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REIMUND NEUGEBAUER *, *JOHANNES QUELLMALZ* *, *RUBEN SCHÖNHERR* *, *HOLGER SCHLEGEL* *

CONTROL OF A SERVO-PNEUMATIC EXPERIMENTAL RIG

The paper presents the control concept for an experimental rig with closed-loop controlled pneumatic axis. The objective is the convenient execution of diverse control technologic experiments using free implementable control structures. Since two actuators can be mechanically linked to one another, one is force controlled to generate defined disturbances. Furthermore, a particular simulation model, which can be integrated in the controllers' user program, is pointed out including non-linear effects. Finally, selected experiments are discussed.

1. The experimental rig

1.1. Task

As an introduction, the purpose of the experimental rig and with it the goal of the task of automation shall be described. The usage is research and education in the field of mechatronic axes and the control of which at the example of closed-loop controlled pneumatic drives. Such systems are used in machines for purposes like testing on mechanical stability [1]. So besides intuitive handling also flexible usability and customization is the focus. For this reason diverse sensors like position, force and pressure sensors are designated. Central element is the controller algorithm of its own. It has to be interchangeable and able to be parameterized and implemented freely – independent from strict structures. Defined interfaces are required, so that the rest of the user program including sequence control, set-point generation, communications and the like can be left untouched of this. For comparison

* *Chemnitz University of Technology, Faculty of Mechanical Engineering, Institute for Machine Tools and Production Processes, Reichenhainer Str. 70, Germany; E-mail: johannes.quellmalz@mb.tu-chemnitz.de*

and evaluation of different controller configurations under varying circumstances diverse motion profiles and load situations are to be generated.

1.2. Mechanical and Pneumatic Setup

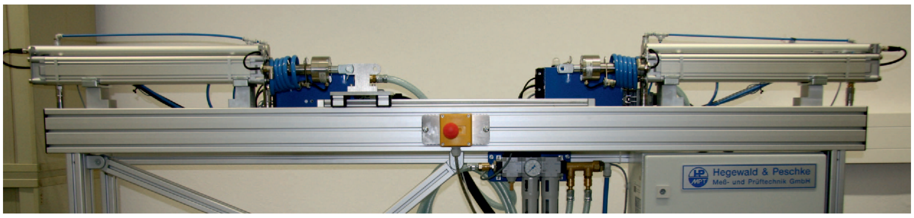

Fig. 1. Servo-pneumatic experimental rig

The experimental rig itself is constructed as shown in Fig 1. It comprises two identical pneumatic cylinders of half a meter, which can be interconnected mechanically via a linearly guided carriage by eyes. The carriage can be charged with additional weights. Absolute position measurement systems with analogue output are used as well as analogue sensors for force and chamber pressures. After the central compressed-air conditioning unit, the cylinders are charged with compressed air via electrically operated 5/2-directional proportional valves, i.e. with internally position controlled valve pistons. The exhaust air of both chambers of one cylinder is discharged via one common damper. In the case of emergency stop, the system is discharged centrally while movement is stopped by means of pressure operated non-return valves mounted directly to the cylinder chambers.

2. Control

2.1. Controller setup

It is chosen to use a decentralized controller setup. The separation is fostered by the fact that the drives can be parameterized and controlled completely independent from one another. The type of control used is non-standard, being developed internally at Chemnitz University of Technology and getting used for research purposes in the field of measurement and control engineering. So, a maximum of flexibility and transparency (down to driver and hardware layers) is guaranteed. The basis is a digital signal processor (DSP) of high performance. Features distinguishing to PLCs are constant cycle time, predestining the control for realization of digital controllers, as well as the high computing power of the DSP allowing for low cycle times.

The controllers are interfaced to each other and to the PC by CAN-Bus. The software running on the PC is used to configure the controllers and compose and parameterize the user programs. Furthermore, variables can be monitored, visualized and logged. With a controller-intern buffer data can be logged with short sample times for a defined period of time. The data can then be analyzed within the PC software.

2.2. Control concept

Besides the possibility of mounting additional weights, advanced load situations shall be enabled through the second pneumatic cylinder (say the load side) being force-controlled an allowing the first cylinder (say the drive side) to be exposed to defined disturbances. Experiments shall be executed from defined initial conditions with reproducible results. The concept focussed on the user program(s), which was realized for this purpose, shall be made clear at the following schema Fig. 2.

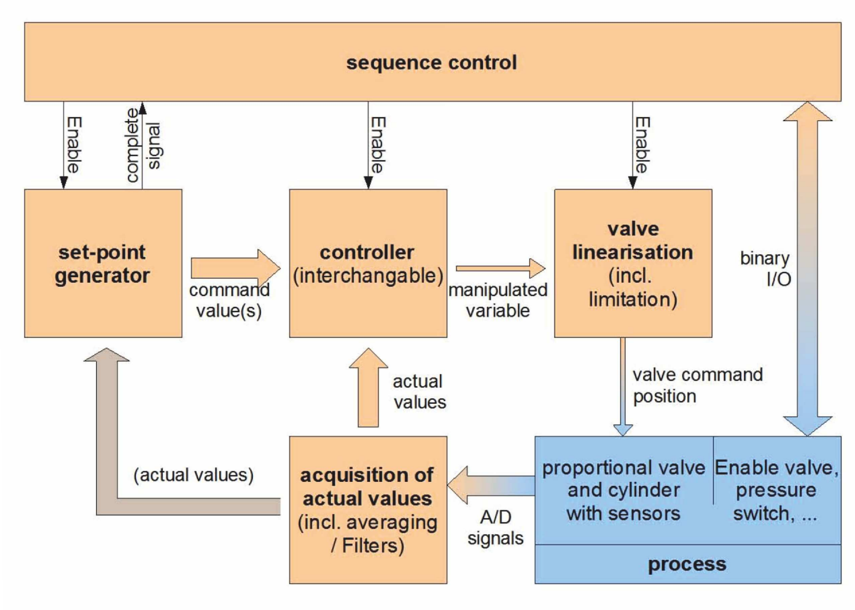

Fig. 2. Overview of the control concept

The structure is equal to a standard closed-loop control plus superordinated sequence control. The branch of the actual values to set-point generation plays a role only for state-dependent load characteristics e.g. emulation of a spring or velocity-dependent loads. The sequence control works according to a state machine having several state chains.

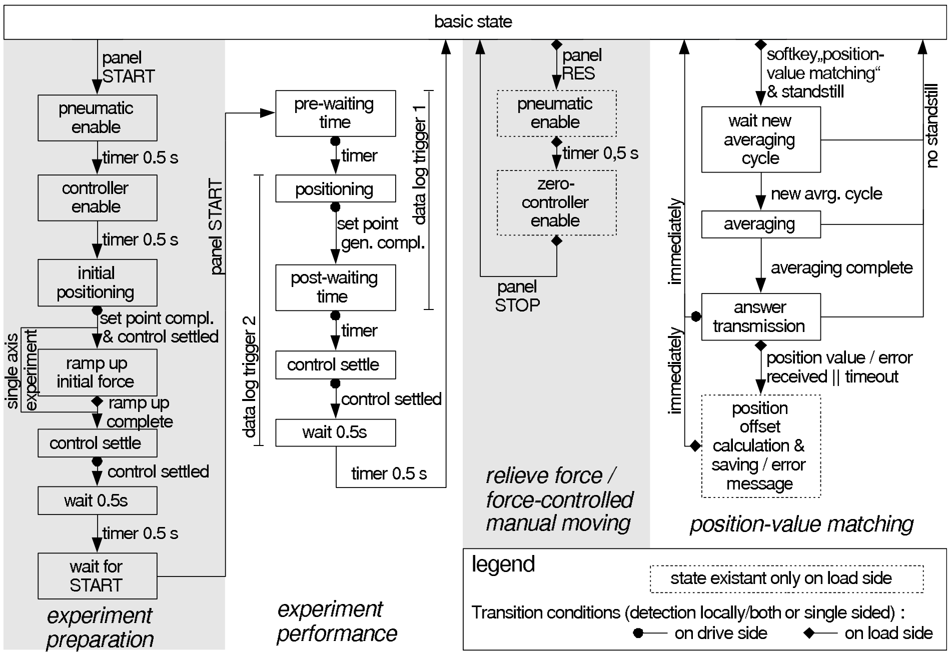

Fig. 3. Overview of the state machine behind the sequence control

The main chain establishes experiment preparation and performance. Preparation includes initial positioning by a modified sinoid according to Bestehorn regardless from the configuration of the actual positioning. If there is configured an initial force on load side, it is ramped up smoothly separately. After that, and at the end of an experiment, waiting states for settlement of the control are implemented. The second chain is available on load side only for force relieving in the linked system or manual moving by force control, see section 2.4. The last chain is used for matching the position values to one another in the case of mechanical linkage. This is done by applying an offset on load side (difference between the averaged, measured position values on both sides). As one can see, most transition conditions are detected only on one side. This is achieved through exchanging an OK-flag, which is done through a software interface provided by the firmware of the controller, managing communications over the CAN-Bus.

The control is configured for a basic cycle of one millisecond, taking account of the dynamics of the setting element with a time constant of about 1.6 msecs. On user program level, several task of different cycle times (in multiples of the basic cycle time) can be defined. The main tasks are implemented with one msec. However, the controller algorithm itself (together with output and linearization of the manipulated variable) gets a dedicated task of modifiable cycle time. Therefore, appropriate signal and implementation interfaces are created as shown subsequently.

2.3. Controller algorithm integration

The embedment of the open controller algorithm is accomplished according to the principle illustrated by Fig. 4. The pre- und post-controller sub tasks have to be executed in the same task – that is with the same cycle time – as the controller algorithm. The pre-controller sub task is executed prior to calling the controller algorithm. For the post-controller sub task there is a separate include file, which has to be included at the end of the implementation of the controller algorithm. The choice for the cycle time is made within the definition of the end of the task after including the file mentioned before. Parameters for the controller can be defined via variables to be accessed by the PC software.

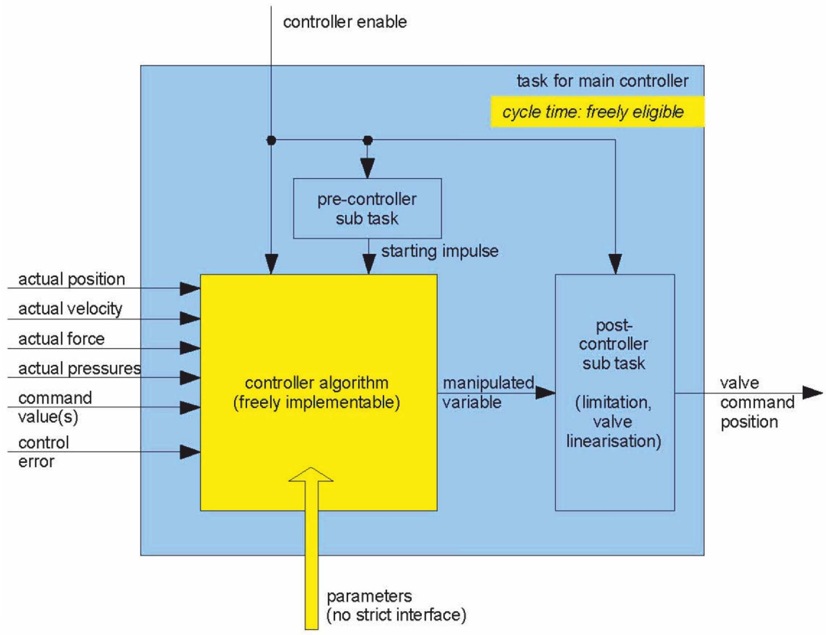

Fig. 4. Embedment of the controller algorithm

2.4. Force control on load side

As an example for a possible controller algorithm, force control on load side shall be an example. For force control, a possible movement is a disturbance measured in terms of velocity. Since this value is measured one can compensate its influence. To achieve this, feed forward control can be utilized, because imaginable the manipulated variable, being linearised against the valve characteristic, is a measure for air flow leading to both velocity and/or rise of force. So a proportional controller with additional disturbance feed forward is used like shown in Fig. 5.

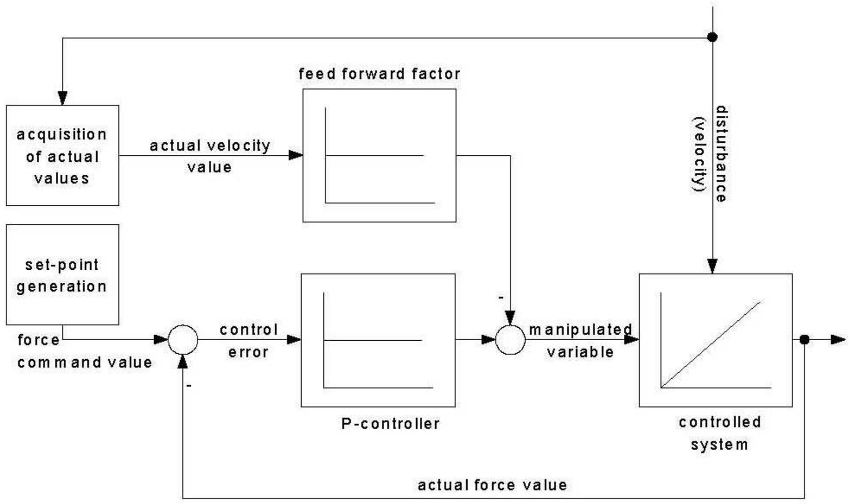

Fig. 5. Disturbance feed forward force control

As mentioned in Fig. 3, a mode is available to conduct the movement of the pneumatic drive by hand directly. This is realized through a fixed controller algorithm with fixed set-point of zero (set-point generation is ignored) implemented in the post-controller sub task overriding output of the main controller algorithm. The used controller is slightly different from the shown example, although basically a P-controller and velocity feed forward is used as well. A case differentiation is made on the existence of significant movement detected through limit check of absolute velocity. In the case of standstill velocity, feed forward is disabled and a defined death zone for control error is used to ignore noise. In the other case (movement) no death zone is applied to reduce sticking and feed forward is enabled to allow for faster movement with lower force. Along with good parameterization this gives the user the impression of a favorable static to gliding friction ratio, so that good and intuitive handling is possible. Usage of this mode is to easily establish (or loose) the mechanical linkage of the piston rods. Furthermore, an initial limitation of the manipulated variable when high forces are present allows for smooth relieve of possible stress from the linked system.

3. Simulation

3.1. Model

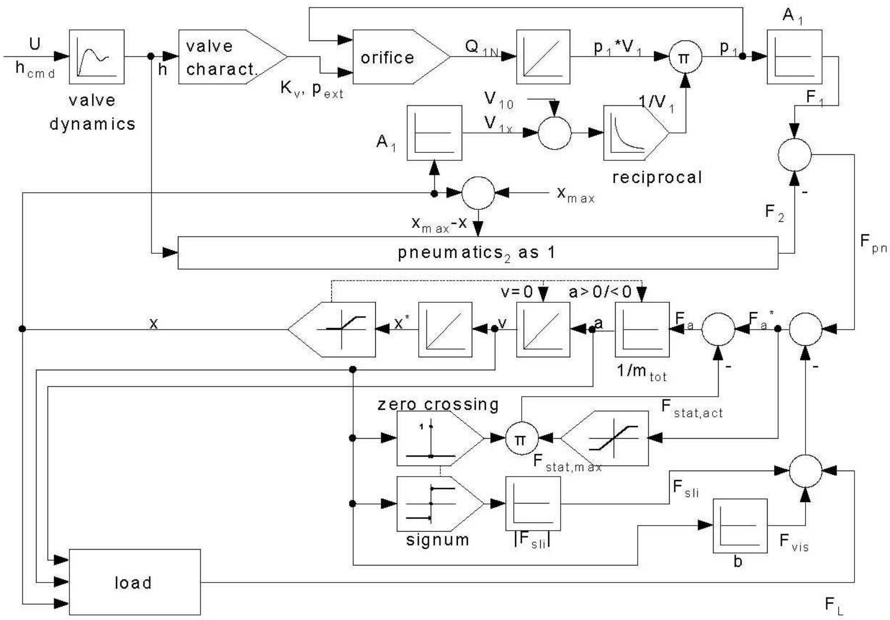

Fig. 6. Simulation model of one pneumatic axis

In Fig. 6 the block diagram used for simulation is shown. In the upper part the schema for the pneumatic system (for one cylinder chamber) can be seen. Central element is the integrator of the amount of air, measured in terms of pressure p multiplied by the volume V. The relation behind is the ideal gas equation, whilst constant temperature is assumed ignoring possible adiabatic effects:

$$p * V = n * R * T \tag{1}$$

With the temperature T assumed as constant – ignoring possible adiabatic effects – and R as universal gas constant it can be written as:

$$p * V = n * const \tag{2}$$

Since n is obviously determined by the time integral of the air flow through the corresponding valve port it can be written as

$$p * V = \int Q_N \tag{3}$$

with Q_N being normalized to standard conditions for temperature and pressure. The flow through the valve into the cylinder chambers is modelled by a simplified orifice formula as follows (compare [2]):

$$Q_N \approx \begin{cases} 25.9 \cdot K_V \cdot \sqrt{(p_P - p_S) * p_S} & \text{if} \quad p_P \geq p_S \geq \dfrac{p_P}{2} > 0 \\ 12.9 \cdot K_V \cdot p_P & \text{if} \quad \dfrac{p_P}{2} \geq p_S > 0 \end{cases} \tag{4}$$

with

$flow$ ($standard$ $conditions$) Q_N	$nominal$ $flow$ K_V	$pressures$ p_P, p_S
$^l/_s$	$^l/_s$	bar ($absolute$)

The case differentiation is made at the critical pressure ratio (rounded to 0.5) above which sonic speed is reached, making the flow independent from secondary pressure. In literature on modelling pneumatic valves, there are examples with more complex relations used [3, 4]. Beside for convenience, the simplification made here is owing to be able to implement the model in the controllers programming environment and run it in real time together with the user program on the controllers itself.

The modelling of the mechanical system (lower part of Fig. 6) is mainly about the integrators from acceleration to velocity to position x. These are fed by the resulting pneumatic force F_{pn} minus frictional forces composed of the two coulomb components static (F_{stat}) and sliding friction (F_{sli}) and F_{vis}, a velocity-proportional (factor b) part, similar to [5].

This can be expressed by a set of equations:

$$F_{pn} = F_a^* + b * \dot{x} + F_{sli} * sign(\dot{x}) + F_{load}(x,\ \dot{x},\ \ddot{x}) \tag{5}$$

$$F_a = m * \ddot{x} = \begin{cases} F_a^* & \text{if} \quad \dot{x} \neq 0 \\ F_a^* - sign(F_a^*) * F_{stat,\max} & \text{if} \quad \dot{x} = 0 \ \wedge (\left|F_a^*\right| > F_{stat,\max}) \\ 0 & \text{if} \quad \dot{x} = 0 \ \wedge (\left|F_a^*\right| \leq F_{stat,\max}) \end{cases} \tag{6}$$

For the implementation, a zero detection mechanism for the velocity is used to check the condition of $\dot{x}$. By this, the simulation model is capable of showing the stick-slip-effect.

3.2. Parameters and validation

Many parameters for the model can be found in the appropriate data sheets. These are maximum stroke x_{max} of 0.5 m, the piston areas A_i and moving masses of total 5 kg (m_{tot}). For the valve characteristics, only a clue diagram for one flow direction can be found in the data sheet. Piece-wise linear characteristics are implemented for forward and backward flows, so that nearly equal velocities in the simulation and the model are reached at given valve positions. Valve dynamics is modelled as PT1-element according to the limiting frequency given in its data sheet, resulting in a time constant of 0.0016 msecs. But by this, the overall dynamics for the simulation model is not yet able to represent the dynamics of the real system. A better result is achieved through adding another much greater time constant after orifice calculation, neglecting physical modelling at this point. Friction parameters can be estimated through experiments. These are $F_{stat,max}$ of 35 N and F_{sli} of 26 N for coulomb friction. The viscous friction factor b has been numbered to 1.1 N/(mm/s).

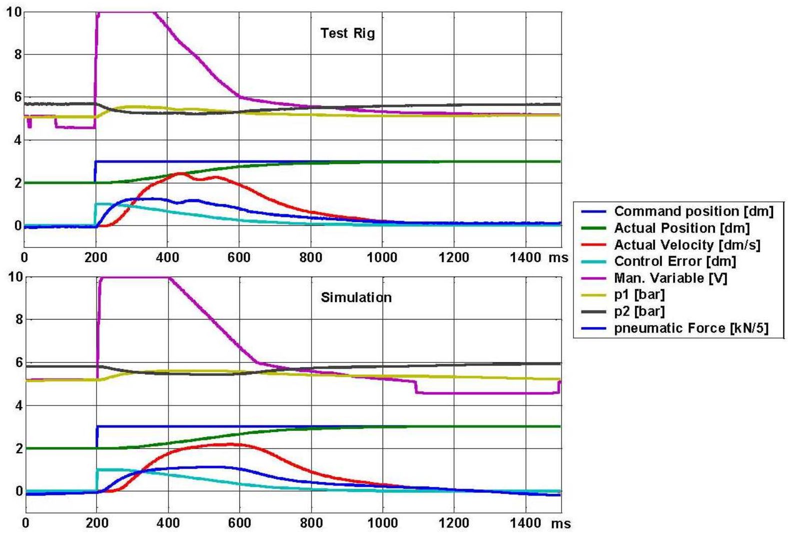

Fig. 7. Closed-loop position control step response (test rig compared to simulation)

The validation shall be shown by means of the comparison of the closed-loop step responses of the simulation and the test rig itself. The most important signals over time are shown in Fig. 7. There is good accordance in qualitative and quantitative manners. Maximum velocity and forces as well as dynamic parameters (settle time) are nearly identical. Differences can be seen in the dynamic behaviour (pressures, velocity) immediately after the step is applied in the way that the experimental rig reacts somewhat faster. A result is that the manipulated variable stays in the limitation for a longer period of time. Inaccuracies in the modelling of the valve characteristic are seen as the main cause for these discrepancies. Also, in [3, 5] it is pointed out, that standard-flow formulas fall short of describing the pneumatic system of valve and tubes quite closely, advanced approaches were used to get better results.

However, the simulation described here can be used to initially test changes in the user program or new controller algorithms quite good by keeping the inadequacies in mind.

4. Experiments

4.1. Point-to-Point positioning

To demonstrate the usage of the test rig, some results of experiments easy to arrange are to be shown. At first, a positioning by modified sinoid according to Bestehorn is apparent in Fig 8. A static reproducible positioning accuracy in dimension of tenths of a millimeter can be achieved. The dynamic error is up to a half a centimeter. The two maximuma in control error are supposed to be caused by inaccuracies in linearization of the valve characteristics. These makes the command velocity feed forward performing worse, obligating the proportional controller to compensate, which is not completely possible by principle. A cascaded speed controller with integral part would likely do the job better here.

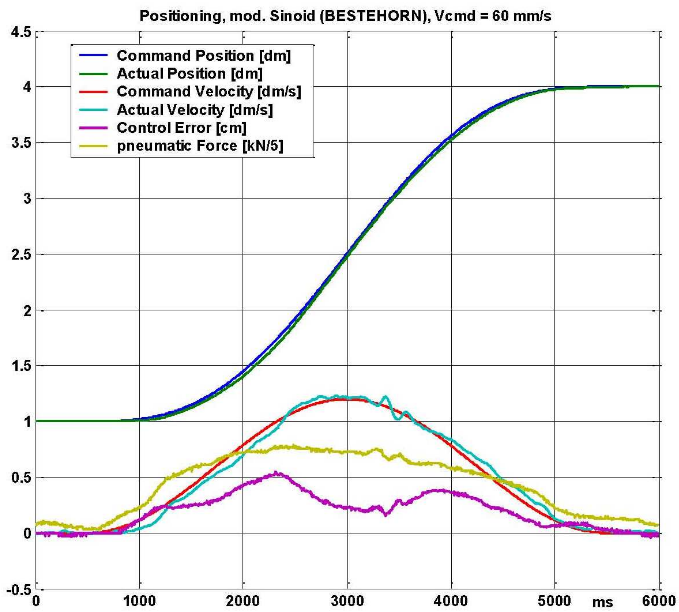

Fig. 8. Positioning by modified sinoid according to BESTEHORN

4.2. Applying a load step with the second axis

By the way of linking the pneumatic cylinders mechanically, a defined disturbance can be generated through force control. The same motion profile according to Fig. 8 was used, with a load step applied at 2.5 seconds. A low initial force was configured, so that possible mechanic backlash is compensated. The result is shown in Fig. 9. The (idealistic) force step was applied through the force control aperiodically in about 100 msecs. This nearly causes a standstill for a short time, the positioning error rises more than one centimeter. As one can see, this is evened out by the controller quite well. But of course, the positioning accuracy (dynamic as well as static) compared to the unloaded case (Fig. 8) can not be achieved.

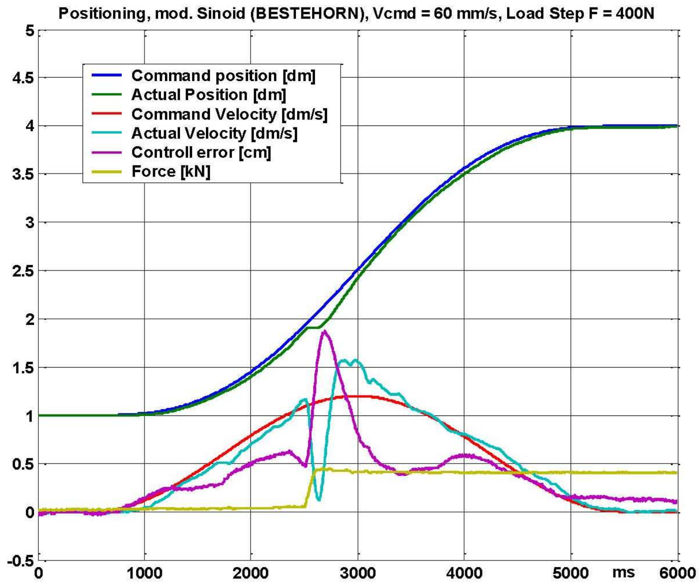

Fig. 9. Positioning disturbed by a load step

5. Conclusion

The servo-pneumatic experimental rig, automated by the concept presented in the paper, can be used for the proposed purpose that is to say for research and education in the field of motion control. It has been shown that the generation of defined disturbances by mechanical linkage of two pneumatic actuators of the same type works exemplarily with the experimental rig, due to application of force control. The force control loop performance is even supposed to be improvable e.g. through a more complex controller, like a PID-controller possibly combined with a characteristic map. A detailed, widely validated simulation model for a pneumatic axis was presented, which can be used through integration in the user program to perform initial tests (e.g. for intended changes) independently from the experimental rig.

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Sterowanie serwo-pneumatyczne w urządzeniu eksperymentalnym

S t r e s z c z e n i e

W artykule przedstawiono koncepcję sterowania w urządzeniu eksperymentalnym z osią sterowaną pneumatycznie w zamkniętej pętli. Celem była dogodna realizacja eksperymentów technicznych, w których swobodnie implementowano różne struktury sterowania. Ponieważ dwa siłowniki mogą być mechanicznie powiązane ze sobą, jeden z nich jest sterowany siłą w celu generowania zakłóceń. Ponadto, wskazano szczególny model symulacyjny uwzględniający efekty nieliniowe, który może być zintegrowany z programem sterownika tworzonym przez użytkownika. W części końcowej przedyskutowano wyniki wybranych eksperymentów.