Modelling and control of a flexible 'goliath' robot - a case study using ADAMS/Controls

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Abstract

The paper deals with the modelling and control of a large flexible arm from a 'goliath' transfer robot. The simulation environment is composed of the tools ADAMS, ANSYS and MATLAB/SIMULINK which are integrated via suitable interfaces. For the exchange of a FE structure a Modal Neutral File is used whereas the online module ADAMS/Controls is taken for the combination of mechanical and control environment. The flexible beam is described with mathematical Lagrange/Ritz equations on the one hand and with a FE model on the other hand. The different techniques are compared using eigenfrequency analyses. The controller design is performed according to the derived mathematical equations. Next this feedback loop is tested with the 'real' FE system. The obtained simulation results are discussed and verified. Advantages and disadvantages of the chosen approaches are stated.

1 Introduction

Integrated environments consisting for example of design and analysis tools have the potential to tremendously shorten design cycles and increase the productivity. Two needs are more and more given. First of all it is necessary to integrate (mechanical) motion simulation and control system design. Secondly it is important to include flexible bodies in a rigid system so that overall 'hybrid' simulation studies can be performed. In the present paper interface structures are investigated which are directly provided by the engineering tools.

As case study the design of a 'goliath' robot is described. The module is developed for the automatic container transfer between trains and trucks [3;4]. Due to the high loads the process is very challenging and requires thorough simulation studies in an integrated simulation tool environment.

2 The tool environment

In the present study three simulation tools are used:

ADAMS: for the dynamic modelling and simulation of the mechanical structure [1]

MATLAB: for the controller synthesis [7]

ANSYS: for the structural analyses [2]

Especially important in a combined simulation environment are suitable interface structures [5]. The following interface structures are discussed in detail:

1. ANSYS - ADAMS
   In order to integrate flexible bodies in the (rigid) mechanical system Modal Neutral Files are used. The model validity is checked using eigenfrequency analyses.

2. ADAMS - MATLAB / SIMULINK
   To include suitable control structures mainly two possibilities can be distinguished:

   ADAMS/Linear: is an offline structure which performs an exchange of linearized state space matrices.

   ADAMS/Controls: allows an online integration of both tools where the ADAMS model is included as SIMULINK super block. One distinguishes a (discrete) cosimulation mode (where both systems use their own solvers) and a (continuous) function evaluation mode (in which an overall integration is performed in the control environment).

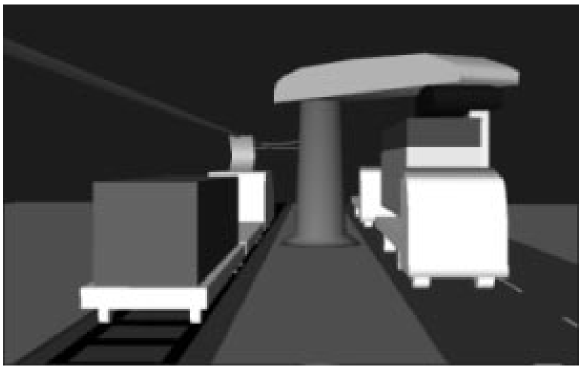
The finally chosen procedure is the following. First of all the (rigid) mechanical system is build in ADAMS. Next the flexible arm is imported via MNF. The resulting hybrid model is verified using eigenfrequency analyses. Now the system behaviour is investigated in ADAMS. Parallel a mathematical model is build up using MATLAB. Both approaches are compared. Then a suitable controller is designed using the mathematical equations. Finally this feedback loop is tested with the 'real' FE robot using the online ADAMS/Controls interface.

3 Description of the process

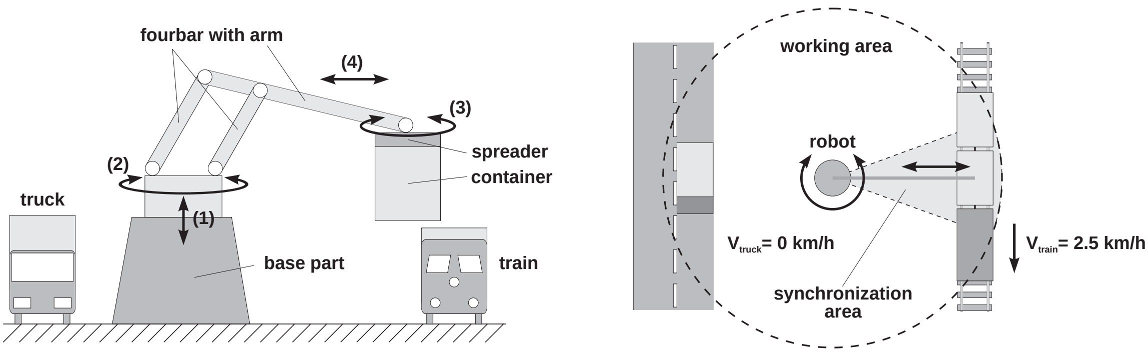
During the last years the traffic situation on German highways has worsened. Traffic jams are daily 'phenomenons'. A possible alternative is offered by the combined transport. It tries to combine the long distance capacity potentials from trains with the high flexibility of the trucks. The critical point is the exchange where the goods have to be transloaded from one transport media to the other.

(a) Design study

(b) Model (1:45)



(c) Simulation study



(d) The robot structure

Figure 1: The 'goliath' robot

The Institute for Control and Automation Engineering at the TU Braunschweig is engaged in this topic since several years. One main topic is the design of a 'goliath' transfer robot. Its tasks are

- to transship containers directly between different medias like trains and trucks
- to grab goods from moving transport medias, e.g. a train can move at about 2 to 3 km/h
- to work in a safe critical environment, e.g. under an aerial contact line

Extremely challenging are the difficult conditions resulting from the process. The different container types to be handled weigh up to 35 tons. The cycle time for one complete transloading should be less than one minute. Fig. 1 shows results from different studies.

The following reflections will concentrate on the rotational movement of the arm together with the masses of spreader and container.

4 Modelling of a flexible beam structure

Different methods of modelling flexible beam structures can be distinguished according to their special tasks.

- On the one hand mathematical equations are e.g. needed for a controller synthesis.
- On the other hand simulation and control studies of a (hybrid) mechanical system require an exact Finite Element model.

4.1 Mathematical description

For the mathematical approach the Bernoulli beam theory is taken, which is sensible for thin, long structures. A 4th order partial equation is obtained

$$EI\frac{\partial^4 w(x,t)}{\partial x^4} + \frac{m_B}{l} \cdot \frac{\partial^2 w(x,t)}{\partial t^2} = 0 \tag{1}$$

with the elasticity module E, inertia I, beam mass m_B, beam length l, bending w, beam coordinate x and time t.

The arm structure is modelled as a beam with one free end and one rotationally fixed end. The joint angle Φ_G represents the torsion of the joint whereas the overall movement of the free end is described with the total angle Φ_T which is the sum of joint angle and deflection (fig. 2).

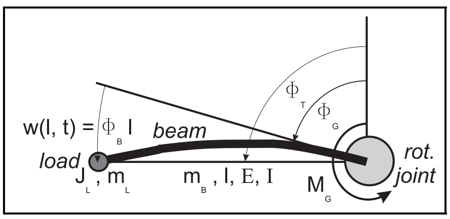


Figure 2: Beam with rotational joint and free end

As the exact solution of the bending vibration results in model of infinite order, a discretized Ritz approximation is implemented for the time (t) and location (ξ) dependent deflection

$$w(\xi,t) = \sum_{i=1}^{N} \overline{\overline{w}}_i(\xi)\overline{w}_i(t) \tag{2}$$

The static deflection is discretized using basic functions:

$$\overline{\overline{w}}_1(\xi) = \frac{\xi^2}{l^2},\ \overline{\overline{w}}_2(\xi) = \frac{\xi^3}{l^3},\ \overline{\overline{w}}_3(\xi) = \frac{\xi^4}{l^4}, \ldots \tag{3}$$

Now the Lagrange equations 2nd order are set up

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\boldsymbol{q}}}\right) - \frac{\partial T}{\partial \boldsymbol{q}} + \frac{\partial U}{\partial \boldsymbol{q}} + \frac{\partial R}{\partial \dot{\boldsymbol{q}}} = \frac{\partial W}{\partial \boldsymbol{q}} \tag{4}$$

with the kinetic energy T, potential energy U, dissipation R, work of external forces W and general coordinates $\boldsymbol{q}$ resulting in the following differential equation set

$$\boldsymbol{J}\begin{pmatrix}\ddot{\Phi}_G(t)\\ \ddot{\overline{\boldsymbol{w}}}(t)\end{pmatrix} + \boldsymbol{D}\begin{pmatrix}\dot{\Phi}_G(t)\\ \dot{\overline{\boldsymbol{w}}}(t)\end{pmatrix} + \boldsymbol{C}\begin{pmatrix}\Phi_G(t)\\ \overline{\boldsymbol{w}}(t)\end{pmatrix} = \begin{pmatrix}M_G\\ \boldsymbol{0}\end{pmatrix} \tag{5}$$

with the terms for inertia $\boldsymbol{J}$, dissipation $\boldsymbol{D}$, elasticity $\boldsymbol{C}$, the joint angle Φ_G and the beam moment M_G.

4.2 Finite Element model description

For the modelling of a FE structure the tool ANSYS is used. The beam is composed of shell elements which include warping and the occurring stress effects. In a next step the flexible beam is imported into ADAMS using a Modal Neutral File which includes information on modes, geometry and masses.

Of course other modelling approaches (in both tools) exist:

- In ANSYS thin-walled plastic beam elements (St. Vernant's theory) could be used instead. They do not include warping effects but are very similar to the mathematical approach.
- In ADAMS itself discrete flexible links exist which consist of several rigid bodies connected by massless beams.

4.3 Comparison

Table 1 shows a comparison of the eigenfrequencies for the following modelling approaches:

- mathematical equations (Bernoulli beam, Lagrange equations with 4 basic Ritz functions)
- ANSYS FE model with thin walled plastic beam elements
- ANSYS FE model with shell elements
- ADMAS flexible body using MNF integration
- ADAMS flexible link

Table 1: Beam eigenfrequencies [Hz] for different modelling approaches

	Lagrange/	ANSYS model		ADAMS model	
	Ritz model	beam	shell	MNF	link
1. rot. pole	0	0	0	0	0
1. rot. zero	2.13	2.13	2.19	2.19	3.89
2. rot. pole	32.3	32.7	31.5	31.5	30.6
2. rot. zero	50.9	50.7	44.3	44.5	n.m.
1. long.	n.m.	30.7	31.1	31.1	n.m.
1. tors.	n.m.	69.7	69.2	74.6	n.m.

The following results are obtained. The system 'poles' are equivalent to the resonance frequencies in horizontal direction whereas the 'zeros' correspond the resonances in vertical direction. Especially important for the controller design are the first (vertical and horizontal) resonance frequencies at 2 and respectively 32 Hz, which are both below 50 Hz.

When having no load mass instead of the 40 tons the frequency increases to 11 and 50 Hz.

The FE beam model shows a very nice correspondence with the Lagrange/Ritz equations. The shell element model shows some small differences (for the first eigenfrequencies) which are mainly due to warping effects. The import of the FE model in ADAMS works well like the small deviation in the modes show. In contrary the flexible link has quite some differences and is therefore not suitable for further investigations.

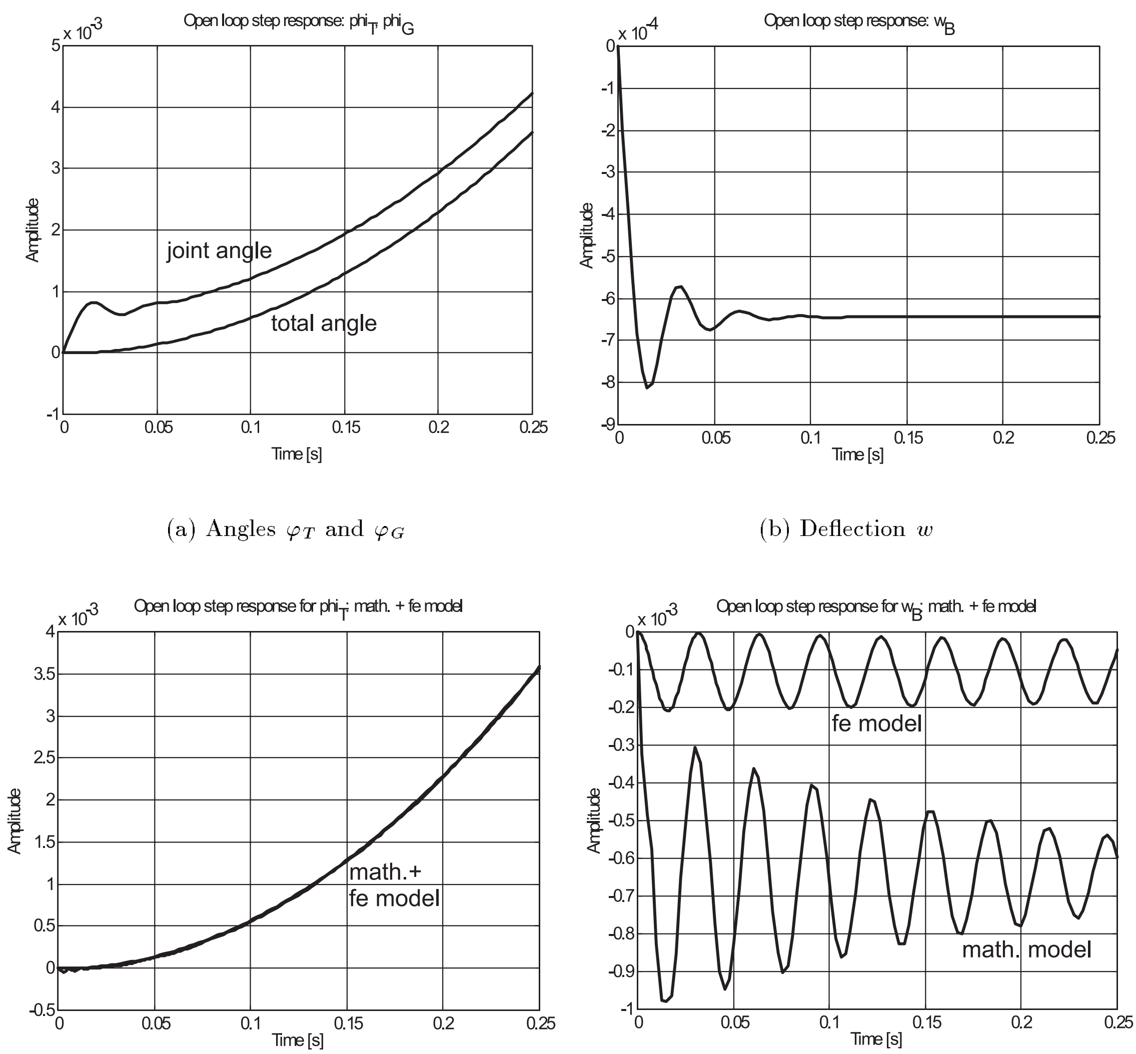
Later on attention should be paid to the first torsional and longitudinal resonances which are below 70 Hz as well.

4.4 Simulation results

Fig. 3 shows some simulation results for an open loop step response. Clearly visible is the quadratic increase of the angles φ_T and φ_G which is due to the constant acceleration acting on the joint. The deflection shows the eigenfrequency of about 30 Hz corresponding to table 1.

When comparing the mathematical and the FE model the beam angle shows about no difference at all whereas the bending shows few differences in damping and deflection ratio. The reason is that the damping is chosen a little different in both approaches (according to the different theories). The exact value could only be obtained from measurements on a real beam structure.

All in all the results are very satisfying. Hence it will be possible to design a controller using the mathematical Lagrange/Ritz equations and then test it with the FE system.



(a) Angles φ_T and φ_G

(b) Deflection w

(c) Comparison of the mathematical and the FE model for φ_T (left) and w (right)

Figure 3: Simulation results: step response open loop system

5 Controller design

An effective control structure is heavily depending on the available process model. Different model types can be distinguished:

1. white-box model: physical equations and parameters known
2. light-grey-box model: physical equations known, parameters unknown, in-/output signals measurable
3. dark-grey-box model: physical equations known, model structure and parameters unknown, signals measurable
4. black-box model: in-/output signals measurable, model class assumed

When now looking at an ADAMS model, it is often found to be a black-box type. Without some knowledge of the process or of possible control structures, it is very difficult to find a good controller design. Different approaches exist which are more or less effective. Examples are:

- *try and error:* most ineffective
- *separate mathematical equations:* time intensive, may be faulty
- *system linearization (ADAMS/Linear)*: easy, nice if it is possible [5]
- *system identification:* quite time consuming, only possible with a suitable input signals [6]
- *fuzzy:* possible if expert knowledge is available
- *neuronal net:* quite complicated

The approach using ADAMS/Linear is discussed in [5]. In the following the proposed mathematical Lagrange/Ritz model will be used.

Fig. 4 shows the closed loop block diagram whereas the obtained simulation results are presented in fig. 5.

The control structure is composed of an outer feedback loop for the beam angle and inner active velocity and acceleration damping loops for the joint angle. Important is the use of the lowpass filters which are designed according to the eigenfrequencies of the beam.

Both closed loop models show a very good correspondence. A step input is controlled in about 7 s with only a small overshoot. Interesting is the vibration of the FE beam. It is due to its

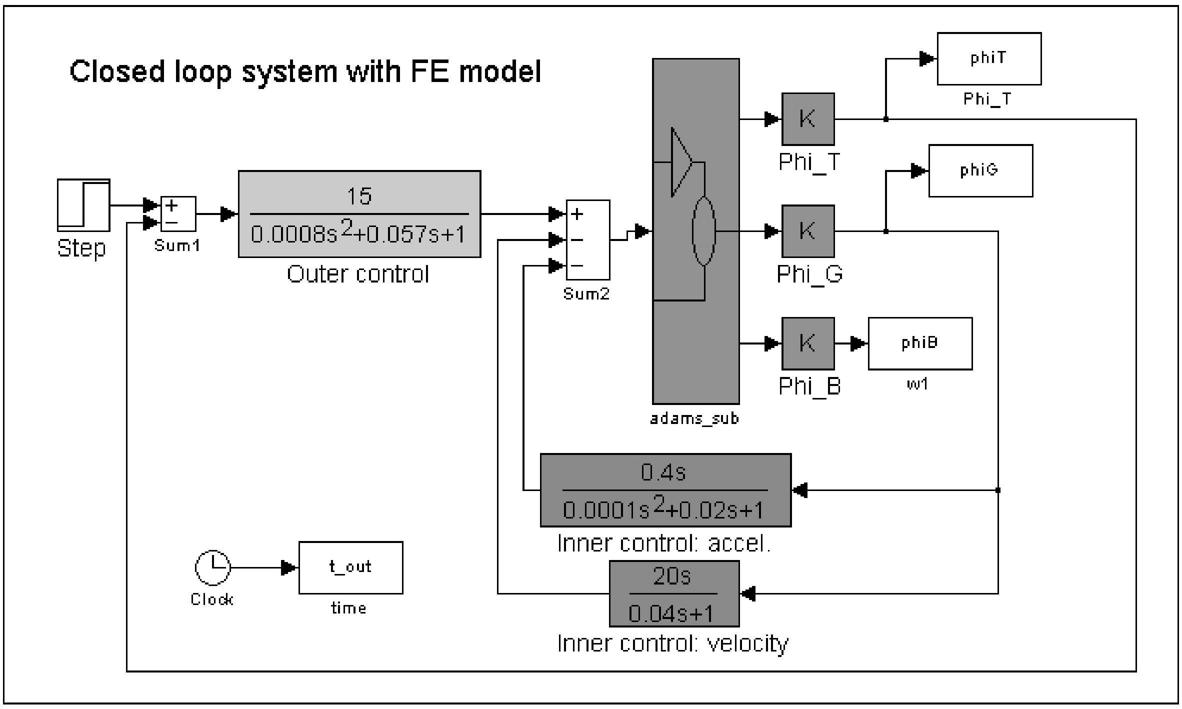
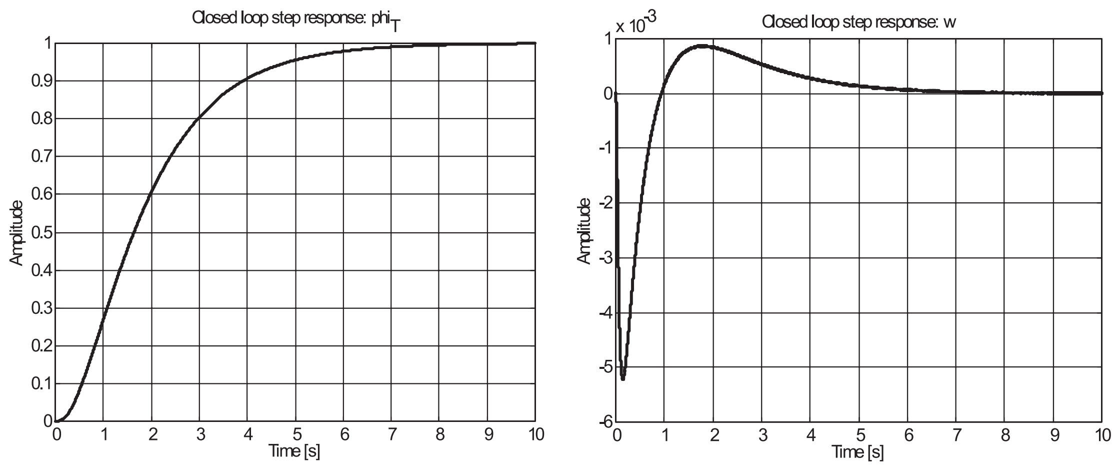
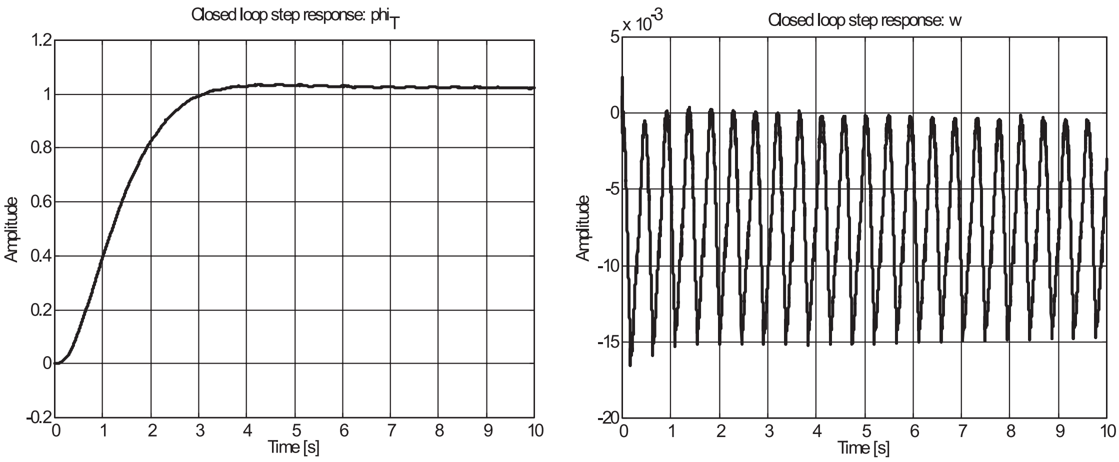


Figure 4: Block diagram: closed loop system



(a) Mathematical model (φ_T and w)



(b) Finite Element model (φ_T and w)

Figure 5: Simulation results: step response of the closed loop system

vertical resonance frequency of about 2 Hz (see table 1). Consequently it can not be observed in the mathematical model which only includes the equations for the horizontal beam model. The deflection can be eliminated using a further vertical active damping structure. The derived conclusion is very important. Although the beam is only excited in horizontal direction, active damping structures for all modes have to be included.

For the present simulation the continuous solver mode was chosen. The CPU time needed for a 10 s simulation was over 400 s (Pentium II, 300 MHz). When using the discrete method, this time is still exponentially increasing (with decreasing stepsize) due to the big amount of communication time needed [5]. Attention has to be paid to the relative and absolute tolerances of the solvers. If they are too big noise is occurring. The same is true for the measuring points used in the process which must have quite a high resolution.

6 Conclusions

For the modelling and control of a flexible 'goliath' robot a tool environment is proposed which consists of the tools ADAMS, ANSYS and MATLAB/SIMULINK. Especially the provided interface structures using both the MNF import and the online ADAMS/Controls module are implemented and verified.

The mathematical modelling of the beam using Lagrange functions together with a Ritz discretization are found to be sufficient for a following controller design. The verification is based both on eigenfrequency and on step response analyses which compare the mathematical and the FE approach in detail.

The designed controller is based on an active damping concept. A step response (for both model types) is controlled in about 6 s. Important is the result that an active damping has to be implemented for all (horizontal and vertical) eigenfrequencies of beam.

The computational demands for the online interface with the flexible model are quite high. In the continuous mode over 400 s of CPU time are necessary for a 10 s simulation, the discrete mode needs even more.

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