressure levels. Normally, several tools are fed by a single tank whose pressure will vary depending on the compressor control strategy and air consumption profile. To stabilise pressure of air reaching a tool, a pressure regulating valve is normally installed upstream of the tool. In this section, air and energy consumption of a system with and without pressure regulation was evaluated for the same tool activation schedule.

The total mass consumption of the three pneumatic tools, the tank pressure and the total compressor energy consumption are shown in Figures 5, 6 and 7 respectively. Figure 5 compared unregulated (blue curve) and regulated (red curve) air consumption, and results indicate that unregulated system consumed more compressed air over the course of the simulation. This result was expected since equations (8), (9), (10) and (11) indicated that air consumption was proportional to upstream pressure.

The decrease in air consumption was reflected on the air pressure in the tank and on compressor energy consumption. Figure 6 shows the tanks pressure for the unregulated and regulated cases in blue and red respectively. Due to the decreased consumption for the regulated case, it took longer for the tank pressure to decrease to the lowest allowable pressure limit. Over a long period of operation, and assuming identical schedules, the compressor would need to switch on less often for the regulated case compared to the unregulated, leading to some energy savings.

The energy consumption of the system is shown in Figure 7, where cumulative energy consumption for regulated and unregulated cases are shown in blue and red respectively. Over the course of the simulation, system with unregulated pressure consumes more energy than the system with regulated pressure.

Figure 6 Air pressure inside tank for regulated and unregulated air supply (see online version for colours)

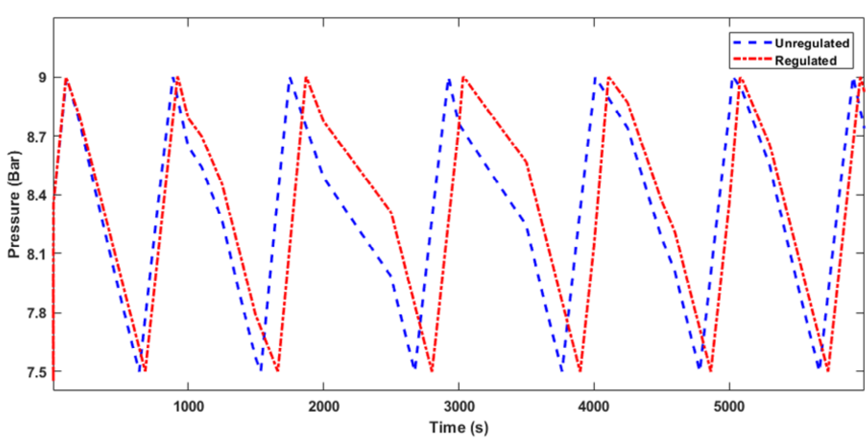
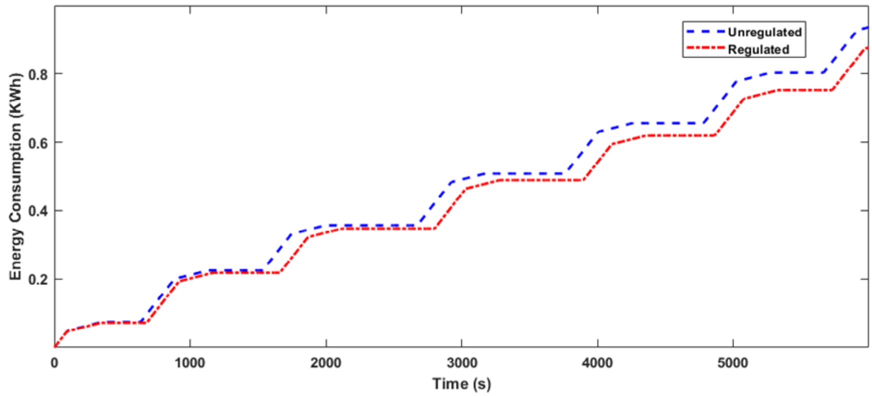


Figure 7 Cumulative compressor energy consumption for regulated and unregulated air supply (see online version for colours)

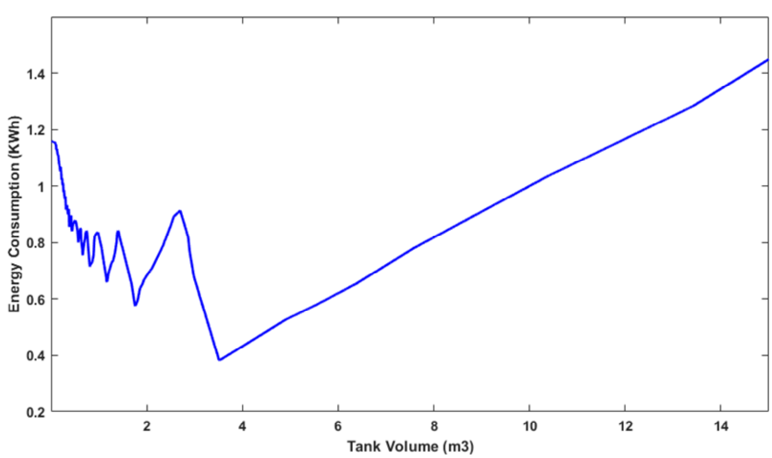


6.2 *Storage tank volume*

The size of the storage tank was usually determined from the compressor capacity and air consumption profile. Several other factors, such as number of compressors and type of drive system impact storage tank selection. The system performance was studied as tank volume changed for the assumed tool operation schedule and compressor capacity.

Figure 8 shows total energy consumption as a function of tank volume. The general shape of the plot indicates that too small or too big tank volume would lead to higher energy consumption. For the air consumption profile used in this simulation, the optimal tank volume was around 3.5 m3. A smaller tank would consume more energy since it would require the compressor to constantly be on or unloading. A large tank would require the compressor to be on for long periods of time to reach the required pressure levels. The optimal tank size would provide a balanced performance and therefore a reduced energy consumption.

Figure 8 Energy consumption as tank volume was varied (see online version for colours)



7 Conclusions and future work

A new CAS model that coupled supply and demand sides of the system was considered. The model was used to study compressed air consumption by end user tools in addition to compressor energy consumption to generate air pressure. Two simulations to study pressure regulation and tank volume were performed.

Simulations showed that pressure regulation plays an important role in reducing energy consumption and improving pneumatic tools performance. Moreover, it was concluded that that too large or too small a tank volume leads to excessive energy consumption. An adequate tank volume reduced energy consumption.

Future work will address model validation, variation in compressor efficiency as discharge pressure changes, air leakage in system and pressure drop due to compressed air treatment (cooling, filtering and drying) and friction in piping. Moreover, the possibility of using the model to develop intelligent systems to save energy in CAS will be evaluated (Thabet et al., 2020; Sanders et al. 2018).

Table 1 Simulation parameters

3 variable	*Description*	*Value (unit)*
V_i	Compressor flow capacity	0.0042 (m^3/s)
P_i	Air inlet pressure	101,325 (Pa)
P_o	Compressor discharge pressure	900,000 (Pa)
η_{ds}	Drive system efficiency	90 (%)
η_c	Compressor efficiency	80 (%)
n	Polytropic compression exponent	1.4
T_{amb}	Ambient air temperature	293 (K)
R	Air gas constant	287 (J/kg•K)
ρ_0	Air density at atmospheric pressure	1.2754 (Kg/m^3)
E	Heat exchanger effectiveness	0.95
S	Stroke length	0.05 (m)
D	Bore diameter	0.025 (m)
d_t	Tubing diameter	0.006 (m)
d_r	Rod diameter	0.01 (m)
l	Tubing length	0.48 (m)
C	Blower sonic conductance	6×10^{-10} (m^3/s.Pa)
b	Critical pressure ratio	0.4
a_1	Single stroke frequency	1 (stroke/second)
a_2	Double stroke frequency	1 (double strokes/second)

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