ENERGY EFFICIENCY AND PERFORMANCE OF SERVOPNEUMATIC DRIVES FOR SPEED GOVERNORS BASED ON OPERATING POINTS

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ABSTRACT

In this paper, the energy efficiency and dynamic performance of servopneumatic drivers are studied aiming at the correct sizing of the system components. The study is carried out through a non-linear dynamic model of a servopneumatic system. The simulation data are used along with the operating point method in order to understand the effect of loading conditions in the system performance, which is then used as a tool for component sizing. The results show the effects that the piston area has on the energy efficiency and dynamic behavior of the system. Moreover, the dependency of the servovalve size on the piston area is elucidated. A procedure to design an effective combination of piston area and sonic conductance of the servovalve is presented. A practical case where a servopneumatic system is being designed for the speed governor of a hydraulic turbine is described, evidencing the suitability of the proposed method for servopneumatic systems.

Keywords: Servopneumatic systems, sizing method, operating point, turbine speed governors

1. INTRODUCTION

Pneumatic systems are widely used in the automation industries due to its low acquisition and maintenance costs, high reliability and high power/weight ratio, beyond that they provide speedy actuations. Pneumatic systems are being used for point-to-point automation purposes since the 60's [1] and in the 80's the first commercial servopneumatic system has been made available [2]. Since then, the application of servopneumatic systems has increased widely, being applied in the fields of automation, robots and manufacturing. According to [3], pneumatic servosystems are capable to replace electromechanical and hydraulic actuators, which are less efficient and costly [4] and [5].

In the past few decades, many control strategies have been developed for pneumatic servosystems, including linear and nonlinear controls, fuzzy based techniques and state observers with compensators strategies [3]. In the same way the mathematical modelling also had many improvements, for which models for nonlinearities such as the piston friction, end

stroke cushioning and valve mass flow rate have been developed. These models are combined with equations based on the physical principles that govern the system, resulting in a global model capable to predict the behavior of the state variables of the drive with high accuracy [2].

However, in the sense of pneumatic system sizing, there is still the need of consistent method for the selection of the system components. An optimal system design is proposed by [6], even so the method is based on an optimization algorithm, which limits the applicability of the method for software implementing tools. On its review on recent research trends in servo pneumatic systems, [3] describes the systems parameters that have a significant impact in the system dynamic. Still, no guidelines are given in order to select the main components.

In this context, this paper studies the characteristic behavior of a servopneumatic system in order to propose guidelines to size the systems parameters. The inflection point of the velocity profile and the operating point of system are used as key points in the design process of the servovalve and the cylinder diameter.

2. MATHEMATICAL MODEL

A conventional servopneumatic system is presented in **Figure 1**. It is composed by a cylinder, a servovalve (SV), a position transducer, a set of hoses and a controller, which receives the position error and sends the control signal to the SV.

Figure 1: Conventional servopneumatic system

In this study, the components were modelled according to the physical laws that govern the system, including the mass flow rate through the valve, the pressure and temperature inside the cylinder chambers, the motion and friction of the piston. The heat exchange in the actuator is modelled by the Newton's law of cooling and the friction is described by the LuGre Model [7]. These equations result in a non-linear model which is described in [8] and implemented in MATLAB/Simulink® for simulation purposes. The pressure and thermal losses in the hoses were neglected since the PDV is usually connected close to the actuator and their effects on the system dynamics might be included in the dead volume of the cylinder chambers. An experimental validation is performed in order to evaluate the effectiveness of the model and the results are presented in [8].

3. ANALYSIS OF THE SYSTEM DYNAMICS

A servo pneumatic actuator is, in the point of view of control, a system working in closed loop, where the output is the piston position and the input is a reference position of the actuator stroke. As in many control systems, the general response of a servo pneumatic system might be approximate by a second order system, that is

$$\frac{y(s)}{y_{Ref}(s)} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2}. \quad (1)$$

This approach is convenient since the design requirements are easily converted into a second

order system parameter, such as natural frequency and damping ratio. A similar approach is used in [9] and [10], where a design method for hydraulic positioning system is proposed.

3.1. Inflection point of a step response

The design requirements of a positioning system usually are composed by the maximum displacement, settling time, maximum forces and overshoot. Therefore, the design requirements can be converted into natural frequency and damping ratio by the settling time

$$t_s = \frac{6}{\omega_n} \quad \text{for } \zeta = 1 \quad (2)$$

or

$$t_s = \frac{5.7}{\omega_n} \quad \text{for } \zeta = 0.7. \quad (3)$$

In a second order system, the maximum velocity occurs at the inflection point of the velocity profile. At this point, the acceleration is null and the velocity is given by.

$$v_{max} = 0.4668x_{max}\omega_n \quad \text{for } \zeta = 0.7 \quad (4)$$

or

$$v_{max} = 0.3678x_{max}\omega_n \quad \text{for } \zeta = 1 \quad (5)$$

where x_{max} stands for the expected maximum step response [9].

Figure 2 presents an example of a step response of a servo pneumatic system along with the response of second order system with $\omega_n = 1.772 \text{ rad/s}$ and $\zeta = 1$.

Figure 2: Comparison of a simulation model with a second order system

It can be seen that the characteristic second order model is capable to represent accurately the dynamic of the servopneumatic system, demonstrating the applicability to estimate the

maximum velocity through the design requirements.

3.2. System simulation analysis

The effects of the chamber pressures during the displacement of the piston have an important impact on the system dynamics. Consequently, it is important to understand the characteristic behavior of the system in order to size it accordingly. As an example, a sequence of 10 step inputs, each one of 10% of the actuator stroke, was analyzed.

An important aspect of the study is that the closed loop control has a direct impact in the results. Therefore, the closed loop control is made with a simple proportional controller. The gain is calculated in such a way that the maximum error results in 100% of the valve opening and the sonic conductance was adjusted to have a critical damping behavior. The parameters of the system are presented in **Table 1**.

Table 1: Parameters of the simulation

Parameter	Value
Piston diameter D_p	0.032 m
Rod diameter D_r	0.012 m
Area ratio r_A	0.8594
Cylinder stroke L	0.5 m
System mass M	8.6 kg
Load force F_L	470 N
Viscous friction coefficient B	200 N.s/m
Supply pressure p_s	8 bar
Sonic conductance C	$6 \times 10^{-10} \text{ m}^5/\text{N.s}$
Critical pressure ratio b	0.3

Results from simulation of the pneumatic positioning system are presented in Figures 3 to 5.

It can be seen that the maximum velocity is not constant throughout the piston stroke. This fact is mainly related to the amount of air inside the counter pressure chamber, which is maximum in the beginning of the movement and reduces at each step. However, either the pressures and the maximum velocity tends to be constant as the piston reaches its end stroke. Further analysis related to these simulations are presented in Chapter 4.

Figure 3: Reference and piston positions

Figure 4: Piston velocity

Figure 5: Chamber pressures

4. STEADY STATE ANALYSIS

The inflection point of the velocity has important characteristics that might be used for sizing purposes. For this reason, the pressure of both chambers was recorded at the inflection point in order to obtain the pressure ratios at this condition.

The operating point method was introduced by [11] and [12] and it describes the pressures of the system as a function of pressure ratios, for instance, in the extending movement p_A/p_s is a function of p_0/p_B .

As shown in **Figure 6**, the pressure ratios (p_A/p_S and p_0/p_B) at the maximum velocity points result in ten different operating points (red circles). They are analyzed along with the operating point chart, which is composed of a constant pressure curve (*Op* curve) and a constant velocity curve (*Ldg* curve). The intersection of both curves occurs when both pressure and velocity are constant. The curves are described by

$$\frac{p_0}{p_B} = b + \sqrt{\frac{b^2 - 2b + 1 + r_A^2 + (2br_A^2 - r_A^2)\left(\frac{p_S}{p_A}\right)^2 - 2br_A^2\left(\frac{p_S}{p_A}\right)}{(2br_A^2 - r_A^2)\left(\frac{p_S}{p_A}\right)^2 - 2br_A^2\left(\frac{p_S}{p_A}\right)}} \quad (6)$$

and

$$\frac{p_A}{p_S} = \frac{F_T}{p_S A_A} + \left(\frac{p_B}{p_0}\right)\left(\frac{p_0}{p_S}\right)r_A + \frac{p_0}{p_S}(1 - r_A). \quad (7)$$

where Equation (6) corresponds to the *Op* curve and Equation (7) to the *Ldg* curve. The cylinder area ratio is defined as $r_A = A_B/A_A$. The equation derivations for this forward movement as well as for returning movement are presented in [12].

Figure 6: Operating point chart

The highlighted point corresponds to the last maximum velocity peak, whereas the first circle from right to left corresponds to the first maximum velocity peak. It can be seen that during the displacement of the piston along its stroke, the pressure ratios of the maximum velocity points are located exactly on the *Ldg* curve, which is due to the nullity of acceleration at this point. It is also important to notice that the intersection between both curves occurs at the higher value of the maximum velocities and all the remaining points are below to the *Op* curve.

The *Op* curve is dependent of critical pressure ratio (b) and the area ratio (r_A), which are parameters that have a small influence on the dynamic behavior of the system. Usually, even

when these parameters are not known, assuming typical values is enough to obtain the *Op* curve. However, the *Ldg* curve depends on the load force and the piston area, that means, from the different loading ratios ($F_T/p_S A_A$). Therefore, the obtaining of the *Ldg* curve depends on the determination of the chamber A piston area and the supply pressure, which will be discussed in Chapter 5.

5. PISTON DIAMETER DETERMINATION

For the same pneumatic actuation system, the variation of the piston diameter has a significant influence on the dynamic of the system. The main aspects are related to velocity, displacement time and energy efficiency [8].

In order to understand the effects of the piston diameter in the dynamic of a pneumatic servosystem, a sensitivity analysis is performed with 6 different diameters simulated under the same loading conditions, including the viscous friction coefficient and the area ratio. Table 2 presents the parameters of the different cylinder used on the sensitivity analysis. The loading ratio (L_r) is given by:

$$L_r = \frac{F_L + p_S A_A \mu}{p_S A_A} = \frac{F_T}{p_S A_A} \quad (8)$$

where F_L stands for the load force (470 N) and the parcel $p_S A_A \mu$ represents the friction force, where μ is the coefficient that correlates the friction force with the maximum force in the chamber A of the actuator. Due to the low velocities, μ was assumed to be 1%.

Table 2: Cylinder data for the sensitivity analysis

Piston diameter [mm]	Rod diameter [mm]	Loading ratio [1]
30	11.25	0.8411
32	12	0.7405
34	12.75	0.6571
36	13.5	0.5872
38	14.25	0.5280
40	15	0.4775

Figure 7 presents the operating point chart with the results of the 6 simulations from the sensitivity analysis.

It can be seen that each loading ratio resulted in a different *Ldg* curve and the pressure ratios at the maximum velocity points lie on the *Ldg*

curve. Moreover, for lower loading ratios, the pressure ratios no longer reach the Op curve, a similar behavior occurs for point to point applications, as described in [8].

Figure 7: Operating point chart for different loading conditions

The main aspect related to this analysis is the choice of the best loading ratio. To support this decision, **Figure 8** presents the maximum velocity of the last step (Figures 3) and the energy efficiency along with the loading ratio. The energy efficiency is the ratio of the delivered pneumatic energy with the supplied exergy, the calculus procedure is presented in [8].

Figure 8: Maximum velocity and energy efficiency for different loading ratios

Figure 8 shows that as the piston diameter increases, the maximum velocity and the energy efficiency increases up to a limit. At a certain

point, both start to decrease, pointing out the importance of a correct sizing of the actuator. In this case, the best configuration occurs with $Lr \cong 0.63$, which is close to the maximum velocity condition.

In order to estimate this optimal condition for a general system, Equation 9 might be used, resulting on an expression of Velocity/Sonic Conductance (C) ratio as a function of the pressure ratio under the hypothesis of constant speed and pressure [8], as exemplified in **Figure 9**.

Figure 9: Determination of the maximum velocity point

By the Velocity/ C curve, the p_A/p_S value corresponding to the maximum steady state velocity is obtained. This pressure ratio is then applied in Equation 6 in order to obtain the corresponding p_0/p_B ratio. Both pressure ratios are substituted in Equation 7, yielding a loading ratio corresponding to an optimum condition (Lr_{opt}), which is, in this case, 0.5789. As can be seen, this method results in a loading ratio close to the maximum velocity condition shown in **Figure 8**, which has a loading ratio of 0.5847.

Once the optimum loading ratio is defined, the piston area or the supply pressure can be calculated by

$$A_A = \frac{F_L}{p_S(Lr_{opt} - \mu)}. \quad (10)$$

$$\frac{v_{e ss}}{C} = \frac{p_S p_0 \sqrt{1 - \left(\frac{p_A - b}{p_S} \right)^2} \left(\frac{p_A}{p_S} \frac{p_0}{p_S} (1 - r_A) - \mu - \left(\frac{\left(\frac{p_0}{p_S} \right)}{\frac{b}{r_A} + \sqrt{\frac{b^2 - 2b + 1}{r_A^2} + 1} + \left(\frac{p_S}{p_A} \right) \left(-2b + \left(\frac{p_S}{p_A} \right) (2b - 1) \right)} \right) \right)}{\left(\frac{p_A}{p_S} \right) F_L} \quad (9)$$

6. SERVOVALVE DESIGN

The sizing of the servovalve is an important step to guarantee the system dynamics since it controls the airflow rate to the cylinder [3]. The sizing of the proportional valve (or servovalve) might not be done isolate, since the piston diameter also affects the dynamic behavior of the system [8]. For this reason, the continuity equation for the cylinder chambers must be analyzed along with the mass flow rate equation through the valve.

Applying the continuity equation to the chamber A of the actuator (**Figure 1**), under the hypothesis of isothermal process, results in:

$$\frac{dp_A}{dt} = \frac{1}{V_A} \left(q_{mP-A} T_A R - p_A A_A \frac{dx}{dt} \right). \quad (11)$$

The mass flow rate from the constant pressure source to the chamber A is defined by the ISO 6358-1 (2013) standard [13]. Assuming a subsonic flow and a supply temperature equal to the ambient temperature:

$$q_{mP-A} = p_S C \frac{U}{U_n} \rho_0 \sqrt{1 - \left(\frac{p_A - b}{p_S} \right)^2}. \quad (12)$$

Combining Equations 11 and 12, and assuming the chamber A temperature to be equal to the ambient temperature, the sonic conductance becomes

$$C = \frac{\frac{dp_A}{dt} V_A + p_A A_A \frac{dx}{dt}}{p_S \frac{U}{U_n} \rho_0 \sqrt{1 - \left(\frac{p_A - b}{p_S} \right)^2}}. \quad (13)$$

Equation 13 presents the parameters that should be accounted during the sizing process. Considering that the maximum velocity point is a critical condition for the servovalve, Equation 13 might be used with the parameter values of the inflection point of the step response.

During the design process, the maximum velocity is estimated using Equation 4 or 5. The piston area shall be determined using the Equation 10. The ratio U/U_n might be considered as 1, since the control signal at the maximum velocity usually is close to the maximum opening of the servovalve. The critical pressure ratio (b) has its maximum theoretical value equal to 0.528, however, commercial valves always have a lower value, usually around 0.3.

Through simulation analysis, the parcel relative to the derivative of pressure of Equation

13 has been found to be negative at the point of maximum velocity. Besides that, the product of $V_A dp_A/dt$ represents around 15% of $p_A A_A dx/dt$ parcel. Consequently, neglecting the derivative of pressure parcel it results on a higher sonic conductance, which is favorable for the sizing process.

The remaining parameter to be determined in order to calculate the sonic conductance in Equation 13 is the p_A/p_S ratio. Analyzing the Figures 6 and 7, it can be seen that the pressure ratios are not constant throughout the piston stroke and they lie over the Ldg curve. Since the p_A/p_S for the maximum velocity can be calculated by Equation 9, it is suggested to choose an average value between 1 and the pressure ratio at maximum velocity, as it represents an average working condition along the cylinder stroke. The average pressure ratio is given by

$$(p_A/p_S)_{Av} = \frac{1 + (p_A/p_S)_{v_{max}}}{2}. \quad (14)$$

Under the assumptions presented above, the servovalve can be sized according to

$$C = \frac{(p_A/p_S)_{Av} \cdot A_A \cdot v_{max}}{p_0 \sqrt{1 - \left(\frac{(p_A/p_S)_{Av} - b}{1-b} \right)^2}}. \quad (15)$$

7. CASE STUDY

In this chapter, the method presented above is applied for sizing the actuation system of a speed governor of a Francis hydraulic turbine, which is part of a research and development project that aims the replacement of hydraulic servomotors in small hydropower plants. The applicability of pneumatic servosystem for hydraulic speed governor was previously studied by [14] and [15].

7.1. System description

The speed governor of a hydraulic turbine is the system used to control the rotational velocity of the turbine, which should be held constant in order to maintain a constant frequency of the generated electric energy. The control is made by positioning of the guide vanes, increasing or decreasing the amount of water flowing through the turbine.

In this study, a real hydropower plant with net height of 18.5 meters and a Francis turbine with 438 kW of generation capacity is evaluated. The

main requirements for the actuation system design are the speed governor time constant, the minimum opening time, the load force and the total stroke. Other requirements such as minimum closing time and emergency closing are related to an auxiliary actuation system and will not be discussed in this paper. Additional information about this system is presented in [5].

7.2. Design requirements

According to the standard IEC 61362 (2012) [16] the time constant (τ_y) of the speed governor should range between 0.10 to 0.15 seconds. Assuming a criterion of 1% of error, the settling time is 0.5 to 1.25 seconds ($5\tau_y$).

The load force caused by the water flow through the turbine is not constant. [17] presents a load profile where the maximum load occurs around 80-90% of the actuator stroke and the force is zero around 15% of the stroke.

The load profile proposed by [17] has an hysteresis between the opening and closing movement, which is due the water force in the blades. Due to the uncertainty on determining the transition between the opening and closing forces, [5] proposed an adjusted load profile which includes the maximum load force and the null force, as presented in **Figure 10**. This curve will be used for sizing and simulation purposes.

Figure 10: Load profiles in a Francis hydraulic turbine

The mounting of the guide vanes requires a total stroke of 131 mm for the actuation system. A step of 10% of the total stroke can be considered for the dynamic sizing of the actuation system.

7.3. System sizing

The first step to size the actuation system is the definition of the best piston area considering the system loading. In this case, the load is not constant throughout the piston stroke, however, in order to guarantee the dynamic performance required by the IEC 61362 Standard, the system must be sized for the worst-case scenario, which is the maximum load force.

The available pneumatic compressor is capable to work up to 10 bar_{abs}. Due to the high load involved, two actuators working in opposing conditions are used. It is equivalent of a single actuator with the resulting area equal to the sum of area A_A and area A_B of the actuators and the area ratio (r_A) equal to 1. The friction coefficient is estimated to be 1% of the maximum force in chamber A and the maximum load according to **Figure 10** is 26.16 kN. The critical pressure ratio of the servovalve is estimated in 0.3. Based on that, Equation 9 is used to determine the velocity/ C as a function of the p_A/p_S .

Figure 11: Point of maximum velocity

The pressure ratio p_A/p_S at maximum velocity is 0.7956. The corresponding p_0/p_B ratio is obtained by the Equation 6, which yields 0.6223. The resulting loading ratio ($F_T/p_S A_A$) is 0.6309, which is given by Equation 7.

Using Equation 10, the resulting piston area (A_A) is 0.0421 m², which is equivalent to a cylinder with 231 mm of piston diameter.

The closest commercial area is the combination of two cylinders with 160 mm of piston diameter and 40 mm of rod diameter. The total area is 0.038956 m², resulting in a loading ratio of 0.6815, which is higher than the ratio of the maximum velocity (0.6309). However, as it can be seen in **Figure 8**, loading ratios slightly

higher than the maximum velocity results in higher energy efficiency. An acceptable loading ratio must be chosen carefully in order to avoid an undersized system. **Figure 12** presents the Ldg curve of both loading ratios where it can be seen that the increasing of L_r has a small impact at the velocity curve, which decreased just about 2%.

Figure 12: Velocity variation of both loading ratio

The natural frequency of the closed loop positioning system is given by Equation 3, where the settling time is set in 1.25 seconds ($5\tau_y$), in accordance with the IEC 61362 Standard. A small overshoot is acceptable, thus a damping ratio of 0.7 is assumed. Hence, the requested natural frequency is 4.56 rad/s.

The velocity is obtained by Equation 4, where the expected maximum displacement is 10% of the total stroke, which yields 0.0219 m/s.

This velocity is used in Equation 15 to be calculating the sonic conductance of the servovalve. The p_A/p_S of the dashed Ldg curve (**Figure 12**) may be adjusted according to the Equation 14 resulting on 0.8978. Therefore, the result is $C = 14.3629 \times 10^{-9} \text{ m}^5/\text{N.s}$. A commercial model with sonic conductance of $16.5 \times 10^{-9} \text{ m}^5/\text{N.s}$ is chosen for this application. If the same procedure is applied to the dash-dotted line, the resulting sonic conductance is $C = 15.6579 \times 10^{-9} \text{ m}^5/\text{N.s}$.

Simulations were carried out with the designed values of the piston area and the sonic conductance of the servovalve. **Figure 13** presents the results from two different simulations. One is with the exact parameters obtained from the design process and the other is with the commercially available components.

Figure 13: Simulation results of the designed systems.

The simulations are carried out at the most critical point, which is at 85% of the total stroke (maximum load). The proportional controller gain is set to 382, which results in the total opening of the servovalve at the maximum velocity. According to the design process, the system should have a settling time of 1.25 seconds with overshoot. As can be seen, the settling time in both simulations is quite close from the design requirement and no significant overshoot is present in the step response.

It is important to highlight that the control plays an important role in the closed loop results. For this reason, the control gains should be tuned in a more effectively way, and/or an improved controller such as an adaptive PID controller should be used. As an example, the **Figure 14** presents the system using commercial parameter with an increased gain of the proportional controller. The applied control signal to the servovalve is presented in **Figure 15**.

Figure 14: Simulation result with higher proportional gain

Figure 15: Servovalve control signal

As can be seen in **Figure 15**, with an adjusted controller the design requirements are achieved and the control signal sent to the servovalve shows that its input voltage is saturated during less than 10% of the displacement, evidencing that the sonic conductance of the servovalve is adequate for the positioning task.

8. CONCLUSION

In this paper, the characteristic behavior of servopneumatic systems was studied aiming the correct sizing of the system components. The analysis was carried out by a non-linear dynamic simulation model implemented in MATLAB/Simulink® and the results from the inflection point of the position curve were studied with the operating point chart, which describes the system pressures under the hypothesis of constant speed and pressure.

The study showed that the piston size and the supply pressure affect the dynamic behavior and the energy efficiency of the system. An approach based on the operating point method and the position inflection point was proposed in order to design the system achieving the best dynamic and energetic performance as possible.

A case study was presented where the actuation system of a hydropower plant is being designed. The design requirements were converted into natural frequency, damping ratio and load force. An optimal loading ratio was determined for the piston diameter sizing and the combination of piston area with dynamic requirements were used to size the pneumatic servovalve.

The results presented in this paper evidence that the system sizing method is able to

satisfactorily calculate the system parameters in order to attend the design requirements.

The main advantage of the proposed method is the possibility to design an efficient servopneumatic system, either in terms of energy efficiency and dynamic performance, through an analytical process which is based in the fundamental equations of the system. Since it does not require optimization algorithms, the method can be carried out without the need of software or with simple spreadsheets.

It is important to highlight that the method presented in this paper is not concluded yet, since many aspects of the design process are still object of study, such as the non-linear friction forces and different loading conditions. As a final result, a new methodology for pneumatic system sizing is expected.

NOMENCLATURE

ζ	Damping ratio [1]
μ	Friction coefficient [1]
ρ_0	Density at ambient conditions [kg/m ³]
τ_y	Time constant of speed governor [s]
ω_n	Natural frequency [rad/s]
A_A, A_B, A_r	Chamber A, B and rod area [m ²]
B	Viscous friction coefficient [N.s/m]
b	Critical pressure ratio [1]
C	Sonic conductance [m ⁵ /N.s]
D_p	Piston diameter [m]
D_r	Rod diameter [m]
F_L	Load force [N]
F_{max}	Maximum force [N]
F_T	Total force [N]
L	Actuator stroke [m]
L_r	Loading ratio [1]
M	Load mass [kg]
p_A	Chamber A pressure [N/m ²]
$p_{s,0}$	Supply and ambient pressure [N/m ²]
q_m	Mass flow rate [kg/s]
R	Ideal gas constant [J/kg.K]
r_A	Cylinder area ratio [1]
T	Temperature [K]
t	Time [s]
t_s	Settling time [s]
U	Control signal [V]
U_n	Nominal control signal [V]
V_A	Chamber A volume [m ³]
$v_{e ss}$	Extending steady state velocity [m/s]
v_{max}	Maximum velocity [m/s]
x	Piston displacement [m]
x_{max}	Maximum desired step
y, y_{Ref}	Position and reference position [m]

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