Trajectory Tracking Control for a new Generation of Fire Rescue Turntable Ladders

Steven Lambeck
Institute of Automation and System Science
Technische Universität Ilmenau
PO. Box 100565, D-98684 Ilmenau , Germany
steven.lambeck@tu-ilmenau.de

Oliver Sawodny, Eckhard Arnold
Institute for System Dynamics
University of Stuttgart
Pfaffenwaldring 9, D-70569 Stuttgart, Germany
[sawodny,arnold]@isys.uni-stuttgart.de

***Abstract*—A trajectory tracking control was developed for a new generation of fire turntable ladders with an additional joint in the upper ladder part in order to reduce swaying of the moving cage because of the limited stiffness of the ladder set. Therefore, the arm elasticity, which is characterized by significant flexion and torsion, plays an important role in the design process, especially in the case of large ladder lengths. A decentralized control strategy based on a dynamic model of the ladder has been developed. The implementation of this control strategy leads to an active oscillation damping for all ladder movements and therefore allows higher velocities for the rescue operations.**

***Keywords*—trajectory tracking control, decentralized control, manipulator with arm elasticity**

I. INTRODUCTION

The turntable ladder treated in this paper can be interpreted as a large scale robot with three rotatory axes, which represents the turning movement (φ_T : not limited), the raising movement of the whole ladder set (-15°< φ_R < 75°), the raising and lowering of the upper ladder part (0°< φ_J < 70°) and one translational axis (9.2m < l_L < 30m) related to the telescoping of the ladder (see Fig. 1).

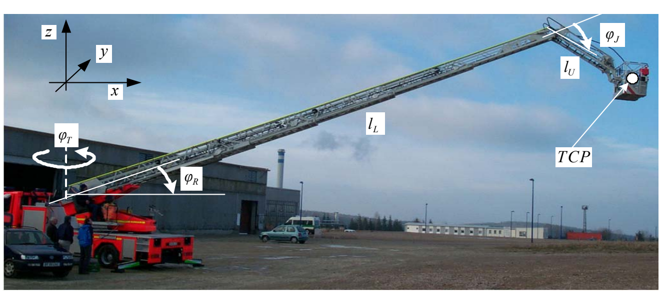

Figure 1. Fire turntable ladder with additional joint in the upper ladder part

The stiffness of the ladder set is limited, because a lightweight construction should guarantee a minimized weight in order to achieve a large workspace. This limited stiffness leads to a significant swaying of the cage especially in the case of large ladder lengths. For a convenient description of this effect additional degrees of freedom which represents the arm elasticity in the model to be developed (see Section II) are introduced. Besides the flexion in vertical and horizontal direction the torsion of the ladder set is a significant effect due to the additional joint.

The swaying of the cage can be avoided by reducing the maximum velocity, which is not desirable due to the purpose of the ladder as a rescue vehicle, or by a suitable shaping of the reference functions [8], which leads also to a reduction of the achievable axes velocities. So a control strategy for this new generation of fire-rescue turntable ladders which guarantees an active damping of the upcoming oscillations with the aim of increasing velocities for the movement of the axis was developed. As in a former project [2],[3] the combination of a control strategy and a trajectory generation seems to be useful. The trajectory generation module, which will not be explained further in this paper, is responsible for the evaluation of time indexed reference functions for the joint position, velocity, acceleration and jerk based on signals either from the handlevers (semi-automated mode) or auxiliary stored points in a target point matrix from a former teached-in path (automated mode) [1], [3]. An important difference to [1],[2],[3] is the significant torsion of the ladder set. This additional degree of freedom as a result of the arm elasticity implies consequences concerning the modeling of the system and the control design.

Following the remarks in [1] and [2] a decentralized control approach is sufficient for the present application, because the resulting control algorithm needs to be transferred to a micro-controller system with limited computational power used as control unit in the vehicle. The control design is based on a dynamic model of the turntable ladder. This model was derived using the Lagrange formalism instead of the Newton-Euler-Method [9], because the Lagrange formalism is well suited for complex systems and leads to a separate equation of motion for each degree of freedom. The derivation of this model will be explained in Section II. Some remarks on the chosen automation concept will be made in Section III. In Section IV a brief description of the decentralized control design as a combination of an adaptive feedforward control and a robust designed feedback module is given. The measurement results shown in Section V demonstrate the effectiveness of the proposed decentralized control design.

II. DERIVATION OF THE DYNAMIC MODEL

The system is characterized by four rigid degrees of freedom, which represent the joint positions, and six additional degrees of freedom for the arm elasticity (flexion of the ladder in vertical and horizontal direction as well as the torsion) of the ladder set and the upper ladder part as can be seen in Fig 2.

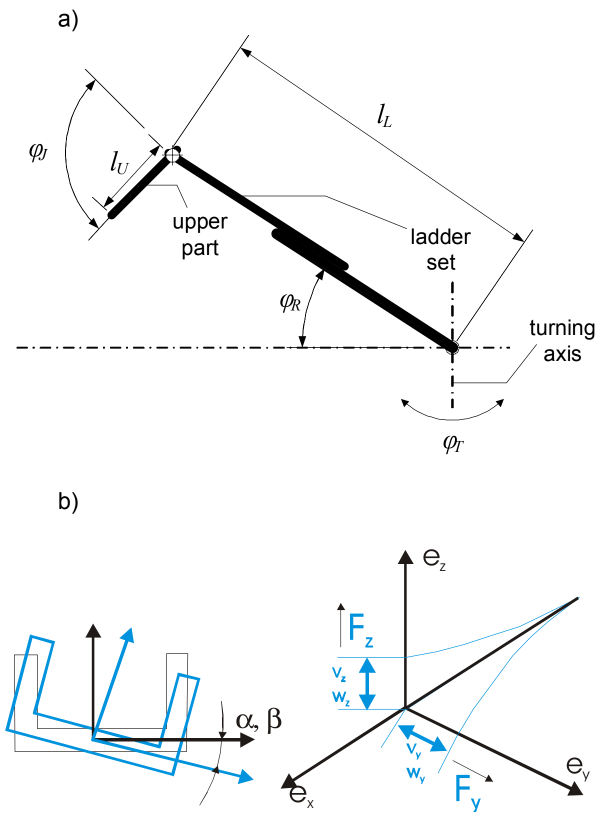

Figure 2. a): rigid degrees of freedom ; b): arm elasticity

The dynamic model of the turntable ladder can be derived using the Lagrange formalism on a multi-body system with elasticity and damper-elements, where the different bodies are approximated by three equivalent masses (concentrated in three point masses) and the arm elasticity is approximated by spring-damper elements (see Fig 3).

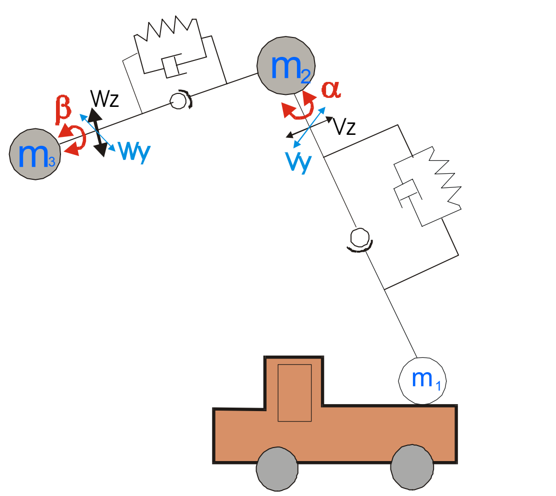

Figure 3. Approximation of the turntable ladder by equivalent masses and spring-damper elements

Equation (1) represents the resulting system of the equations of motion with ten interdependent equations.

$$\underline{M}\,\underline{\ddot{q}} + \underline{D}\,\underline{\dot{q}} + \underline{C}(\underline{\dot{q}},\underline{q}) + \underline{K}\,\underline{q} + \underline{G}(\underline{q}) = \underline{F}\ . \qquad (1)$$

$\underline{q}$ is the combined vector of the joint positions (rigid degrees of freedom) and the components representing the arm elasticity (the vertical and horizontal flexion of ladder set and upper ladder part as well as the torsion), $\underline{M}$ is the inertia matrix, $\underline{D}$ is a matrix of damping factors, $\underline{C}$ is the matrix of Coriolis and Centripetal forces, $\underline{K}$ is the matrix of equivalent stiffness coefficients, $\underline{G}$ the vector of gravitational forces and $\underline{F}$ is the vector of the driving forces and torques. Because of the high stiffness of the upper ladder part compared to the ladder set (caused by the nearly tenfold length of the ladder set) the arm elasticity in the upper ladder part can be neglected. So a reduction of the system order from ten to seven can be realized. This is necessary because the control algorithm has to be realized on a micro-controller with limited computational power. The vector $\underline{q}$ is shown in (2) with α_x as the torsion, v_y as the flexion in horizontal direction and v_z as the flexion in vertical direction:

$$\underline{q} = \begin{bmatrix} \varphi_T & \varphi_R & \varphi_J & l_L & \alpha_x & v_y & v_z \end{bmatrix}^T \qquad (2)$$

Based on the structure of the simplified inertia matrix shown in (3)

$$\underline{M}\,\underline{\ddot{q}} = \begin{bmatrix} m_{11} & 0 & 0 & 0 & m_{15} & m_{16} & 0 \\ 0 & m_{22} & m_{23} & m_{24} & 0 & 0 & m_{27} \\ 0 & m_{32} & m_{33} & m_{34} & 0 & 0 & m_{37} \\ 0 & m_{42} & m_{43} & m_{44} & 0 & 0 & 0 \\ m_{51} & 0 & 0 & 0 & m_{55} & m_{56} & 0 \\ m_{61} & 0 & 0 & 0 & m_{65} & m_{66} & 0 \\ 0 & m_{72} & m_{73} & 0 & 0 & 0 & m_{77} \end{bmatrix} \begin{bmatrix} \ddot{\varphi}_T \\ \ddot{\varphi}_R \\ \ddot{\varphi}_J \\ \ddot{l}_L \\ \ddot{\alpha}_x \\ \ddot{v}_y \\ \ddot{v}_z \end{bmatrix} \qquad (3)$$

two groups of differential equations can be distinguished. In the first group all degrees of freedom, which are influenced by the turning movement (φ_T, α_x and v_y) are subsumed in (4). The second group in (5) contains all degrees of freedom influenced by the raising movement (φ_R and v_z). Furthermore, the influence of the Coriolis and Centripetal forces can be neglected ($\underline{C}=0$). The equations (4) and (5) are derived by applying the Lagrange formalism assuming the mentioned simplifications:

$$\begin{aligned} \varphi_T &: m_{11}\ddot{\varphi}_T + m_{15}\ddot{\alpha}_x + m_{16}\ddot{v}_y + d_{11}\dot{\varphi}_T = M_T \\ \alpha_x &: m_{51}\ddot{\varphi}_T + m_{55}\ddot{\alpha}_x + m_{56}\ddot{v}_y + k_{55}\alpha_x + d_{55}\dot{\alpha}_x = 0 \\ v_y &: m_{61}\ddot{\varphi}_T + m_{65}\ddot{\alpha}_x + m_{66}\ddot{v}_y + k_{66}\alpha_x + d_{66}\dot{v}_y = 0 \end{aligned} \qquad (4)$$

$$\begin{aligned} \varphi_R &: m_{22}\ddot{\varphi}_R + m_{27}\ddot{v}_z + g_{21} + d_{22}\dot{\varphi}_R = M_R \\ v_z &: m_{72}\ddot{\varphi}_R + m_{77}\ddot{v}_z + g_{71} + d_{77}\dot{v}_z + k_{77}v_z = 0 \end{aligned} \qquad (5)$$

Nevertheless, the particular elements of the inertia matrix are complex and dependent on the joint positions, the equivalent masses and other values, which is exemplified in (6) by means of m_{51}.

$$m_{51} = (m_3 l_U^2 cos(\varphi_R) cos(\varphi_J) + m_3 l_U cos(\varphi_R) l_L) \cdot \ldots \\ \ldots \cdot sin(\varphi_R) + m_3 l_U^2 sin(\varphi_J)^2 sin(\varphi_R) \tag{6}$$

The elements of the stiffness matrix $\underline{K}$ are determined by experimental measurements and approximate polynomial functions of 3rd order depending on the ladder length. The damping coefficients (elements of $\underline{D}$) are also the result of experimental measurements.

The torques M_T and M_R from (4) and (5) are caused by hydraulic drives and representing the inputs of the system. The hydraulic drive system is influenced by servo valves with underlying flow rate control loops. In contrast to [1], [2] a simpler approach for the description of the dynamic behaviour of the drives is used [7], which will be exemplified by the hydraulic motor for the turning movement. The flow rate Q_{FT} of the hydraulic oil is influenced by the control voltage u_T. This relation can be approximated by a simple first order system with delay time T_T and proportional constant K_{PT}:

$$Q_{FT} = \frac{K_{PT}}{1 + sT_T} u_T \tag{7}$$

With the steady state approximation

$$\frac{2\pi}{i_T V} Q_{FT} = \dot{\varphi}_T \tag{8}$$

the combination of (7) and (8) in time domain leads to

$$\ddot{\varphi}_T = -\frac{1}{T_T} \dot{\varphi}_T + \frac{K_{PT}}{i_T \frac{V}{2\pi} T_T} u_T \tag{9}$$

with i_T as the gear transmission ratio and V as the volume of the hydraulic motor. Following (9), instead of M_T the velocity $\dot{\varphi}_T$ can be considered as the input of the system, which simplifies the assembly of the equations for the control design in IV. The corresponding equations for the hydraulic cylinders can be derived in a similar way.

III. AUTOMATION CONCEPT

The automation concept for the application was chosen as proposed in [2] extended by an additional servo valve in order to control the additional joint φ_J (Fig. 4)

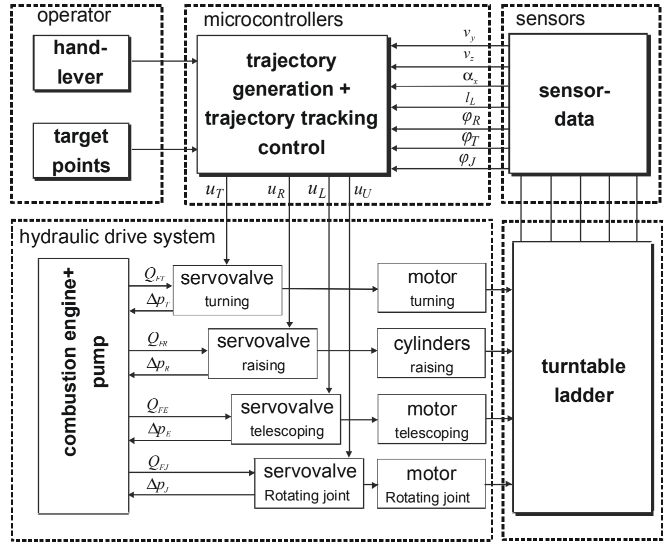

Figure 4. Applied automation concept

The ladder is driven by a hydraulic drive system consisting of the four servo valves, the combustion engine with the hydraulic pump, the motors and cylinders. The input voltages u_T, u_R, u_L and u_U, which are the outputs of the corresponding control modules, influence the different movement directions. The ladder length l_L and the three angles φ_R, φ_T and φ_J are evaluated by encoder measurements. The associated velocities are determined by real differentiation. The sensor values representing the arm elasticity (horizontal flexion v_y, vertical flexion v_z and torsion α_x) are provided by a set of four strain gauges.

The trajectory tracking control consists of a trajectory generation, which is responsible for the generation of time reference functions for the different movement directions [1], and the control modules, which have to be designed. Other solutions for the problem of trajectory control and tracking control are proposed in [10],[11].

The evaluated time reference functions are the inputs of the corresponding control modules (Fig. 5). The control design is based on the dynamic model derived in Section II. According to [2], [4] the decentralized control approach [5], [6] was chosen for the control design as explained in the next section at the example of the turning movement. The corresponding control module for the raising movement can be derived in a similar way.

IV. DECENTRALIZED CONTROL DESIGN

The dynamic model was divided into the two sub-models shown in (4) and (5) for the different movement directions. It is assumed that the elasticity degrees of freedom are influenced separately by the different movement directions, that means the horizontal flexion and the torsion are influenced only by the turning movement (represented by the equations in (4)) respectively the vertical flexion is influenced by the raising movement only (represented by the equations in (5)). Under this assumption the coupling terms due to the synchronous movement of the different axes are neglected. For the sub-model of the turning movement a state space representation in the form

$$\begin{aligned} \dot{\underline{x}}_T &= \underline{A}_T(\underline{q})\underline{x}_T + \underline{B}_T(\underline{q})u_T \\ \underline{y}_T &= \underline{C}_T\,\underline{x}_T \end{aligned} \tag{10}$$

with

$$\underline{x}_T = \begin{bmatrix} \dot{\varphi}_T & \alpha_x & \dot{\alpha}_x & v_y & \dot{v}_y \end{bmatrix}^T \tag{11}$$

can be derive. Based on this system representation the combined use of (4) and (9) with (11) as state space vector results in

$$\begin{aligned} &\underbrace{\begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ m_{51} & 0 & m_{55} & 0 & m_{56} \\ 0 & 1 & 0 & 0 & 0 \\ m_{61} & 0 & m_{65} & 0 & m_{66} \\ 0 & 0 & 0 & 1 & 0 \end{bmatrix}}_{\underline{H}} \cdot \dot{\underline{x}}_T = \ldots \\ &\ldots = \begin{bmatrix} -\dfrac{1}{T_T} & 0 & 0 & 0 & 0 \\ 0 & -k_{55} & -d_{55} & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & -k_{66} & -d_{66} \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \underline{x}_T + \begin{bmatrix} \dfrac{K_{PT}\,2\pi}{i_T V T_T} \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} u_T \end{aligned} \tag{12}$$

A multiplication of the right side of (12) with $\underline{H}^{1}$ leads to the matrices $\underline{A}_T$ and $\underline{B}_T$. The output of the system is the turning movement:

$$y_T = \underbrace{\begin{bmatrix} 1 & 0 & 0 & 0 & 0 \end{bmatrix}}_{\underline{C}_T} \cdot \underline{x}_T \tag{13}$$

Following the solution proposed in [2], the combination of a feedforward control and a feedback loop, both characterized by adaptive properties, is chosen. Fig. 5 shows the realized structure of the turning control module.

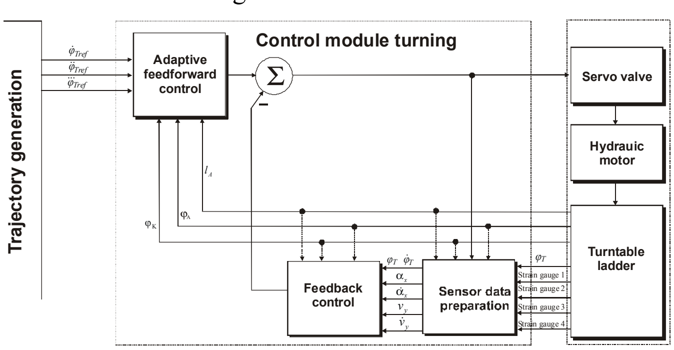

Figure 5. Structure of the control module for the turning movement

Input of the adaptive feedforward control is the vector of the time reference functions generated by the trajectory generation module:

$$\underline{w}_T = \begin{bmatrix} \dot{\varphi}_{Tref} & \ddot{\varphi}_{Tref} & \dddot{\varphi}_{Tref} \end{bmatrix}^T \tag{14}$$

The elements of the feedforward control are the gains K_{FT0} to K_{FT2}, which represent weighting factors of the time reference functions and are subsumed in the feedforward vector $\underline{S}_T$:

$$\underline{S}_T = \begin{bmatrix} K_{FT0} & K_{FT1} & K_{FT2} \end{bmatrix} \tag{15}$$

The introduction of

$$\underline{K}_T = \begin{bmatrix} k_{T1} & k_{T2} & k_{T3} & k_{T4} & k_{T5} \end{bmatrix} \tag{16}$$

as the feedback vector leads to the following extended state space representation of (11):

$$\begin{aligned} \dot{\underline{x}}_T &= (\underline{A}_T - \underline{B}_T\underline{K}_T)\underline{x}_T - \underline{B}_T\underline{S}_T\underline{w}_T \\ y_T &= \underline{C}_T\,\underline{x}_T \end{aligned} \tag{17}$$

For the determination of the elements of $\underline{S}_T$ the equations (14) and (15) are transformed into frequency domain:

$$\begin{aligned} \underline{x}_T &= (s\underline{I} - \underline{A}_T + \underline{B}_T\underline{K}_T)^{-1} \cdot \underline{B}_T\underline{S}_T \begin{bmatrix} 1 \\ s \\ s^2 \end{bmatrix} w_T(s) \\ y_T &= \underline{C}_T \cdot (s\underline{I} - \underline{A}_T + \underline{B}_T\underline{K}_T)^{-1} \cdot \underline{B}_T\underline{S}_T \begin{bmatrix} 1 \\ s \\ s^2 \end{bmatrix} w_T(s) \end{aligned} \tag{18}$$

Based on (18) the overall transfer function of the system can be determined:

$$\frac{y_T}{w_T} = \underbrace{\frac{b_0 + \ldots + b_{n-2}s^{n-2}}{N(s)}\left(K_{FT0} + K_{FT1}s + K_{FT2}s^2\right)}_{\widetilde{G}_T} \tag{19}$$

An advantageous behaviour can be achieved, if the coefficients in (20) are

$$\begin{aligned} \widetilde{G}_T(s) &= \frac{\widetilde{b}_0(K_{FTi}) + \widetilde{b}_1(K_{FTi})s + \widetilde{b}_2(K_{FTi})s^2 + \ldots}{a_0 + a_1 s + a_2 s^2 + \ldots} \\ a_0 &= \widetilde{b}_0(K_{FTi}) \\ a_1 &= \widetilde{b}_1(K_{FTi}) \\ a_2 &= \widetilde{b}_2(K_{FTi})\ldots \end{aligned} \tag{20}$$

which means a steady state behaviour concerning $\dot{\varphi}_T$ and the derivatives $\ddot{\varphi}_T$ and $\dddot{\varphi}_T$.

All elements of $\underline{S}_T$ depend on the system parameters like actual ladder length, raising angle, turning angle etc. Due to these dependencies, an adaptation of the feedforward gains in relation to the actual values of the different system parameters is achieved. The evaluated feedforward gains are also dependent on the gains of the feedback loop. To compensate external disturbances a state feedback control is designed based on pole assignment according to

$$det(sI - A_T + B_T K_T) \equiv P_T(s) \qquad (21)$$

The evaluation of the left side of (21) leads to a polynomial of fifth order, whose coefficients depend on the gains of the feedback vector:

$$\begin{aligned} det(sI - A_T + B_T K_T) &= s^5 + a_4(k_{T5}, k_{T3}, k_{T1})s^4 + \ldots \\ &\ldots + a_3(k_{T5}, \ldots, k_{T1})s^3 + a_2(k_{T5}, \ldots, k_{T1})s^2 + \ldots \\ &\ldots + a_1(k_{T4}, k_{T3}, k_{T1})s + a_0(k_{T1}) \end{aligned} \qquad (22)$$

The polynomial $p_T(s)$ contains the desired poles of the closed loop system:

$$\begin{aligned} p_T(s) &= (s - p_h)(s - (p_{\alpha_x,r} + ip_{\alpha_x,i}))(s - (p_{\alpha_x,r} - ip_{\alpha_x,i}))\cdot \\ &\ldots\cdot(s - (p_{v_y,r} + ip_{v_y,i}))(s - (p_{v_y,r} - ip_{v_y,i})) \end{aligned} \qquad (23)$$

The open loop pole location is one real pole reflecting the delay time of the hydraulic system and two pairs of conjugate complex poles reflecting the horizontal flexion (real part: $p_{v_y,r}$, imaginary part: $p_{v_y,i}$) and the torsion (real part $p_{\alpha_x,r}$, imaginary part: $p_{\alpha_x,i}$). The separated access on the real and imaginary parts allows a separated adjustment of the frequency and damping attributes of the regarded elasticity degree of freedom. The feedback gains are resulting from solving the equation system (21)-(23) and depending, like the feedforward gains, on different system parameters (ladder length, raising angle etc.).

The output of the feedforward part results in

$$u_{T,ff} = K_{FT0}\dot{\varphi}_{Tref} + K_{FT1}\ddot{\varphi}_{Dref} + K_{FT2}\dddot{\varphi}_{Tref} \qquad (24)$$

and that of the feedback loop in

$$u_{T,fb} = k_{T1}\dot{\varphi}_T + k_{T2}\alpha_x + k_{T3}\dot{\alpha}_x + k_{T4}v_y + k_{T5}\dot{v}_y \qquad (25)$$

The control signal of the control module results from the difference of (24) and (25) (Fig. 5).

V. Measurement Results

In the following the efficiency of the proposed control concept will be demonstrated by measurement results. Fig. 6 and Fig. 7 are illustrating the reaction on a manual excitation of the ladder. The oscillations of the flexion as well as the oscillations of the torsion are low damped only (Fig. 6a and Fig. 7a). The activation of the control module increases the damping significantly (Fig. 6b and Fig 7b).

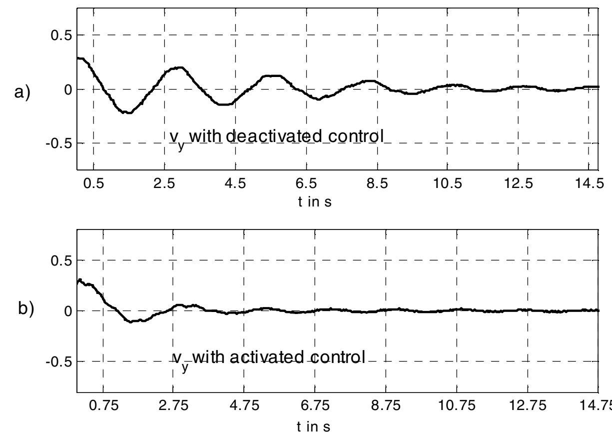

Figure 6. Horizontal flexion after a manual excitation of the ladder set with activated and deactivated control

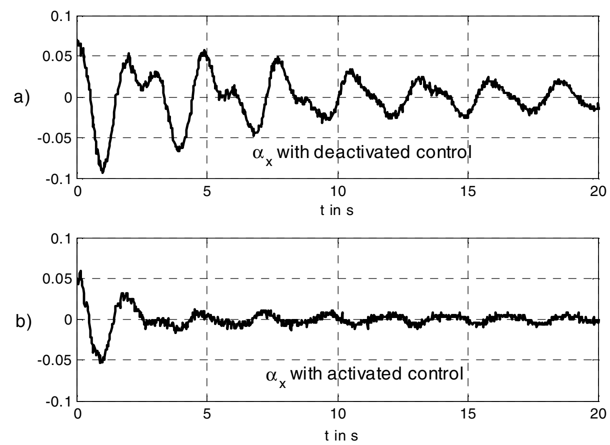

Figure 7. Torsion after a manual excitation of the ladder set with activated and deactivated control

In Fig. 8 and Fig. 9 a turning movement of the ladder with deactivated control (Fig.8) and with activated control (Fig.9) is shown. Again swaying of the cage can be significantly reduced by the control. The flexion is in a interval of a few centimeters and is also well damped compared with the behaviour in the case of deactivated control.

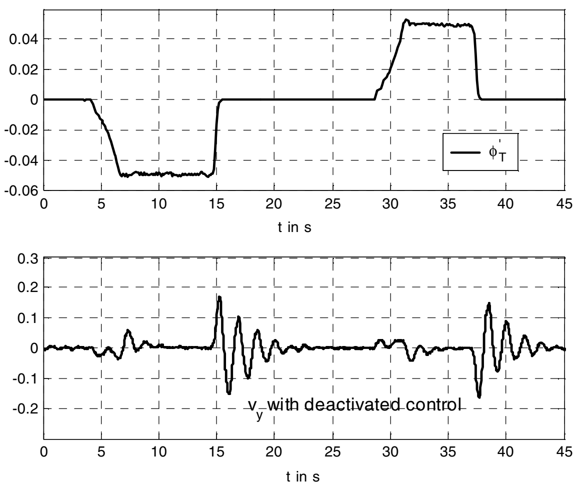

Figure 8. Turning of the ladder with deactivated control – horizontal flexion

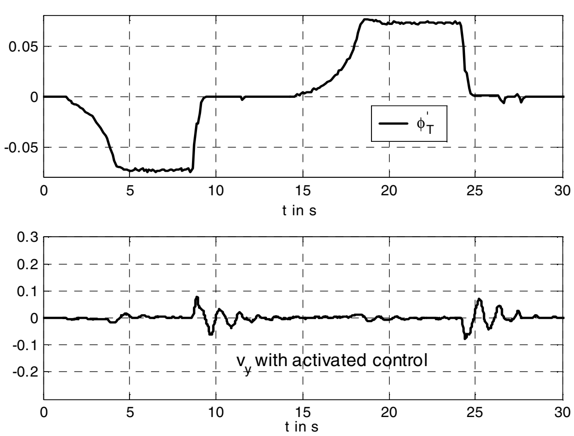

Figure 9. Turning of the ladder with activated control – horizontal flexion

VI. CONCLUSION

The paper presents a trajectory tracking control for the purpose of damping the swaying of the cage for a fire rescue turntable ladder based on a decentralized control concept. The developed control algorithm was transferred on a micro-controller system with limited computational power. Therefore the dynamic model of the system, was derived by applying the Lagrange formalism resulting in a multi-body system with elasticities. The dynamic model was splitted up in two sub-models. Based on these two sub-models a control module consisting of an adaptive feedforward part and a adaptive state feedback loop as a gain scheduled approach was developed for each sub-model. Therefore, changing system parameters like varying ladder lengths and raising angles can be considered by the control strategy. The efficiency of the decentralized control strategy is illustrated in several measurement plots. Regarding the arm elasticity of the ladder, a efficient damping of flexion and torsion can be realized with the activated control after a manual excitation as well as during the movement of the ladder. The control is already realized in the new generation of fire rescue turntable ladders of an important manufacturer of fire turntable ladders and belongs to the standard equipment of the vehicle.

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