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A Trajectory Tracking Control with Disturbance-Observer of a Fire-Rescue Turntable Ladder

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Abstract— **Modern fire-rescue turntable ladders are constructed in a lightweight mode to increase their maximum operation velocities, maximum length, and outreach respectively. Hence, the ladder has a limited stiffness and will be more and more subject to oscillations of deflection along with dominant higher modes. This paper deals with the active oscillation damping of such ladders. For active oscillation damping by feedback the ladder is equipped with a gyroscope additionally to strain gauges. A Luenberger-type observer is used to separate the fundamental oscillation from the high-frequency modes. Due to computational efforts and measurement noise, only the estimated state of the fundamental oscillation is disposed for feedback. The proposed control approach allows damping the fundamental oscillations and asymptotically stabilizing the system around a reference trajectory. Measurement results from the IVECO DLK 55 CS fire-rescue turntable ladder validate the efficiency of the proposed control structure.**

I. INTRODUCTION

In this paper a new control strategy for active damping of a fire-rescue turntable ladder is presented. Lightweight construction is applied to modern turntable ladders such as the IVECO DLK 55 CS (cf. Fig. 1) in order to increase their maximum operation speeds, maximum ladder's length, and its horizontal outreach. Hence, the ladder has a very limited stiffness and will be more and more subject to oscillations of deflection along with dominant overtones. The IVECO DLK 55 CS turntable ladder is characterized by a maximum extension of the ladder set of $L = 53.2\,\mathrm{m}$ and a maximum outreach of 22 m (at $\varphi_{A,min} = 68\,°$) respectively. The raising angle covers a space of $\varphi_A = [-12\,° \ldots 75\,°]$. In the vertical plane the ladder is driven by hydraulic cylinders, which provide a maximum angular velocity of $\dot{\varphi}_A = 3\,°/\mathrm{s}$. The ladder set itself has a dead load of 4820 kg. The cage has a dead load of 200 kg and can carry a maximal pay load of 300 kg, which corresponds to three fire fighters with full equipment.

The ladder has not just one eigenfrequency (1st mode), but a second one (2nd mode) within the frequency spectrum of the hydraulic actuator (see Fig. 5). Fortunately, the oscillation of the second eigenfrequency has a high natural damping. For safety reasons and performance improvement the task for the control is damping at least the fundamental oscillation without exciting high-frequency oscillations and realizing higher velocities at the same time.

The ladder is equipped with two measurement systems as shown in Fig. 1. Strain gauges are mounted at the lower

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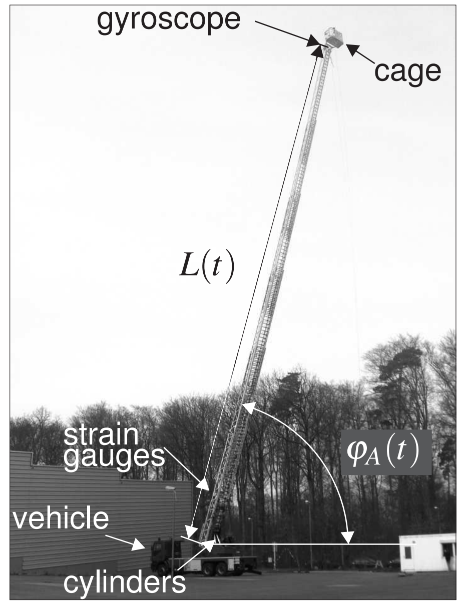

Fig. 1. Turntable ladder: IVECO Magirus DLK 55 CS at a raising angle of $\varphi_A \approx 68\,°$ and a ladder's length of $L = 53.2\,\mathrm{m}$)

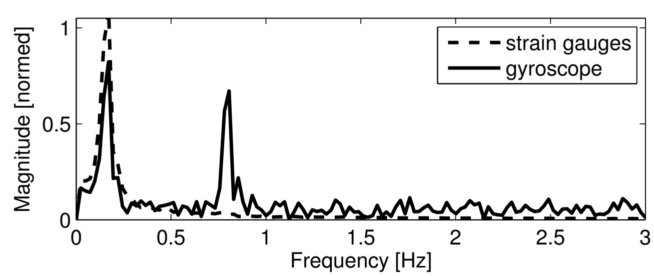

Fig. 2. Discrete fast fourier transform of a step response with a ladder's length of $L = 53.2\,\mathrm{m}$

(fixed) end of the ladder and at the tip (upper end) a gyroscope is installed. As shown in Fig. 2 the gyroscope is more sensitive to the second mode than the strain gauges, which is caused by construction.

In recent years the task of active oscillation damping of the turntable ladders with the length up to 30 m was considered in various publications. In [6], [7], [9] a trajectory tracking control based on a dynamic model of the ladder and decentralized control strategy has been developed. In these works the information about the system state is limited to the erecting angle and the strain gauges at a point close to the hub. By using these signals, a feedback was designed within

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the framework of a multi-body system model. The designed controller is capable of reducing the swaying concerning the fundamental oscillation effectively. However, it cannot damp undesirable high–frequency oscillations (overtones), which become more dominant especially for a large length of the ladder.

In [11] the turntable ladder was considered as a flexible manipulator model with passive joints based on the Euler–Bernoulli beam concept. The feedback design is based on the Galerkin approximation. In [12], [13] a similar approach is applied on a Timoshenko model of a beam. For the realization of the designed feedback law it is proposed to use the dynamical observer needing more computational power of a micro controller.

An Euler-Bernoulli model of a beam with a point mass at the end is considered in [5]. Based on the analytical eigenfunctions, the modal description of the plant is constructed, considering only the first and the second mode. On this basis a feedback law is derived. The plant is equipped with gyroscopes in addition to the strain gauges. Hence an observer is unnecessary because all states are determined by solving a system of algebraic equations of the two measurements.

In the present contribution we shall use the mathematical model of a multi-body system. The differential equations of motion are derived by applying the Lagrangian formalism ([2]). The dynamics of the hydraulic actuators are approximated by a 1st-order transfer function. The design of the feedforward and feedback loop is formulated based on simplifications of the dynamical model. Because the dominant high-frequency oscillations will cause problems if the feedback loop will be closed. A disturbance observer is proposed which separates the fundamental oscillation from the overtones. Due to the fact that there are two measurements, the observer is of MIMO-type. Therefore, the observer is designed by using the observer canonical form for MIMO-systems ([3] and [4]). The control concept has been applied to a IVECO DLK 55 CS fire–rescue turntable ladder which is driven by a micro controller working with fix-point arithmetic. This has to be taken into account during the development of the control strategy and observer design.

In Section II a mathematical model of the plant is derived. An overview on the control structure is given at the beginning of Section III. Section III-A deals with the design of the feedforward loop and Section III-B deals with the feedback loop. To eliminate the overtones from the measurement signals a disturbance observer is presented in Section IV. Measurement results are obtained and analyzed in Section V to show the effectiveness of the proposed control. In Section VI concluding remarks are given and aspects of future work are discussed.

II. Dynamic Model

The dynamic model of the turntable ladder is derived by using the Lagrange formalism on a multi-body system with spring and damper elements (cf. Fig. 3). Where the ladder set (including the cage) and the vehicle are approximated by two equivalent masses ($m_h \ldots$ vehicle, $m_l \ldots$ ladder set and cage). The arm elasticity is approximated by spring-damper elements. Thus, the ladder is considered a homogeneous leaf spring with length L, stiffness coefficient $c_v(L)$, damping coefficient $d_v(L)$, and point mass m_l at the end of it. The length of the ladder set can be extended or reduced by telescoping which is very slow. Though the plant is considered a time-invariant system, but the parameters depending on L are updated with every change in the ladder's length. The deflection at the end of the ladder is named $v_z(t)$. The turntable ladder has more degrees of freedom, e.g. the turning motion of the hub $(\varphi_R(t))$ and the horizontal deflection at the ladder's end $v_y(t)$. However, the focus of the paper is on the motion in the vertical plane.

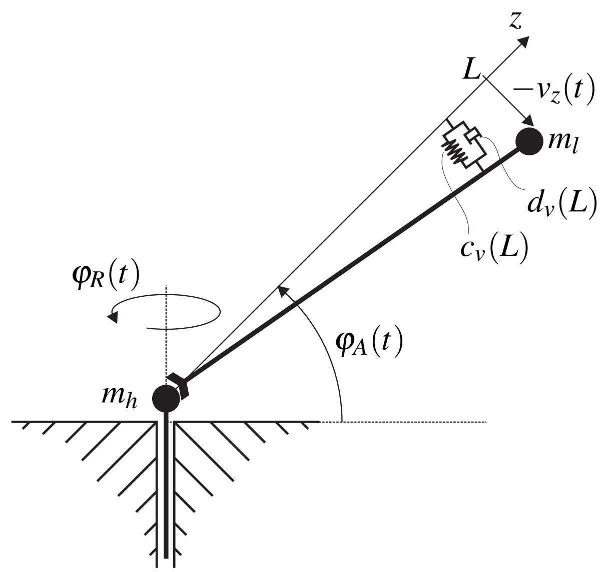

Fig. 3. Multi-body system model with elastic degree of freedom $(v_z(t))$ as model for feedforward control design

The dynamic equations of motion were derived by computing the kinetic energy T and the potential energy U of the system and then forming the Lagrangian $L = T - U$. By satisfying the Lagrange's equation

$$\frac{d}{dt}\frac{\partial L}{\partial \dot{\vartheta}_i} - \frac{\partial L}{\partial \vartheta_i} = u_i \quad i = 1, \ldots, n,$$

the dynamic model is obtained. The variables $\vartheta_i(t)$ are the generalized coordinates or the degrees of freedom mentioned before $(\varphi_A(t)\,,\, \varphi_R(t),\, v_z(t),\, v_y(t))$, respectively.

$$\underline{M}\,\ddot{\underline{\vartheta}} + \underline{D}\,\dot{\underline{\vartheta}} + \underline{C}\left(\dot{\underline{\vartheta}}, \underline{\vartheta}\right) + \underline{K}\,\underline{\vartheta} + \underline{G}\left(\underline{\vartheta}\right) = \underline{u} \qquad (1)$$

Where $\underline{M}$ is the matrix of inertia, $\underline{D}$ is the matrix of damping factors, $\underline{C}$ is the vector of Coriolis and Centripetal forces, $\underline{K}$ is the matrix of equivalent stiffness coefficients, $\underline{G}$ is the vector of gravitational forces, and $\underline{u}$ is the vector of the driving forces and torques.

The oscillation of deflection will be considered in the vertical plane, whereas the action of gravity on the concentrated mass is neglected. Since the mathematical model is linear for small deflections and steady state solution taking into account the action of gravity and prestressing of the ladder can be always subtracted for the task of stabilization. Hence, for the controller design from (1) the equation important for the planar vertical motion is extracted

$$\begin{aligned} 0 &= m_l \ddot{v}_z(t) + m_l L \ddot{\varphi}_A(t) + d_v \dot{v}_z(t) + c_v v_z(t) + \\ &+ \underbrace{\frac{\left(L\dot{\varphi}_R(t) + \dot{v}_y(t)\right)^2}{2L} \sin\left(2\varphi_A(t) + \frac{2}{L} v_z(t)\right)}_{p_R(t)} \end{aligned} \tag{2}$$

Even though the term of the influences by the Coriolis-force does not bother during the controller design, it is neglected ($p_R(t) \equiv 0$) for the sake of simplicity. As the rotary motion is quite slowly ($\dot{\varphi}_R \ll 1$) and this motion axis is actively damped by a separate controller ($\dot{v}_y \ll 1$), as well. For the sake of less computational costs this simplification is quite convenient.

The dynamics of the actuator is approximated by a 1st-order transfer function, because there is a subsidiary flow control of the cylinders' hydraulics. The nonlinear dc–gain depending on the erecting angle $k_A(\varphi_A)$ is determined from the construction.

$$\ddot{\varphi}_A = -\frac{1}{\tau_A}\dot{\varphi}_A + \frac{k_A(\varphi_A)}{\tau_A} u \tag{3}$$

By choosing the state vector as

$$\underline{x}(t) = \begin{bmatrix} \varphi_A(t) & \dot{\varphi}_A(t) & v_z(t) & \dot{v}_z(t) \end{bmatrix}^T$$

the following state space model results in

$$\begin{aligned} \dot{\underline{x}} &= \underline{f}(\underline{x}) + \underline{g}(\underline{x})\, u \\ &= \begin{bmatrix} x_2 \\ -\frac{1}{\tau_A} x_2 \\ x_4 \\ \frac{L}{\tau_A} x_2 - \frac{c_v}{m_l} x_3 - \frac{d_v}{m_l} x_4 - \frac{1}{L} p_R \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{k_A(x_1)}{\tau_A} \\ 0 \\ -\frac{L k_A(x_1)}{\tau_A} \end{bmatrix} u \quad (4) \\ y &= h(\underline{x}) = x_1 + \arctan\left(\frac{x_3}{L}\right) \quad (5) \end{aligned}$$

This model is the basis of the controller design.

III. Trajectory Tracking

The control consists of a feedforward and a feedback loop. The structure is presented in Fig. 4. The feedforward control is calculated within the framework of a multi-body system model utilizing a flatness based approach for model inversion. The reference input to the closed loop system is the signal from the operators hand leveler ($\dot{y}_{ref}(t)$). Therefore, a trajectory generator is needed to provide feasible reference trajectories $\left(\underline{z}_{ref}(t)\right)$ which fulfill the kinematic constraints. The estimated state v_{0z} is fed back for stabilizing the plant along the reference trajectory. The disturbance observer separates the fundamental oscillation v_{0z} from the other oscillation amplitude (v_{1z}) and measurement noise referring to the two measurements ($m_1(t),\ m_2(t)$).

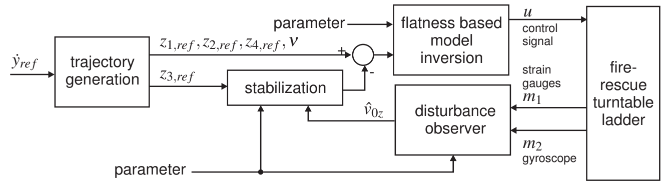

Fig. 4. Scheme of the control structure

Due to the fact that the ladder can be extended to different lengths the parameters of the ladder have to be updated in every time step (e. g. stiffness, the eigenfrequencies, the damping coefficients of the dominant modes, et cetera). Fortunately, the ladder's length is changing very slowly, so it can be considered a parameter. The time variance of the system can be neglected for the controller design, because all the parameters depend on the length of the ladder. However, a gain scheduling depending on the ladder's length is necessary to obtain a good performance within the whole workspace of the ladder.

A. Feedforward Control

To invert the model mentioned in (4), a differentially flat output with the relative degree equal to the system's order ($r = n$) has to be found. The relative degree r is defined as follows

$$\begin{aligned} L_g L_f^i h(\underline{x}) &= 0 \quad \forall\, i = 0, \ldots, r-2 \\ L_g L_f^{r-1} h(\underline{x}) &\neq 0 \quad \forall\, \underline{x} \in R^n \end{aligned} \tag{6}$$

The differential operator L_f represents the Lie derivative of the argument along the vector field $\underline{f}$ and L_g is the Lie derivative along the vector field $\underline{g}$, respectively. The real output mentioned in (5) has a relative degree $r = 2 < n = 4$. Thus, y is not a flat output. If we assume $v_z/L \ll 1$, a new output

$$\tilde{y} = \tilde{h}(\underline{x}) = x_1 + \frac{x_3}{L}$$

with the relative degree $r = n = 4$ is obtained. The difference between the control output and the differentially flat output is negligible.

By using the Byrnes-Isidori normal form

$$\begin{aligned} \bar{S}: \quad & \tilde{y} = z_1,\ \dot{z}_1 = z_2,\ \ldots\ , \dot{z}_{r-1} = z_r \\ & \dot{z}_r = \left[L_f^r h + L_g L_f^{r-1} h u\right] \circ \phi^{-1}(\underline{z}) = \underline{a}(\underline{z}) + \underline{b}(\underline{z})\, u, \end{aligned}$$

and a diffeomorph state transformation

$$\underline{z} = \underline{\phi}(\underline{x}),\ z_i = \phi_i(\underline{x}) = L_f^{i-1} h(\underline{x}) \quad i = 1, \ldots, r,$$

the system can be written as

$$z_1 = x_1 + L^{-1} x_3 \tag{7a}$$

$$z_2 = x_2 + L^{-1} x_4 \tag{7b}$$

$$z_3 = -\frac{c_v}{m_1 L} x_3 \tag{7c}$$

$$z_4 = -\frac{c_v}{m_1 L} x_4 \tag{7d}$$

$$\dot{z}_4 = -\frac{c_v}{\tau_A m_1^2 L}\left(m_1 L x_2 - \tau_A c_v x_3 - m_1 L k_A u\right). \tag{7e}$$

The model can be inverted with respect to the differentially flat output. The new control input is defined as n-th derivative with respect to the time of the flat output

$$\nu = \dot{z}_4 = \tilde{y}^{(IV)}$$

So the control signal u is determined by

$$u = \frac{-\underline{a}(\underline{z}) + \nu}{\underline{b}(\underline{z})}$$
$$u = \frac{c_v z_2 + c_v \tau_A z_3 + m_l z_4 + \tau_A m_l \nu}{c_v \tilde{k}_A(z_1, z_3)} \quad (8)$$

By applying the model inversion to the system, we obtain a chain of four integrators (system's order) with the input ν and the differentially flat output $\tilde{y} = z_1$.

B. Feedback Control

By applying the feedforward control law (8) with feasible trajectories for ν and $\underline{z}$ accordingly, the load sway will be smaller compared to an uncontrolled motion. But for the reason of assumptions which have been made in the design and model mismatches, the load sway can not be eliminated totally. Perturbances such as wind and people moving or working on the platform cannot be fully compensated by feedforward control. So a feedback loop is needed for stabilizing the system around the reference trajectory as well as compensating for perturbances to ensure a minimal load sway.

In order to stabilize the system, a feedback of the error between the reference trajectory $z_{3,ref}$ and the second Lie derivatives of the output $\tilde{y}$ along the vector field $\underline{f}(\underline{x})$ (see (7c)) is derived

$$u = \frac{-L_f^r \tilde{h}(\underline{x}) + \nu - k_3 \left[L_f^2 \tilde{h}(\underline{x}) - z_{3,ref} \right]}{L_g L_f^{r-1} \tilde{h}(\underline{x})} \quad (9)$$

As feedback, a scalar proportional controller has been chosen. It is for the convenience of the operator, because he is not interested in a exact trajectory tracking. His concerns are about a smooth motion with a minimum load sway. But if we use a complete state feedback, the stiff degrees of freedom will compete with the flexible system states. However, there is small loss of accuracy for the trajectory tracking, but the advantage is a higher user's acceptance. Another reason are lower costs because an encoder with smaller resolution can be used for sensing the raising angle (φ_A). Last but not least, the computational costs are lower by using a one-dimensional proportional controller, too.

The controller gain k_3 can be dimensioned by standard means, e.g. root locus, since the Kalman controllability criterion for the system (4) is satisfied ($d_v(L) \equiv 0,\ p_R(t) \equiv 0$)

$$\det \left[\begin{array}{cccc} \underline{B} & \underline{A}\,\underline{B} & \underline{A}^2\,\underline{B} & \underline{A}^3\,\underline{B} \end{array} \right] = \tfrac{k_A{}^4 L^2 c_v(L)^2}{\tau_A{}^4 m_l{}^2} \neq 0$$
$$\text{with } k_A, \tau_A, m_l, L, c_v(L) > 0 \ \forall\ L.$$

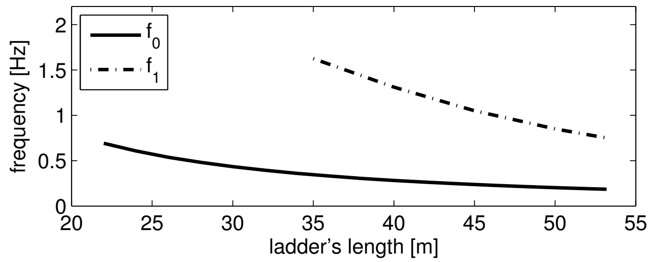

Fig. 5. Eigenfrequencies ($f_i = \omega_i/2\pi$) of the ladder set over its length

IV. Disturbance Observer

The disturbance observer is necessary because the overtones need to be eliminated from the feedback. Filtering the sensor signal, e. g. with nth-order lowpass filter, is not applicable. This is because the frequencies of the first two modes are too close together (s. Fig. 5), so that a high-order lowpass would be needed to ensure a sharp decline beyond the cutoff frequency. This would cause a very strong phasing distortion, which can lead to instability.

Higher modes do not need to be considered, because the cut-off frequency of the hydraulic actuator is about 3 Hz. The information of the two modes is distributed equally in the gyroscope signal. The peaks of the gyroscope measurement in Fig. 2 are level. The strain gauges sense the second mode (overtone) less, but especially in the closed loop control this results in an unacceptable performance. Hence, an observer is used to estimate the states. In simple terms, the observer will have to use the information from the gyroscope to eliminate the overtone from the strain gauges' signal.

The dynamic model for the observer consists of two forced oscillations. The deflection $v_{0z}(t)$ is the amplitude of the fundamental ladder's sway and $v_{1z}(t)$ is the amplitude of the overtone which is considered a disturbance and is discarded for the controller feedback. Parameters like damping coefficients D_i, the circular eigenfrequencies ω_i, and the dc–gain κ_i have been identified by theoretical and experimental analysis. All parameters depend on the length of the ladder (e. g. $f_i = \omega_i/2\pi$ see Fig. 5). Therefore, the observer gain $\underline{H}$ is derived analytically depending on L. During operation the gain is updated in every time step. The oscillations are enforced by the angular acceleration of the ladder ($\ddot{\varphi}_A(t)$).

$$\ddot{v}_{iz}(t) + 2D_i\omega_i\dot{v}_{iz}(t) + \omega_i{}^2 v_{iz} = \kappa_i\omega_i{}^2\ddot{\varphi}_A(t) \quad i = 0, 1 \quad (10)$$

The offset drift and measurement noise in the gyroscope signal $\dot{\bar{v}}_z(t)$ is taken into account by a trivial dynamic

$$\ddot{\bar{v}}_z = 0. \quad (11)$$

By choosing the estimated state vector to

$$\hat{\underline{x}}(t) = \left[\begin{array}{ccccc} \hat{v}_{0z}(t) & \dot{\hat{v}}_{0z}(t) & \hat{v}_{1z}(t) & \dot{\hat{v}}_{1z}(t) & \hat{\bar{v}}(t) \end{array} \right]^T,$$

we obtain a linear state space description

$$\dot{\hat{\underline{x}}}(t) = \underline{A}\,\hat{\underline{x}}(t) + \underline{B}\,u(t)$$
$$\underline{y}_m(t) = \underline{C}\,\hat{\underline{x}}(t)$$

with a system matrix and an input vector as follows

$$\underline{A} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ -{\omega_0}^2 & -2D_0\omega_0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & -{\omega_1}^2 & -2D_1\omega_1 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \tag{12}$$

$$\underline{B} = \begin{bmatrix} 0 \\ \kappa_0{\omega_0}^2 \\ 0 \\ \kappa_1{\omega_1}^2 \\ 0 \end{bmatrix}. \tag{13}$$

Because there are two measurements, we have to take into account that the system's output is a vector with the dimension two. It is assumed that the strain gauges sense the strain free of any overtone and any offset. The gyroscope measures the fundamental oscillation as well as the overtone and it has an offset as mentioned before.

$$\hat{\underline{y}}_m(t) = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 1 & 1 \end{bmatrix} \hat{\underline{x}}(t) \triangleq \begin{bmatrix} \text{strain gauges} \\ \text{gyroscope} \end{bmatrix} \tag{14}$$

The observability matrix has full rank

$$\underline{Q} = \begin{bmatrix} \underline{C} & \underline{C}\,\underline{A} & \underline{C}\,\underline{A}^2 & \underline{C}\,\underline{A}^3 & \underline{C}\,\underline{A}^4 \end{bmatrix}^T$$
$$\operatorname{rank}\left(\underline{Q}\right) = 5.$$

Hence, all states can be estimated. With $\underline{y}_m(t)$ as the real measurement vector a Luenberger-type of observer can be written as

$$\dot{\hat{\underline{x}}}(t) = (\underline{A} + \underline{H}\,\underline{C})\,\hat{\underline{x}}(t) + \underline{B}u(t) - \underline{H}\,\underline{y}_m(t).$$

For the design of the observer feedback gain $\underline{H}$, the description in observer canonical form is being used. After the transformation into observer canonical form ([3] p. 200 et seqq.) the system matrix and the output matrix have the form (only elements different from zero are mentioned)

$$\underline{A}_O = \begin{bmatrix} 0 & -a_{10} & 0 & 0 & 0 \\ 1 & -a_{11} & 0 & 0 & 0 \\ 0 & a_{20}^1 & 0 & 0 & 0 \\ 0 & a_{21}^1 & 1 & 0 & -a_{21} \\ 0 & 0 & 0 & 1 & -a_{22} \end{bmatrix} \tag{15}$$

$$\underline{C}_O = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{16}$$

In this form the calculation of the elements of the observer gain matrix is easier, because one can use standard means like pole placement ([4] p. 115 et seqq.). The first elements $h_{O,11}$ and $h_{O,21}$ are influenced by the placed eigenvalues belonging to the first dynamic subsystem (1st oscillation), only. The elements $h_{O,j2}$ $(j = 1, 2, 3)$ depend on the three poles (and their placement), which refer to the second dynamic subsystem (2nd oscillation and offset). The elements $h_{O,31}$ and $h_{O,41}$ are cross terms of the two subsystems and they are not influenced by any placed eigenvalue.

$$\underline{H}_O = \begin{bmatrix} h_{O,11} & 0 \\ h_{O,21} & 0 \\ h_{O,31} & h_{O,32} \\ h_{O,41} & h_{O,42} \\ 0 & h_{O,42} \end{bmatrix} \tag{17}$$

The feedback matrix of the observer $\underline{H}$ can be obtained by applying the inverse transformation on the observer gain matrix in observer canonical form $\underline{H}_O$. The transformation matrix is the inverse observability matrix of the system in observer canonical form

$$\underline{T}_O = {\underline{Q}_O}^{-1} = \begin{bmatrix} \underline{C}_O \\ \underline{C}_O\underline{A}_O \\ \underline{C}_O\underline{A}_O^2 \\ \underline{C}_O\underline{A}_O^3 \\ \underline{C}_O\underline{A}_O^4 \end{bmatrix}^{-1}.$$

V. Measurement Results

In this section we present experimental results, which were achieved using the proposed control concept. The efficiency of the disturbance observer is shown in Fig. 6. In this experiment, the ladder was excited with the actuators and external forces, so that the overtones have large amplitudes. The observer (solid) reduces the amplitudes of the high-frequency oscillation appreciably compared to the pure sensor signal (strain gauges).

During the second experiment, the ladder was erected from 65 ° to 71 ° without any controller in action. Only some ramps and filters are smoothing the signals coming from the operators hand lever. Afterward the same motion was repeated with active oscillation damping supported by the disturbance observer. The ladder was extended to a length of 53.2 m, which is its maximum length.

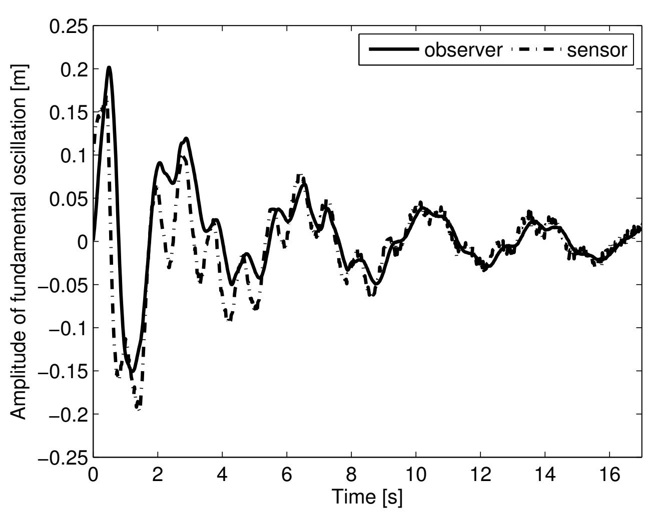

Fig. 6. Amplitude of the fundamental oscillation: sensed state (solid, strain gauges) vs. observed state (dash-dotted)

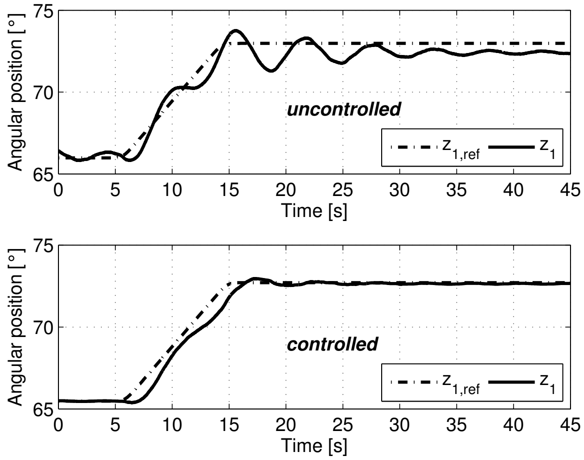

Fig. 7. The angular position of the cage at the end of the ladder in degree

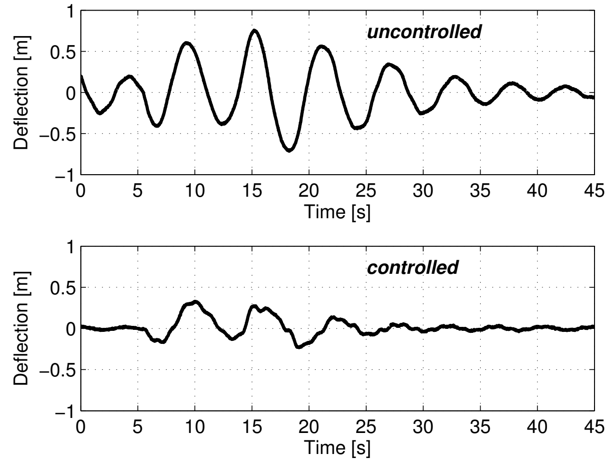

Fig. 8. The deflection at the end of the ladder (v_z) in m

Figure 7 shows the angular position of the cage (ladder's tip) and its reference trajectory. The dash–dotted line is the reference trajectory of the differentially flat output $(z_{1,ref} = \tilde{y}_{ref} \approx y_{ref})$, the solid line is the sensed position $\left(z_1 = \varphi_A + v_z L^{-1}\right)$. One can tell that with active damping (lower plot) there is a acceptable tracking behavior with a small delay. The deflection at the ladder's tip (v_z) is plotted in Fig. 8. The improvement from the active damping is obvious. The amplitude of the oscillation is reduced to about 30 % compared to an uncontrolled motion. The swaying is damped within one periodic time of the fundamental oscillation and the residual ladder's sway is negligible. In the lower plot of Fig. 8, the overtones excited by the actuators are recognizable, because the pure sensor signal of the strain gauges (converted into deflection) is plotted instead of the estimated state $\hat{v}_{0z}$. In the lower subplot it becomes obvious that by closing the control loop, the overtone is gaining weight compared to the fundamental oscillation. However, due to the use of the disturbance observer, which reduces the amplitude of the overtone (cp. Fig. 6), this effects have almost no influence on the controller output.

VI. CONCLUSION

In this paper a 2-degree-of-freedom control for erecting motion of a fire–rescue turntable ladder is presented. For the feedforward control a model inversion based on a differentially flat output is applied to a simple linear model of the ladder. The feedback is a scalar proportional controller stabilizing the plant along the reference trajectory of a time derivative of the differentially flat output. A disturbance observer is developed to eliminate the high-frequency oscillations of the ladder, which are not considered in the model for controller design, to avoid instabilities of the closed loop control. The control law is realized on a micro controller system with fix-point arithmetic and limited computational power. In contrast to earlier works, the proposed approach can be used for ladders with length over 30 m and it is adaptive to varying ladder lengths without additional requirements on a micro controller. At the present, the proposed approach is verified at the IVECO DLK 55 CS turntable ladder. For future work the non-linear dynamics of the hydraulic cylinders will be taken into account more precisely, to improve performance.

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