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MECHATRONICAL DESIGNED CONTROL OF FIRE-RESCUE TURNTABLE-LADDERS AS FLEXIBLE LINK ROBOTS

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Abstract: Turntable ladders with a maximum ladder length of up to 30 m can be assumed as 3-axes flexible link robots. For operation at maximum velocity which is an important factor for turntable ladders as fire rescue vehicles, and to realize an automated teach-in operation mode, a trajectory tracking control based on a decentralized control approach has been developed by a mechatronical design. The implementation guarantees active oscillation damping of all ladder movements. The recently presented active control system will be in future a standard equipment for the new generation of fire turntable ladders of the cooperation partner IVECO Magirus company. *Copyright © 2002 IFAC*

Keywords: mechatronical design, flexible arm robot, trajectory tracking control, decentralized control, mechanical manipulators.

1. INTRODUCTION

Turntable ladders are large scale robots with two rotatory axes and one translational axis mounted on a truck. The manipulator motion possibilites are defined by spherical coordinates (fig. 1). The translational coordinate is identical to telescoping the ladder and described by the variable *l*. Raising of the ladder set is defined by the angle variable j_R, turning the ladder by the variable j_T. The operating range is for *l* 9.18 m up to 30 m, for j_R -15^0 up to 75^0 and for j_T 360^0.

State of the art in control of turntable ladders is a combined electronic and hydraulic control with rate limiters without feedback loop. Because the ladder weight should be minimized in order to achieve large workspaces especially at low raising angles, the stiffness of the ladder is limited which results in swaying of the ladder in case of large ladder lengths. Until now, this swaying was avoided by reducing the achievable velocity depending on the ladder length and raising angle. Aim is now to increase the velocities significantly which is important for the use of fire turntable ladders as emergency vehicles. In addition for the fast evacuation of a large number of people a fully automated tech-in operation mode should be developed. The idea is to integrate a trajectory tracking control with abilities to damp the upcoming oscillations into the electronic and hydraulic vehicle control (fig. 2).

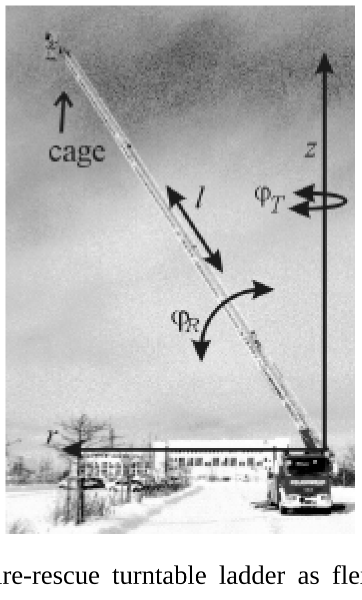

Fig. 1. Fire-rescue turntable ladder as flexible link robot.

Therefore, the algorithm for the trajectory control has to be embedded on the electronic control system of the vehicle which is realized on CAN-bus connected microcontrollers. As sensor data the trajectory control need the position of the ladder in the workspace as well as a signal which reflects the ladder oscillations. The ladder oscillations are detected via the ladder flexion in horizontal direction w_h and vertical direction w_v by a set of four strain gauges in the lowest ladder part. Ladder length l, turning angle φ_T, and raising angle φ_R are evaluated by encoder measurements. By real differentiation of these signals the related velocities are determined. The turntable ladder is driven by a hydraulic drive system. The combustion engine is connected with the feed pump for the hydraulic system. The swept volume of the pump is controlled by a underlying load sensing flow rate control which is affected by the servo valve. That means the hydraulic drives for the different movement directions are influenced by the input voltages u_{StT}, u_{StR}, u_{StL} of the servo valves.

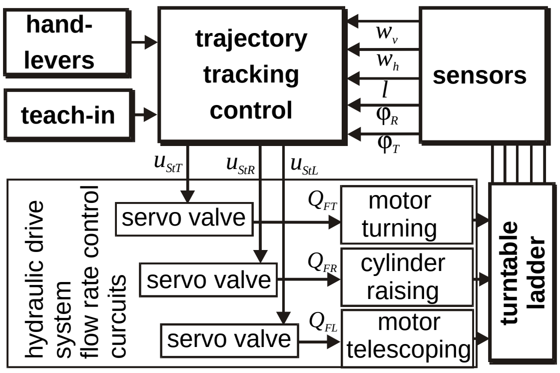

Fig. 2. Trajectory control embedded in the vehicle.

2. AUTOMATION CONCEPT

Input for the trajectory tracking control are the commands of the operator at the hand-levers or the target positions which has been saved in the teach-in mode in a former running of the ladder. The target position or the target velocity is then the input information for the following trajectory generation module (fig. 3). This module generates the time reference functions for the different movement directions considering the kinematic limitations of the system (e.g. maximum velocity, maximum acceleration and maximum jerk) (Sawodny *et al.* 2001). These reference functions are then given on the control module which are connected to a specific movement direction. In the given example this is the turning, raising and telescoping of the ladder.

For the now following design of the control modules for turning, raising and telescoping a dynamic model of the turntable ladder system has to be derived. In spite of deriving a coupled model for the whole system (Schneider *et al.* 1996, de Wit *et al.* 1997) the idea is to derive dynamical models which represent the relevant part for the dynamics in the considered movement direction (Sawodny *et al.* 1999a). The higher nonlinear terms describing the coupling of the system in case of synchronous movement of raising, telescoping and turning the ladder are neglected. Advantage of this decentralized control concept is a compact control algorithm which is able to be transferred on an microcontroller system as the desired hardware platform. At the example of raising the ladder the derivation of the dynamic model and the design of the control will be explained in detail and the results discussed at specific measurements.

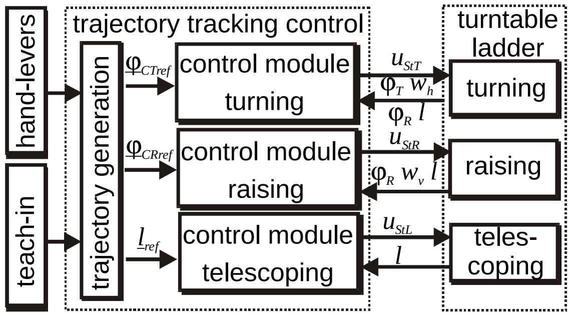

Fig. 3. Structure of the trajectory tracking control.

3. DYNAMIC MODEL

For the derivation of the dynamic model describing raising the ladder the position variable for the raising angle φ_R is introduced (fig. 4). The vertical flexion of the ladder is represented by the variable w_v. As control variable then the raising angle φ_{CR} in respect to the cage position can be defined by

$$\varphi_{CR} = \varphi_R - \frac{w_v}{l} \tag{1}$$

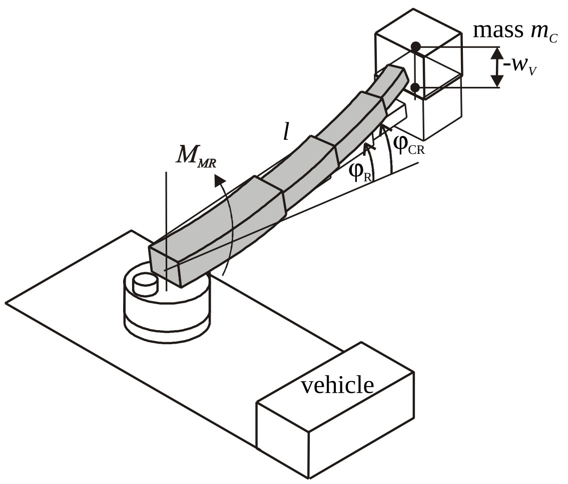

Fig. 4. Modeling the mechanical structure of the ladder.

For the description of the dynamic behavior of raising the ladder a flexible multibody system is chosen (Maier *et al.* 2000). The flexion of the ladder is modeled by superposition of the flexion lines of the four ladder parts (fig. 5) and repatriated to a equivalent stiffness. Therefore, the flexion of the ladder has to be calculated by

$$\frac{d^2 w_v}{dl^2} = -\frac{F \cdot l}{E \cdot I_{l-l}} \qquad (2)$$

F is the interacting force, E the modulus for tension and $I_{l\text{-}l}$ the geometrical moment of inertia regarding to the direction l of the ladder. Because the ladder set consists of four parts, the different moment of inertia of each ladder part has to be considered, if the flexion line has been calculated by twice integration of (2). Result is a polynomial function of third order depending on the ladder length l

$$w_v = f(l) \cdot F \qquad (3)$$

That means, the equivalent stiffness then can be assumed as

$$c_{RL}(l) = \frac{1}{f(l)} \qquad (4)$$

Next step is to cut the ladder into two parts (fig. 5) and to introduce equivalent masses for them. As a simplification the equivalent masses m are both set to half of the complete ladder set mass m_L.

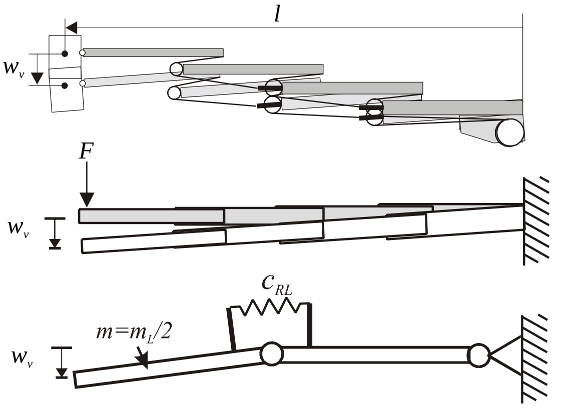

Fig. 5. Ladder set and flexible multibody representation.

Last step before deriving the equations of motion is the evaluation of the mass moment of inertia for the ladder. Therefore the ladder is assumed as a beam with certain cross sectional dimensions. Result is again a function $J_R(l)$ depending on the ladder length l for the mass moment of inertia of the ladder. Lastly the equations of motion derived by the Lagrange formalism for the given system considering the movement direction of raising the ladder are

$$\begin{aligned} (J_R(l) + m_C l^2)\ddot{\varphi}_R - ml \cdot \ddot{w}_v - ml \cdot g \cos\varphi_R &= M_{MR} - b_R \dot{\varphi}_R \\ -ml \cdot \ddot{\varphi}_R + m \cdot \ddot{w}_v + b_{RL} \cdot \dot{w}_v + c_{RL}(l) \cdot w_v &= 0 \end{aligned} \qquad (5)$$

In the following the terms which consider the gravity effects are neglected. M_{MR} is the actuating torque of the hydraulic cylinder on the ladder. m_C is the resulting cage mass and g the gravity constant. b_R is the viscous friction coefficient. The first equation of (5) describes mainly the raising kinematics concerning the lower ladder part, whereas the reaction due to the flexion of the ladder is considered. The second equation of (5) is the equation of motion describing the ladder oscillations due to the ladder flexion. In the second equation beside the equivalent stiffness c_{RL} a damping coefficient b_{RL} for the ladder oscillation is introduced.

As hydraulic drive two hydraulic cylinders are mounted in the ladder gear. The reacting torque on the ladder can be described by the following equations (Wey *et al.*1999).

$$\begin{aligned} M_{MR} &= F_{cyl} d_b \cos\varphi_p(\varphi_R) \\ F_{cyl} &= p_{cyl} A_{cyl} \\ \dot{p}_{cyl} &= \frac{2}{bV_{cyl}}\left(Q_{FR} - A_{cyl}\dot{z}_{cyl}(\varphi_R, \dot{\varphi}_R)\right) \\ Q_{FR} &= K_{PR} u_{StR} \end{aligned} \qquad (6)$$

F_{cyl} is the force of the hydraulic cylinders on the piston rod, p_{cyl} is the pressure in the cylinder, A_{cyl} the cross sectional area of the cylinder, b the compressibility of the oil, V_{cyl} the volume of the cylinder, Q_{FR} the flow rate, and K_{PR} the constant which describes the connection between flow rate and input voltage of the valve. Dynamical effects of the underlying flow rate control are neglected. As the relevant cylinder volume for the oil compression half of the complete cylinder volume is assumed. $z_{cyl}, \dot{z}_{cyl}$ are position and velocity of the piston rod. The angle position φ_R and velocity $\dot{\varphi}_R$ as well as the projection angle φ_p depend on the position and velocity of the piston rod $z_{cyl}, \dot{z}_{cyl}$ (fig. 6). The hydraulic cylinders are mounted in the ladder gear. The piston rods of the cylinders are fixed at the ladder set. The distances d_a and d_b can be evalutated out of technical data of the ladder. This results then in the following equation describing the dependency between the piston rod position z_{cyl} and the raising angle φ_R:

$$z_{cyl} = \sqrt{d_a^2 + d_b^2 - 2 d_b d_a \cos(\varphi_R + \varphi_0)} \qquad (7)$$

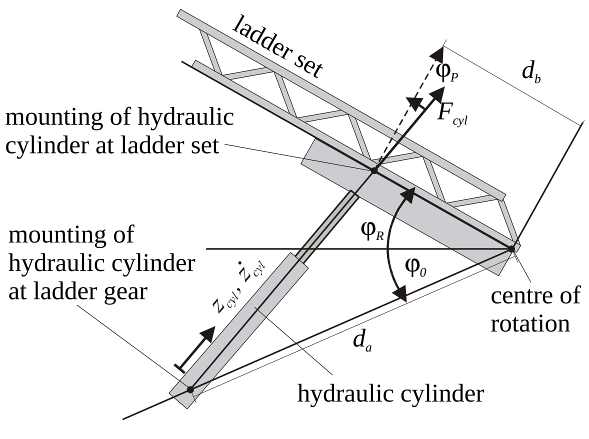

Fig. 6. Kinematic structure of raising the ladder via hydraulic cylinders.

The decomposition of (7) for the variable j_R leads to the equation (8). In addition the relation between piston rod velocity $\dot{z}_{cyl}$ and raising angle velocity $\dot{j}_R$ is necessary.

$$j_R = \arccos(\frac{d_a^2 + d_b^2 - z_{cyl}^2}{2 d_a d_b}) - j_0 \qquad (8)$$

$$\dot{j}_R = \frac{\sqrt{d_a^2 + d_b^2 - 2 d_b d_a \cos(j_R + j_0)}}{d_b d_a \sin(j_R + j_0)} \dot{z}_{cyl} \qquad (9)$$

For the calculation of the actuating torque on the ladder set the projection angle is needed

$$\cos j_p = \frac{d_a \sin(j_R + j_0)}{\sqrt{d_a^2 + d_b^2 - 2 d_b d_a \cos(j_R + j_0)}} = \frac{h_1}{h_2} \qquad (10)$$

In the following instead of the trigonometric functions of (10) the auxiliary variables h_1 and h_2 are used. For the control design the equations (5) to (10) are transformed into linear state space representation. The nonlinearities of the trigonometric terms and the coupling to the other movement directions especially of telescoping the ladder are interpreted as varying system parameters. This leads to the following state space representation

$$\begin{aligned} \dot{\underline{x}}_R &= \underline{A}_R \underline{x}_R + \underline{B}_R \underline{u}_R \\ \underline{y}_R &= \underline{C}_R \underline{x}_R \end{aligned} \qquad (11)$$

with:

$$\underline{x}_R = [w_v \quad \dot{w}_v \quad j_R \quad \dot{j}_R \quad M_{MR}]^T \qquad \underline{u}_R = u_{StR};$$

$$\underline{y}_R = j_{CR} \qquad \underline{C}_R = \begin{bmatrix} -\frac{1}{l} 0 & 1 & 0 & 0 & 0 \end{bmatrix}$$

$$\underline{A}_R = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ a_{21} & a_{22} & 0 & -\frac{b_R l}{J_{RC}} & \frac{l}{J_{RC}} \\ 0 & 0 & 0 & 1 & 0 \\ -\frac{c_{RL} l}{J_{RC}} & -\frac{b_{RL} l}{J_{RC}} & 0 & -\frac{b_R}{J_{RC}} & \frac{1}{J_{RC}} \\ 0 & 0 & 0 & -\frac{2A_{cyl}^2}{b V_{cyl}} \frac{(d_b h_1)^2}{h_2^2} & 0 \end{bmatrix}$$

$$J_{RC}(l) = J_R(l) + m_C l^2 - m l^2$$

$$a_{21} = -\frac{c_{RL}(J_{RC} + m l^2)}{m J_{RC}} \qquad a_{22} = -\frac{b_{RL}(J_{RC} + m l^2)}{m J_{RC}}$$

$$\underline{B}_R = [0 \quad 0 \quad 0 \quad 0 \quad \frac{2 A_{cyl} d_b h_1 K_{PR}}{V_{cyl} b h_2}]^T \qquad (12)$$

The resulting system is now assumed as linear in the small signal behavior concerning to the defined state vector. The nonlinear coupling terms of higher order due to synchronous movement of telescoping and raising are neglected.

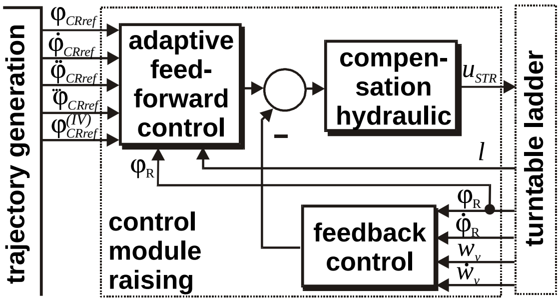

Fig. 7. Control module structure.

4. CONTROL DESIGN

The control module is configured as a combination of adaptive feedforward control and a robust designed feedback loop (fig. 7). In addition the static nonlinearities of the servo valve (hysteresis and death zone) has to be compensated by the inverse function in the block for the compensation of the hydraulics.
The design of the feedforward control is based on the idea of plant inversion. Input of the feedforward control block is the vector of the reference functions for the cage raising angle j_{CR}.

$$\underline{w}_R = [j_{CRref} \quad \dot{j}_{CRref} \quad \ddot{j}_{CRref} \quad \dddot{j}_{CRref} \quad j_{CRref}^{(IV)}]^T \qquad (13)$$

The feedforward control is defined by the following matrix consisting of the feedforward gains K_{FR0} to K_{FR4}.

$$\underline{S}_R = [K_{FR0} \quad K_{FR1} \quad K_{FR2} \quad K_{FR3} \; K_{FRA4}] \qquad (14)$$

The state space representation of the considered system according to (11) is extended by the input vector, the feedforward matrix and the feedback loop. This leads to the following representation of the system.

$$\begin{aligned} \dot{\underline{x}}_R &= (\underline{A}_R - \underline{B}_R \underline{K}_R) \underline{x}_R + \underline{B}_R \underline{S}_R \underline{w}_R \\ \underline{y}_R &= \underline{C}_R \underline{x}_R \end{aligned} \qquad (15)$$

where the feedback matrix is defined by

$$\underline{K}_R = [k_{1R} \quad k_{2R} \quad k_{3R} \quad k_{4R} \quad k_{5R}] \qquad (16)$$

For the feedforward control design the input vector $\underline{w}_R$ is transformed into frequency domain.

$$\underline{W}_R(s) = \begin{bmatrix} j_{CRref}(s) \\ s \cdot j_{CRref}(s) \\ s^2 \cdot j_{CRref}(s) \\ s^3 \cdot j_{CRref}(s) \\ s^4 \cdot j_{CRref}(s) \end{bmatrix} = \begin{bmatrix} 1 \\ s \\ s^2 \\ s^3 \\ s^4 \end{bmatrix} j_{CRref}(s) = \underline{F}(s) \cdot j_{CRref}(s) \qquad (17)$$

The transfer function for the control variable j_{CR} is then

$$G(s)=\frac{j_{CR}}{j_{CRref}} = \underline{C}_R(s\underline{I}-\underline{A}_R+\underline{B}_R\underline{K}_R)^{-1}\underline{B}_R\underline{S}_R\underline{F}(s) \tag{18}$$

The transfer function $G(s)$ has the following form illustrating the dependency of the feedforward gains to the nominator coefficients

$$\frac{j_{CR}}{j_{CRref}}=\frac{...b_2(K_{FRi})\cdot s^2+b_1(K_{FRi})\cdot s+b_0(K_{FRi})}{...a_2\cdot s^2+a_1\cdot s+a_0} \tag{19}$$

An ideal dynamic behavior is achieved, if the coefficients fulfill the following conditions

$$b_i=a_i \quad \text{for} \quad i=0..n-1 \tag{20}$$

where n is the system order. In time domain this corresponds with the following differential equation

$$\begin{aligned}&a_5 j_{CR}^{(V)}+a_4 j_{CR}^{(IV)}+a_3\dddot{j}_{CR}+a_2\ddot{j}_{CR}+a_1\dot{j}_{CR}=\\&b_6(K_{FR4})j_{CRref}^{(VI)}+b_5(K_{FR4},K_{FR3})j_{CRref}^{(V)}+\\&b_4(K_{FR4},K_{FR3},K_{FR2})j_{CRref}^{(IV)}+\\&b_3(K_{FR3},K_{FR2},K_{FR1})\dddot{j}_{CRref}+\\&b_2(K_{FR2},K_{FR1},K_{FR0})\ddot{j}_{CRref}+b_1(K_{FR1},K_{FR0})\dot{j}_{CRref}\\&+b_0(K_{FR0})j_{CRref}\end{aligned} \tag{21}$$

Because the order of the left hand part of the differential equation is lower than the part of the right hand side, the complete compensation by the feedforward control can not be achieved. But the comparison of the coefficients i=0..4 leads to an advantageous behavior in combination with the feedback loop. Solving the resulting linear equation system (20) for the feedforward control gains leads to

$$\begin{aligned}K_{FR0}&=k_{3R}\\K_{FR1}&=\frac{A_{cyl}h_2d_b}{K_{PR}h_1}+k_{4R}\\K_{FR2}&=\frac{b_RbV_{cyl}h_1}{2d_bA_{cyl}K_{PR}h_2}-\frac{m}{c_{RL}}k_{3R}+\frac{ml}{c_{RL}}k_{1R}\\K_{FR3}&=\frac{(J_{RC}+ml^2)c_{RL}bV_{cyl}h_1^2-2mA_{cyl}^2d_b^2h_2^2}{2c_{RL}A_{cyl}K_{PR}h_2h_1d_b}+\\&\frac{m(lk_{2R}-k_{4R})}{c_{RL}}+\frac{mb_{RL}(-lk_{1R}+k_{3R})}{c_{RL}^2}\end{aligned}$$

$$\begin{aligned}K_{FR4}&=\frac{-mc_{RL}b_RbV_{cyl}h_1^2+2mA_{cyl}^2d_b^2h_2^2b_{RL}}{2c_{RL}^2A_{cyl}K_{PR}h_2d_bh_1}+\\&\frac{2m^2(k_{3R}-lk_{1R})}{c_{RL}^2}+\frac{mb_{RL}(k_{4R}-lk_{2R})}{c_{RL}^2}+\\&\frac{mb_{RL}^2(lk_{1R}-k_{3R})}{c_{RL}^3}\end{aligned} \tag{22}$$

The feedforward control gains are depending on the system parameters K_{PR}, A_{cyl}, V_{cyl}, j_R, b, J_R, m, m_C, c_{RL}, b_{RL} , b_R, d_b, d_a. Due to this relation an adaptation of the feedforward control gains is achieved by gain scheduling depending on ladder length l and the raising angle j_R resulting in a trajectory tracking of the given reference functions for the cage angle position, velocity, acceleration, jerk and the derivation of the jerk. In addition changing technical data can be taken into account by changing the relevant parameter in the initialization file. This procedure is related to the design of a flatness based feedforward control (Fliess *et al.* 1991, Delaleau *et al.* 1998). But in fact, it is still applicable although the output is not a flat output and the considered system is characterized by zero dynamics. This may also been seen in the fact that the nominator can not compensate the denominator of the transfer function via the feedforward gains completely.

To compensate the parameter uncertainties and external disturbances, and to suppress the influence of the non feedforward compensated dynamics the system is completed by the robust designed feedback loop by pole assignment according to

$$\det(s\underline{I}-\underline{A}_R(l_0,j_{R0})+\underline{B}_R(l_0,j_{R0})\cdot\underline{K}_R)\equiv\prod_{i=1}^{n}(s-r_i) \tag{24}$$

The feedback loop is designed for the critical operating point which is in the given problem the lowest raising angle, the lowest cage mass and the largest ladder length. r_i are the desired poles of the closed loop system. The numerical values of the feedback gains designed for this working point are then set to constant and the robustness concerning stability of the system is simplified checked by evaluating the eigenvalues of the closed loop system in case of varying ladder length, cage mass, and raising angle in the interval $[l_{min}, l_{max}]$ and $[j_{Rmin}, j_{Rmax}]$.

5. MEASUREMENT RESULTS

The efficiency of the control concept is illustrated in several measurement plots. Fig. 8 shows the reaction with deactivated control. Due to the excitation of the ladder set the resulting oscillations are fairly low damped. The activated control damps the oscillations within a few seconds. The effect is more evidently at the measurement of the derivation of the flexion.

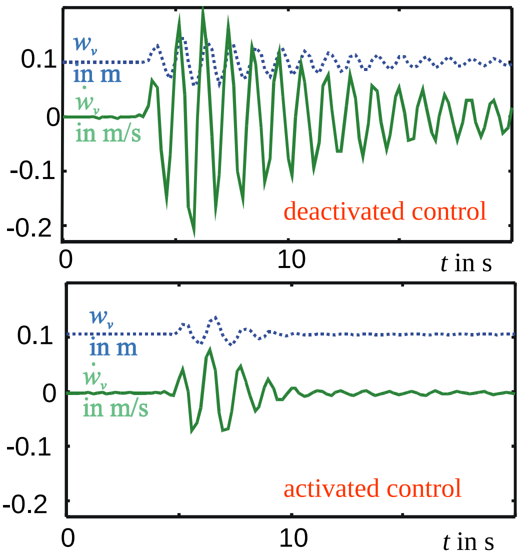

Fig. 8. Manually excitation of the ladder set. Upper: deactivated control; Lower: activated control (l=20 m; m_C=230 kg).

Next, a raising movement of the ladder is shown in fig. 9 with activated control. The upcoming oscillations are very low. The flexion is beyond a few centimeters and the oscillations reflecting the derivative of the flexion signal are damped within 5 sec.

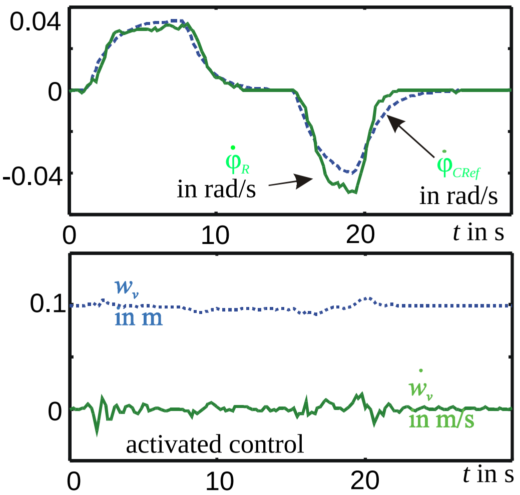

Fig. 9. Movement of the ladder by raising the ladder set with activated control. (l=20 m; m_C=230 kg)

6. CONCLUSION

The presented control for a turntable ladder as a multi-axes robot system, moving in spherical coordinates, is a modelbased decentralized control concept. As a representation for the dynamic behavior a flexible multibody system for the mechanical part is derived connected with the specific movement directions. The mechatronical design includes the dynamic behavior of the hydraulic drive system in the model equations. Because of low computational power of the control hardware the dynamic model neglects the higher nonlinear coupling terms in case of synchronous movement of the axes. The control itself consists of an adaptive feedforward control module based on the inverse dynamics and a state feedback loop. The feedforward module takes varying ladder lengths and raising angle by adaptation of the feed forward control gains into account. The feedback controller is designed on robustness criterions for the most critical operation point. The efficiency of the control is illustrated in several measurement plots and demonstrate the advantages of the new control in comparison to the state of the art. Result are two times higher velocities and complete damping of the arising oscillations. The control is realized on the new generation of fire rescue turntable ladders of IVECO Magirus and will be in future belonging to the standard equipment.

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