# Energy Efficient Adaptive Control of Pneumatic Drives with Switching Valves

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## Abstract

With an increasing interest in energy efficiency in automation processes, pneumatic drive applications are often compared to their complementary technology of electrical drives. Motion tasks performed by pneumatic cylinders are said to be energy inefficient due to the operational mode of throttling the exhaust air at the outlet of the cylinder. A novel, model-based operational strategy is presented in order to improve the energy efficiency of the overall pneumatic drive application. The advantage of huge air savings for the optimized system is accompanied by a reduction of the system's stiffness and robustness. A thorough system analysis shows high sensitivities towards parameter uncertainties and the operational strategy is proposed, which adjusts the dynamic model to the real system dynamics. Thus, maximum robustness of the system is satisfied. The result is a stable pneumatic drive systems.

KEYWORDS: Energy efficiency, pneumatic cylinder, optimization, adaptive control

## 1. Introduction

In view of climate changes and increasing energy costs, the improvement of energy efficiency in automation processes becomes more and more important. In the presented paper, the energy efficiency of a pneumatic positioning drive is improved by modifying the common control approach. Energy efficiency of servo pneumatic



**Figure 1**: Pneumatic drive systems. *Left*: standard pneumatic system with a 4/2 directional control valve and flow restrictors, *right*: Valve structure with 5x2-way directional control valves used for optimization.

applications is widely discussed in the literature, resulting in numerous control concepts. However, most industrial pneumatic applications such as shown in **Figure 1** are standard pneumatic applications with switching valves in open-loop control instead of servo valves. Concepts to improve energy efficiency of pneumatic applications with these valves are rare. The improvement of energy efficiency for drive applications with switching valves requires the development of new valve circuits. In /1/ energy efficiency is increased by closing the supply valve and using the expansion energy of compressed air to lift a mass via an inverted slider crank mechanism. To realize several switching positions, a complex 5/9 directional control valve is developed in /2/. The valve structure shown in Figure 1 on the right is used in /3/, for which an optimal control strategy based on a genetic algorithm was developed. This concept was improved by using another optimization algorithm; results are given in /4/. The presented paper pursues the results of /4/ but focuses on adaptive control approaches.

A typical system setup of standard pneumatic cylinders is shown in Figure 1 on the left. It contains a 4/2 directional control valve and two flow restrictors restricting the exhaust air flow and hence the speed of the cylinder. However, the configuration provides no options for saving air: both chambers are connected directly by the valve. Moreover, the flow restrictors cause high pressure levels in the chambers and therefore high energy consumption. At the stroke end, the pressure of the driving chamber is typically filled up to supply pressure level. A high pressure level at the stroke end is usually not necessary. It is avoided by a model-based control technique which reduces the chamber pressure at stroke end by closing the chamber and using the expansion energy of the compressed air. Therefore, the two flow restrictors are neglected in this

work and the 4/2-valve is substituted by a valve structure containing five 2/2 on/off switching valves (arranged in a bridge structure shown in Figure 1 on the right). With this configuration, both chambers are decoupled such that each chamber of the cylinder can be separately and independently vented, exhausted or closed. With those five valves, enough degrees of freedom are available for optimization.

The outline of the paper is a follows: after a detailed description of the system model in section 2, the optimization problem is stated in section 3. A short overview of the optimization results is given. Section 4 treats the adaption of an optimized control. The results of the adaptive control are presented and discussed in section 5 with a final conclusion in section 6.

#### 2. Modeling

The cylinder model from Eq. (1) is a common model used for pneumatic systems as given in /5/ and /6/. It is described by a second-order motion dynamics and two differential equations for the pressure dynamic in each cylinder chamber. In the equation of motion, the friction force term  $F_r$  considers Coulomb and viscous friction:

$$\dot{x}_{1} = x_{2} \qquad \text{position} \\ \dot{x}_{2} = \frac{1}{m} (A_{k1}x_{3} - A_{k2}x_{4} - A_{r}p_{0} - F_{r}(x_{2})) \quad \text{velocity} \\ \dot{x}_{3} \frac{n}{A_{k1}(x_{1} + l_{r1})} (RT_{0}\dot{m}_{a} - A_{k1}x_{3}x_{2}) \qquad \text{pressure chamber 'a'} \\ \dot{x}_{4} \frac{n}{A_{k2}(l_{z} - x_{1} + l_{r2})} (RT_{0}\dot{m}_{b} + A_{k2}x_{4}x_{2}) \quad \text{pressure chamber 'b'}$$

$$(1)$$

Later on, this model is used in vector notation  $\dot{x} = f(x, u)$ . Concerning the valve structure from Figure1 (right), the mass flows  $\dot{m}_a$ ,  $\dot{m}_b$  entering the chambers 'a' and 'b' are calculated as the sum of the mass flows through each valve *i* subject to the control inputs *u*:

$$\dot{m}_{a}(\boldsymbol{u}) = \dot{m}_{1}(u_{1}) + \dot{m}_{3}(u_{3}) - \dot{m}_{5}(u_{5}), \quad \dot{m}_{b}(\boldsymbol{u}) = \dot{m}_{2}(u_{2}) + \dot{m}_{4}(u_{4}) + \dot{m}_{5}(u_{5})$$
<sup>(2)</sup>

The mass flow rate of each value is calculated based on the C/b-value method considering choked and unchoked flow conditions /7/ neglecting the dependency on the temperature as in /5/ and /6/. The fifth value is used as a short cut between both chambers to reuse compressed air.

#### 3. Offline Optimization

In this section, the optimization strategy used for an energy optimal motion is presented.

## 3.1. Optimization problem

The application, for which an energy optimal control strategy is developed, is a common movement task of a mass in horizontal orientation. The task is described as a point-to-point movement of the mass illustrated in **Figure 2**.



Figure 2: Illustration of the considered Application.

Such a positioning task is formulated as a boundary value problem with well-known boundary values at the beginning  $(t = t_0)$  and end  $(t = t_f)$  of all states. The optimal control sequence of the values is wanted, such that the air consumption is minimized, and the boundary values are fulfilled. This can be stated by the following optimization problem:

$$u^* = \min_{u \in \{0,1\}^5} J(x, u)$$
  
such that  $\dot{x} = f(x, u)$  (3)  
and  $x(t_0) = x_0$  and  $x(t_f) = x_f$ 

with  $u^*$  as the optimal control input. The aim is the minimization of air consumption, which is expressed by the integral of the mass flows  $\dot{m}_1$  and  $\dot{m}_2$  supplied by the source:

$$J(\mathbf{x}, \mathbf{u}) = \frac{1}{\rho_0} \int_{t_0}^{t_f} \dot{m}_1(\mathbf{x}, \mathbf{u}) + \dot{m}_2(\mathbf{x}, \mathbf{u}) dt$$
(4)

## 3.2. Optimal control

The optimization problem from Eq. (3) belongs to the class of Mixed-Integer-Nonlinear-Programming problems (MINLP) due to the binary values for the control inputs and the nonlinear systems dynamics. Such an optimization problem is hard to solve. Therefore it is transformed into an easier setup by relaxing the integrality requirements on the control inputs ( $\boldsymbol{u} \in \{0,1\}^5$ ) by allowing the inputs to be out of the interval between 0..1 ( $\boldsymbol{u} \in [0,1]^5$ ). A collocation method is used to solve the relaxed optimization problem (3); more details are given in /4/. The focus of this work is on the adaptive control presented in section 4. The adaption however bases on the optimization results which are presented in the following. The solution of the relaxed optimization problem is shown in **Figure 3** during one cylinder stroke.





Even though the optimization algorithm was allowed to choose values between 0..1, the input sequence is close to binary values. The boundary values used for this optimization are defined to:

$$x_{10} = 0 [m] \quad x_{20} = 0 [m/s] \quad x_{30} = 10^5 [Pa] \quad x_{40} = 10^5 [Pa] x_{1f} = l_z [m] \quad x_{2f} = 0 [m/s] \quad x_{3f} = 10^5 [Pa] \quad x_{4f} = 10^5 [Pa]$$
(5)

The pressure boundary conditions  $(x_{30}, x_{40}, x_{3f}, x_{4f})$  are chosen to be at ambient pressure level  $(10^5 Pa)$ . The optimization results show that the supply is cut off during motion of the piston, such that no more air is consumed. But the expansion energy of the compressed air inside the chamber is used for the rest of the motion (see t = 0.05s, where valve1 changes its state from 1 to 0). The use of the expansion energy results in a reduced pressure level at stroke end. This implies for the backward stroke a reduced force indicating a faster movement of the piston as in the common case. Such a control sequence was first considered by /1/ for vertical applications lifting a mass via an inverted slider crank mechanism. The reduced pressure level at stroke

end may be only used in applications which don't need a high force at stroke end. This assumption is valid for a lot of applications focusing on the movement of a mass. The optimal solution shown in Figure 3 can hardly be used in real applications because of providing no force at stroke end. In real applications, a minimum force at stroke end is needed guaranteeing a stable position of the mass. The example given here shows the potential in using compressed air more efficient and can be considered as a theoretical limit. An adjustment of the pressure boundary conditions results in a modified trajectory with a defined force at stroke end: the lower the pressure level at stroke end, the higher the air savings. The air consumption and air savings in dependence on the pressure level at stroke end is illustrated in **Figure 4**.



**Figure 4:** Air consumption and air savings in dependence on the pressure level at stroke end of the optimized profile.

A comparison of this optimized control to standard pneumatic control is given in /4/. It is shown that with the optimized control 85% of compressed air can be saved. Besides high air savings, the piston reaches stroke end with zero velocity, whereas in standard control the final speed is close to 1.5m/s. This kinetic energy has to be absorbed by a pneumatic or hydraulic damper. Applying the optimized control an air consumption of only 0.071sl are needed for the movement task. The final speed is zero such that a damper can be omitted. The optimized profile shows huge potential in saving compressed air, but some difficulties still arise when applying it to the real system. An offline optimization requires the precise knowledge of all the system parameters such that the offline generated optimal control sequence results in the same behavior on the real system as in optimization. A small deviation can cause different pressure levels at stroke end, such that the initial pressure of the subsequent movement differs from the



Figure 5: Adaptive control scheme.

Boundary conditions assumed for optimization. This influences the system dynamics and can cause the piston to crash into end position at stroke end. This would decrease the lifetime of the cylinder very quickly. But it may also happen that the piston does not reach stroke end. To overcome these problems, an adaptive control as described in section 4 is suggested.

## 4. Adaptive control

The optimized control strategy from Section 3 is applied repeatedly in the real system. An adaptive control is needed to avoid failures as described above. The following difficulties come up: varying conditions/parameters (friction force, supply pressure), unknown system parameters and control in open-loop, i.e. the valves are controlled in an open loop way without feedback. Systems without feedback are sensitive to disturbances (e.g. external forces), but also to unknown or varying system parameters. Furthermore, a decreased pressure level at stroke end results in a decreased stiffness of the piston and an increased sensitivity. The boundary values of the position (stroke end) and velocity (zero velocity) are very hard to reach for the real application. Hence, very small deviations of system parameters cause the offline generated control to fail on the real system (e.g. causing a high speed at stroke end – influence on lifetime of cylinder). To capture these difficulties an adaptive control technique is used as shown schematically in Figure 5. This adaptive control scheme is based on both parameter identification of the dynamic system and optimization of the control sequence. Both blocks are implemented as a mathematical optimization and executed whenever the piston reaches stroke end. The use of a potentiometer for measuring continuously the position of the mass increases the acquisition costs of pneumatic systems tremendously. Another possibility is to use proximity switches instead of a continuous measurement. Two proximity switches are used in any pneumatic application to detect the stroke end. To use them as feedback five proximity sensors are distributed along the stroke of the cylinder. These signals are finally used for parameter identification.

## 4.1. Online parameter identification

The parameter identification algorithm is used to adjust the model to the real system behavior by using measurements. By this, it is possible to identify unknown or varying system parameters such as the friction forces or supply pressure. The identified system parameters are used to update the dynamic model. The main challenge for parameter identification is the sensor technology for measurements. On the one hand the sensors should be precise to get a good estimate of unknown system parameters, on the other hand are the costs for sensor technology. As a compromise, five proximity switches are distributed over the whole stroke of the cylinder yielding five digital time measurements  $t_i^n$  of predefined positions  $y_i^n$ . With those five measurements the cost function is defined considering the squared deviation of the simulation  $\hat{y}_i$  and the measurements  $y_i^n$ :

$$J_{\theta}(\boldsymbol{\theta}) = \sum_{i=1}^{5} \left( \hat{y}_i(t_i^n, \boldsymbol{\theta}) - y_i^n \right)^2$$
(6)

The simulation  $\hat{y}_i(t_i^n, \theta)$  is based on the actually chosen parameters  $\theta$  and evaluated at the measured time of the proximity switches  $t_i^n$ . The parameters  $\theta$  are adjusted such that the simulated model behavior matches best the measurements. In this way, several simulations are done during motion testing serveral parameter combinations.

## 4.2. Online control optimization

Based on the updated model, a new optimal control is calculated by an optimization of the control sequence. This optimization has to be done online, i.e. while the system is running: Hence, an optimization of the full problem from Eq. (3) is not realizable. Again, the optimization problem (3) is rewritten: Concerning the control sequence given in Figure 3, the time based control signals of all valves are split up into different intervals such that the control signals are constant within each interval. The control sequence can then be described by a matrix V which contains the valve positions  $v_i$  for each interval and a vector  $t_s$  which contains the switching times  $t_i$  indicating the transition from  $v_i$  to  $v_{i+1}$ :

In comparison to the offline optimization problem from Eq. (3), the online optimization does not consider the aspect of energy efficiency. This aspect is reflected in the valve position matrix from Eq. (7) and is a result of the full optimization problem stated in Eq. (3) considering the energy efficiency. The objective of the online control optimization is to fulfill the boundary conditions by adjusting the valve switching times  $t_s$ . The control optimization is done via simulations, testing different valve switching times such that the boundary values hold. Therefore, the new cost function reflects the deviation of the boundary values  $y_i^{ref}$  which is penalized by the squared deviation:

$$J_{u}(\boldsymbol{t}_{s}) = \sum_{i=1}^{4} \left( \hat{y}_{i}(T_{ref}, \boldsymbol{t}_{s}) - y_{i}^{ref} \right)^{2}$$
(8)

#### 4.3. Optimization strategy

Both – the parameter identification and the optimization of the control input sequence - are optimization based approaches. Such online optimizations may be time consuming and suffer from a high computational effort. Furthermore, an adequate optimization routine has to be implemented on the real time system. Due to these reasons, an optimization strategy is chosen which is based on an evolutionary strategy /8/. The advantage of an evolutionary strategy is in its gradient free optimization structure and therefore needs less computational effort. On the other side, evolutionary strategies are known to converge slowly, and need a lot of evaluations of the cost function. Due to runtime restrictions, the optimization has to be ready whenever the piston reaches stroke end. To attain this, the optimization is aborted and restarted with the best fit of the last optimization and the new measurements, such that the convergence of the optimization is distributed over many cycles of the cylinder.

#### 5. Results

The adaptive control is applied to a real test setup (cylinder with a diameter of 32mm, 500mm stroke). It consists of a linear position measurement system for validation, five proximity switches for system identification, two pressure sensors and a flow sensor. The parameter identification routine is set-up to estimate both friction parameters - the viscous friction coefficient and the Coulomb friction. The evolution of the system



**Figure 6:** Variation of the switching times (left), estimated friction parameters and velocity at stroke end (right) during the adaption phase.

parameters over time is illustrated in **Figure 6**. Besides the estimated parameters which are a direct result of the parameter identification, the switching times and the speed at stroke end are shown as a result of the online control optimization in Figure 6. Even though the switching times vary little, the speed at stroke end is reduced from 0.3 m/s to 0.15 m/s within 300s. This is an indication for a high sensitivity of the system from the valve switching times. The evolution of the adaptive control strategy from the beginning of the adaption and to the end of the adaption is shown in **Figure 7** on the left and right respectively. The successful implementation of the adaption is visible in particular in the speed of the backward stroke to the beginning (3s) and at the end (289s), where the undershoot in the velocity is compensated such that the final speed is reduced successfully. The energy saving aspect is visible in the reduced pressure at



**Figure 7:** Velocity and pressures (*black:* chamber 'a', *gray*: chamber 'b') at the beginning and at the end of the adaption.

stroke end (2.3bar, see arrows in Figure 7). During one cycle the cylinder needs 1.93sl, whereas in standard control 5.78sl are needed, indicating a reduction of compressed air of about 67%.

# 6. Conclusion

As shown in /4/ and with the results from section 3, the optimization of the open loop control improves energy efficiency by reducing the needed amount of compressed air (under the assumption that no high force at stroke is needed). The disadvantages resulting from the optimized control approach are compensated by an adaptive control, which bases on an online parameter identification in combination with an online control optimization. As a result, a stabilizing adaptive control which reduces the velocity at stroke end is presented. It is tested and validated on a real system and reduces the air consumption of 67%. Further effort is put into an acceleration of the convergence of the optimization algorithms. By considering the possible high energy savings the presented control approach is worth to be followed up.

# 7. Acknowledgement

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# 8. Literature

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# 9. Nomenclature

| R              | Ideal gas constant                    | J/kg/K |
|----------------|---------------------------------------|--------|
| T <sub>0</sub> | Temperature at standard conditions    | К      |
| $ ho_0$        | Density of air at standard conditions | kg/m³  |
| x              | Position                              | m      |
| V              | Velocity                              | m/s    |
| p              | Pressure                              | Ра     |
| J              | Cost function                         | -      |
| 'n             | Mass flow                             | kg/s   |
| т              | Mass                                  | kg     |
| $l_z$          | Stroke length                         | m      |
| A              | Area                                  | m²     |
| $l_t$          | Dead length                           | m      |
| sl             | Standard liters                       |        |