Theoretical Aspects of Active Skew Control

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Abstract

The cranes at the European Combined Terminal (ECT) in Rotterdam are subject of a long term study to automate the international seaport and make container handling more efficient.
In this paper we describe practical aspects and advantages of the Siemens sway control system and the design and simulation of a skew controller for attenuating the skew motion of the suspended container during transshipment. State feedback control methodology is used for this application. A 3D dynamic model of the container crane was previously developed, and is used for the optimization of the feedback gains.

Keywords: container crane, non-linear control, state feedback, sway control, skew control.

1. Introduction

Electronic sway control systems are being used to an ever-greater extent on crane installations. A number of preconditions must be satisfied, however, to ensure the new equipment is actually used by the crane drivers. For example, the attainable handling capacity must be at least as high as can be reached by the driver by manual control. Furthermore, the driver must not be subjected to uncomfortable jerk loads. Above all, however, the operation of the system must be simple and easy to understand. For example, in the case of automatic or semi-automatic operating sequences, complicated teaching routines should be avoided. Siemens designed and improved it's HIPAC (Highly Intelligent Automatic Control) sway control for container quay cranes to reach all these features.

2. Innovation project FAMAS Jumbo Container Crane

During 1996, the Centre of Transport Technology (CTT) has initiated the FAMAS (First, All Modes, All Sizes) programme. The goal of the programme is to develop a new generation of container terminals capable to handle all modalities of transhipment of all container sizes with an equal service level. To execute this programme a consortium has been formed by several major, transport related companies like ECT, Siemens Nederland, Nelcon and the Delft University of Technology.
To accomplish the goal of the programme, a highly automated terminal has to be developed. Although, robotising is not the main goal of the programme, it is the opinion that handling containers with a throughput of 500.000 TEU a year or more can only be done economically and efficiently by robotising stacking and terminal transport.
The demands made on container handling in international seaports are continuously increasing, because both the capacity of the vessels is increasing, and the service time of container vessels should be decreasing. In order to minimise this service time a new generation of the so-called Jumbo Container Cranes (JCC) will be developed.
This paper presents a control system for this new generation of container gantry cranes. The requirements for this control system are to:

- accurately position the container in its end position from an arbitrary starting position and initial conditions,
- transport the container in a time optimal way,
- avoid large overshoot in container position,
- operate robust e.g. independent of disturbances especially of wind forces,
- take into account constraints on maximum force, acceleration and speed,

The control system applies a state feedback loop with state variable gains. The feedback gains are continuously adjusted using a model in order to optimise the performance of the container crane for all possible initial conditions, container masses, and cable lengths.

The control system activates the drive systems of the crane such that a container is transported in a time-optimal way, and is positioned accurately and robustly on the desired destination.

Figure 1 Schematic drawing of the container crane and its trajectory of the container.

The need for fast and safe loading and unloading container vessels requires a control of the crane motion that optimises the crane’s performance. Figure 1 shows a two-dimensional cycle of the crane motion. The problems are the reduction of the total time of load transport (time-optimal trajectory control [1], [3]) and reduction of the dynamic motion of the load at its end position, including accurate positioning of the load.
During the transport of the load, the planar position is controlled by varying the position of the trolley and varying the length of the suspension rope. Specific constraints such as maximum torque (which is a maximum acceleration or deceleration of the trolley or load), maximum speed and the obstacles on the quay or in the container vessel, have to be incorporated in the trajectory controller.
A validated mathematical model is necessary for a detailed study of the dynamic behaviour of the container crane and control schemes. Although several analyses have been made, using non-linear models [2], [4], further increased complexity may be considered by including the detailed construction of the crane, the stretch in the cables and the influence of wind [6].

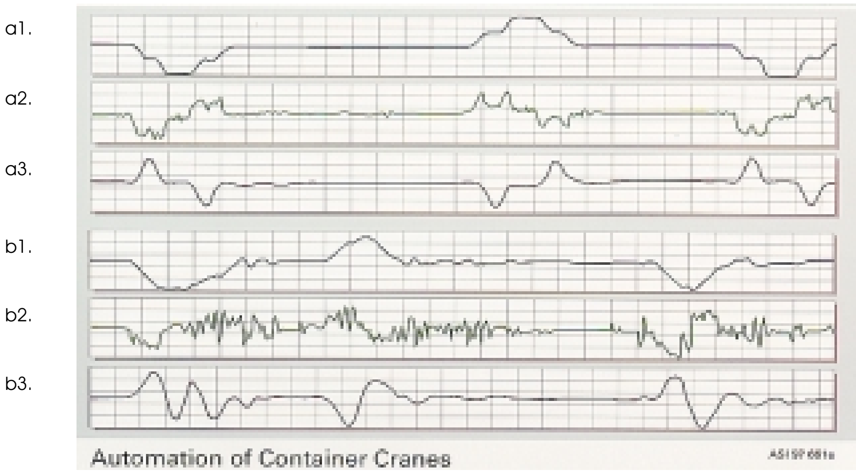

Figure 2 HIPAC versus manual control of sway:

a1. HIPAC: speed of the trolley,
a2. HIPAC: trolley torque,
a3. HIPAC: sway angle,
b1. Manual control: speed of the trolley,
b2. Manual control: trolley torque,
b3. Manual control: sway angle.

The decision for the actuator and control methodology is explained after the options for skew control. Then the state feedback control scheme is explained and the functions for the feedback gains are described.
The implementation of the control scheme in a 3D model provides for the simulation results. Finally, conclusions are presented based on simulation.

3. Highly Intelligent Automatic Control (HIPAC)

The aim of HIPAC[1] is to increase the handling capacity, to increase safety but also to reduce the mechanical stresses (Figure 2). The HIPAC swing control permits four operating modes: (conventional) manual operation, automatic operation, manual operation with swing control and the sway-stop operation. For the operation of the HIPAC controller the variations in the torque are reduced considerably. No complicated teaching routines are necessary to teach the system. It records every movement during automatic or manual control to learn the optimal time curve while avoiding the obstacles.
The latest step in the development of HIPAC is controlling the crane from a separate HIPAC operating-cabin. The mentioned operating modes and skew control are installed on a quay crane of ECT at the Maasvlakte Rotterdam. The unpredictable movements of the trolley will therefor not disturb the driver and HIPAC can operate by what it’s designed for: automatic operation.
The techniques of sway and skew control have now reached a state of progress at which it brings genuine benefits to the crane operator. A control process optimized in many respects, as well as the user-oriented design of the operating elements increase handling capacity and operating safety and reduce the stresses for crane drivers and material.
However continuous improvement of sway, or better automatic control, is necessary to follow the constant increasing performance-requirements of the customers. One of the latest improvements to come towards is the development and integration of the skew controller, which will be discussed, in the next chapters.

4. Newtonian model

A container has six degrees of freedom (6 d.o.f.) for its movements. The container can show a translation in three directions and a rotation around three axes:

- trim,
- list,
- skew.

During the movement of the trolley the container will *sway*. For the reduction of this motion control algorithms are implemented and tested in the laboratory and in practice on real cranes.
Under specific conditions the container can also show a *skew* rotation. Because of this, accurate positioning of the container will be obstructed and therefore the skew has to be reduced.

For the description of these movements a 3D model (or a quasi-3D model) has been developed. This model describes the dynamics of the container suspended on four parallel cables. In this model it is allowed that the length of both cables is different, which may lead to skew. The centre of gravity of each container can also be different. This centre of gravity is an important parameter for the skew motion. Manipulating the length of the cables and/or its position has been considered as a possibility to influence the skew movement.

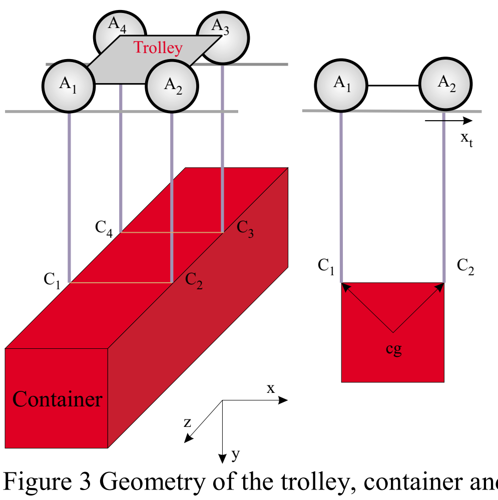

Figure 3 Geometry of the trolley, container and cables.

[1] HIPAC (Highly Intelligent Pendulum Controller) is a product of Siemens.

Figure 3 shows the configuration of the trolley and container system. In order to model the motion dynamics of the container body, two coordinate reference frames are assigned:

- a global reference frame on the base of the rail,
- a container attached reference frame originated in the geometric centre of the container.

It is clear from Figure 3 that the displacement vectors for the cables can be stated as

$$^{O}D_i = {}^{O}A_i - {}^{O}C_i \quad \text{for } i = 1 \cdots 4 \qquad (1)$$

The locations of the pulleys $^{O}A_i$ on the trolley can be derived from the trolley position on the rail x_T and the geometric design of the crane. The locations of the pulleys $^{O}C_i$ on the container are derived from the motion of the container

$$^{O}C_i = \bar{x} + {}^{O}R_C \, {}^{C}C_i \qquad (2)$$

where the pulley locations $^{C}\overline{C}_i$ are in the local reference frame of the container $^{0}\overline{R}_c$. The forces in the cables are given by

$$^{O}\overline{F}_i = \frac{^{O}\overline{D}_i}{|D_i|}\left(K_S S_i + K_D \dot{S}_i\right) \qquad (3)$$

where K_S is the stiffness coefficient, K_D the damping coefficient, and S_i the stretch in cable *i*. With the unloaded cable length L_i, the stretch is computed as

$$S_i = |D_i| - L_i \qquad (4)$$

The total sum of the forces acting on the container and spreader is the sum of the individual cable forces, together with the wind and gravitational force. Since the 3-dimensional direction of the forces is known, and also the locations where these forces act on the container (being the container pulleys $^{O}C_i$), also the torques on the container can be computed.

The resulting dynamic equations for the container mass can be represented by the diagram in Figure 4.

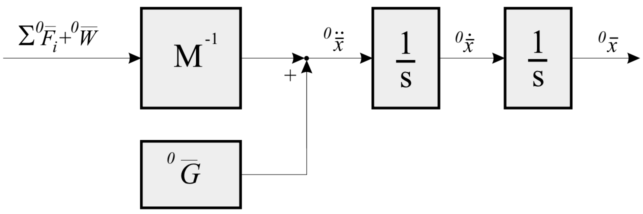

Figure 4 Block diagram for the translational container motion dynamics.

A state space model description for the translational container motion [6] can be expressed as

$$\frac{d}{dt}\begin{bmatrix} ^{0}\dot{\bar{x}} \\ ^{0}\bar{x} \end{bmatrix} = \begin{bmatrix} 0 & 0 \\ \bar{I} & 0 \end{bmatrix}\begin{bmatrix} ^{0}\dot{\bar{x}} \\ ^{0}\bar{x} \end{bmatrix} + \begin{bmatrix} M^{-1}(\sum {}^{0}\overline{F}_i + {}^{0}\overline{W}) + {}^{0}\overline{G} \\ 0 \end{bmatrix} \qquad (5)$$

where

$^{0}\overline{F}_i$	sum of cable forces,
$^{0}\overline{W}$	wind force in inertia reference frame,
M	mass matrix,
$M\ {}^{0}\overline{G}$	gravitational force,
I	inertia vector,
x	position of centre of gravity c.g.

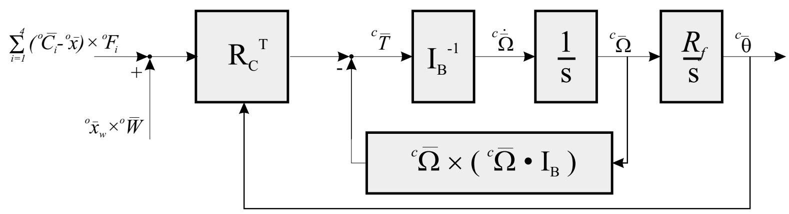

Figure 5 Block diagram for the rotational container motion dynamics.

The diagram in Figure 5 represents the model for the angular dynamics. The body rotation angles are the elements of $^{0}\overline{\Theta}$. A state space model description for the angular dynamics [6] is given by

$$\frac{d}{dt}\begin{bmatrix} {}^0\overline{\Omega} \\ {}^0\overline{\Theta} \end{bmatrix} = \begin{bmatrix} -I^{-1}SI & 0 \\ R_f & 0 \end{bmatrix}\begin{bmatrix} {}^0\overline{\Omega} \\ {}^0\overline{\Theta} \end{bmatrix} + \begin{bmatrix} I^{-1}\overline{R}_C \\ 0 \end{bmatrix}{}^0\overline{T} \tag{6}$$

where

${}^0\overline{T}$	torques acting on the container,
${}^0\overline{\Omega}$	container gyroscopic angle,
${}^0\overline{\Theta}$	Euler angle,
${}^c\overline{\Omega}\times(I\bullet{}^c\overline{\Omega})$	coriolis and centrifugal torques,

$$S = \begin{bmatrix} 0 & -\omega_z & \omega_y \\ \omega_z & 0 & -\omega_x \\ -\omega_y & \omega_x & 0 \end{bmatrix} \tag{7}$$

Other definitions of the various variables are further explained in [6].

5. Control system

The control system is composed of two controllers:
- a sway controller,
- a skew controller.

The purpose of the sway controller is to generate and follow real-time the time-optimal trajectory and to position the container in the end-position with a sway angle zero and within the required accuracy. The controller takes into account the constraints of the crane design.
The skew controller will eliminate the skew angle of the container as fast as possible.
Both controllers are designed as a state-feedback controller with variable feedback gains and an on-line gain optimizer. Both controllers need information from the sensors assembled with the crane.

Skew Control Methods

A variety of methods are available to influence the skewing motion of the suspended container. For skew control a distinction can be made between direct methods and indirect methods.
Examples of *indirect methods* for skew control are based on:
1. individual cable length control by varying the length of the cables separately anywhere in the cable loop. For example by controlling the hoisting drums separately the individual cable length can be varied.
2. applying forces on the container at the trolley for example by mounting additional diagonal cables between the trolley and the spreader or by applying forces to the hoisting pulleys on the trolley in the trolley motion direction, anti-symmetrically. In this way, forces are applied on both sides of the container.

Examples of *direct methods* for skew control are based on:
1. application of flywheels on the spreader,
2. mounting of propellers on the spreader,
3. control of the mass distribution by shifting of the pulleys for the hoisting cables on the spreader,
4. use of linear drives to shift the container in the mounts of the spreader, also in order to control the mass distribution,
5. additional mass to the spreader, to correct for unbalanced loading of the spreader.

The direct methods are generally more efficient, but less robust than the indirect methods.

Skew Control Actuator System

Considering controllability and robustness of skew control methods, the configuration mentioned above under indirect methods number 2 has been selected to attenuate container skew motion. Notice that the trimming of the container, which is current practice in the container crane operation, corresponds with method 1 for passive skew control. It is recommended that container trimming is maintained for the application of active skew control as well.
On the new cranes this active skew actuator will be realised by two anti-parallel movable hoist sheave units on the trolley in the trolley direction. The sheave units will be controlled by the real-time active skew controller integrated in HIPAC via the AC Masterdrive.

On the new cranes this active skew actuator is realised by two anti-parallel movable hoist sheave units on the trolley in the trolley direction. The sheave units will be controlled by the real-time active skew controller integrated in HIPAC through the DC Masterdrive of the Siemens PLC.
The actuator for the skew control system, called a “wagon”, consists of the anti-parallel hoist sheave units. The mechanical construction is given in Figure 6. Two linear wagons can be moved by a reversible gear. The hoisting pulleys are mounted on the wagon units.

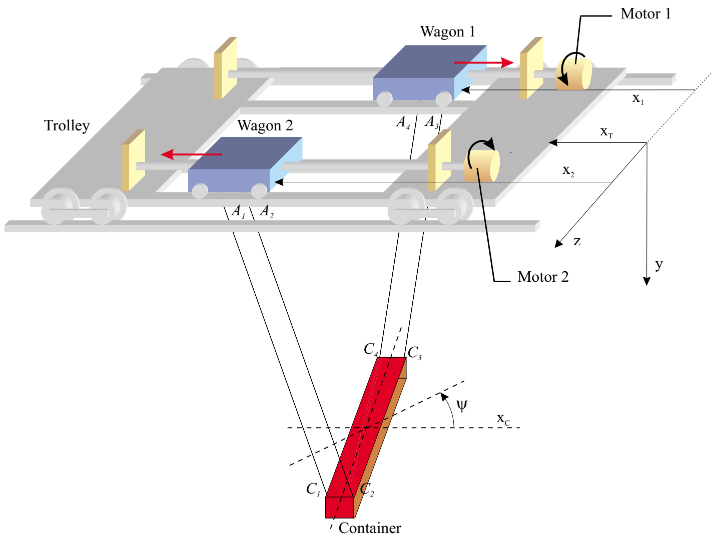

Figure 6 Mechanical construction of the skew actuator with two linear translational wagons.

The dynamic non-linear model of the actuator with signal limitations is given in Figure 7.
The control strategy is based on continuous measurements of the sheave unit position and the skew angle. The goal of the system is the regulation of the skew movement during or after trolley acceleration and deceleration. By regulating this skew movement, the container can be automatically lowered straight on the AGV without the need for further equipment like a chassis loader.

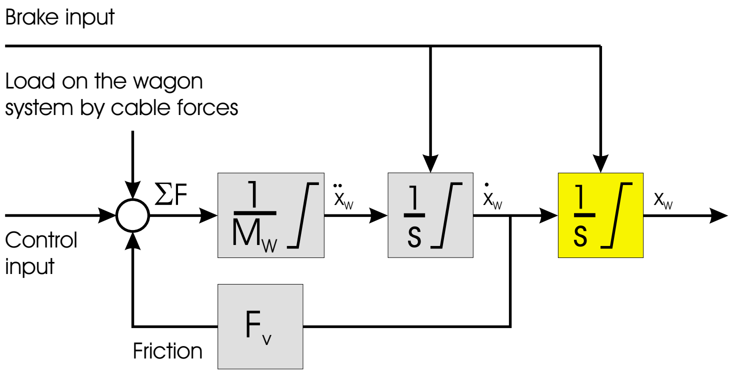

Figure 7 Diagram of the non-linear dynamical model of the skew actuator.

Skew Control Concept
The block diagram of the skew control system is given in Figure 8. The measured value of the skew angle ψ is compared with the defined desired reference value. Normally this reference value equals zero.

The error signal ε_ψ is fed to a PID-controller using the gain parameter K_1, giving an actuation reference value for the positioning of the wagons. The wagon system is implemented in a well-known way, making use of the velocity feedback loop ($\dot{x}_W$) and design gain parameters K_3 and K_4. To improve the dynamic behaviour of the skew control system (especially to increase the bandwidth of the system), the PID-controller in the forward path of the skew control is replaced by a separate velocity feedback loop $\dot{\psi}$ and gain parameters K_1 and K_2. This design method omits an integral action. In a later stage of the design, the possibility is available to add an integral action in the forward loop, so that the accuracy in the last part of the positioning can be improved (the I-action may easily be defined as a function of the positioning error).

State Feedback Control
For the control of skew motion the state feedback control based on its generality and robustness. However, in order for every generated control to be admissible, the state feedback gains should depend on the state of the system:

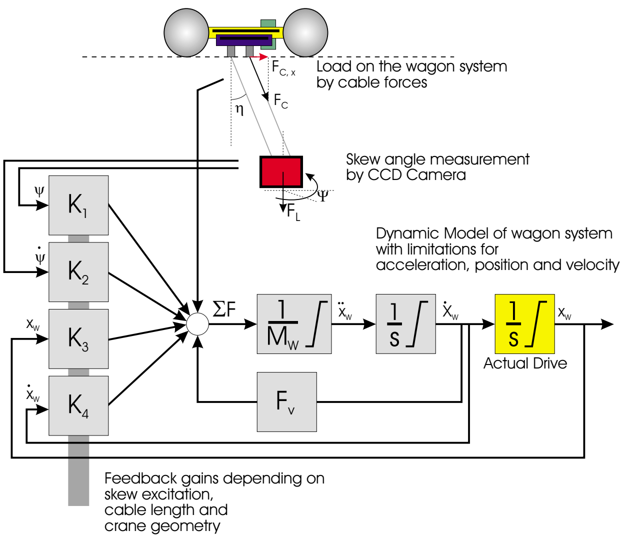

Figure 8 Diagram of control model and the dynamical model of the skew actuator.

- lower gains for large errors, so that the system is not controlled over its limits,
- higher gains for approaching steady state, in order to minimise the error.

Full state feedback control allows for the designer to place the poles of the model anywhere in the complex plane without adding dynamic states to the system. With a limited state feedback control, the designer is not anymore able to place each pole individually.

The state feedback controller for the container crane can be described by the following set of equations:

$$F = K_1(f)\psi + K_2(f)\dot{\psi} + K_3(f)x_W + K_4(f)\dot{x}_W \qquad (8)$$

$$\ddot{x}_W = \frac{1}{M_W}F \qquad (9)$$

$$\dot{x}_W = \dot{x}_W(0) + \int_{t0}^{t}\ddot{x}_W\, dt \qquad (10)$$

where F represents the *virtual force* for the desired skew actuator motion, and $K_i(f)$, $i= 1,\ldots,4$ represents the *gains* of the state feedback controller. The skew angle and skew rate are denoted by ψ and $\dot{\psi}$, and the wagon position and speed with x_W and $\dot{x}_W$, respectively. The gains are dependent on the state of the system. The force F will be transformed to the desired actuator acceleration and the acceleration is then integrated to the desired actuator speed $\dot{\psi}$, which is the output of the controller. In order to save the motor drive from overheating, a moving-average filter is implemented in the code for smoothing the acceleration.

Optimal Control Design

The next problem is to find the controller gains $K_i(f)$. Parametric optimisation is used to find a set of controller design parameters $C=\{K_1, K_2,\ldots,K_n\}$ which can in some way be defined as optimal. In the case of the skew actuator system, the objective function $f(C)$ to be minimised, may be subject to constraints in the form of equality constraints $G_i(C)=0$ (i= 1,…,m_e), inequality constraints $G_i(C)\leq 0$ ($i=m_e+1,\ldots,m$) and parameter bounds C_ℓ, C_u.

A General Problem (GP) description is stated as

$$\min_{C\in\Re^n} f(C) \qquad (11)$$

subject to:

$$G_i C = 0 \qquad i = 1,\ldots,m_e$$
$$G(C) \leq 0 \qquad i = m_e + 1,\ldots,m \qquad (12)$$
$$C_\ell \leq C \leq C_u$$

where C is the vector of controller design parameters (i.e. the feedback gains), ($C \in \Re^n$), $f(C)$ is the objective function that returns a scalar value ($f(C): \Re^n \rightarrow \Re$), and the vector function $G(C)$ returns the values of the equality and inequality constraints evaluated at C ($G(C): \Re^n \rightarrow \Re^m$).

Quadratic Programming (QP) concerns the minimisation or maximisation of a quadratic objective function that is linearly constrained. For QP problems, reliable solution procedures are readily available.
In the case of the skew control problem, the performance criterion can be formulated as the time quadratic penalty of the skew motion angle error with the desired skew angle, i.e.

$$f(C) = \int_t \psi \, t^2 \, dt \qquad (13)$$

The skew angle $\psi(t)$ is found by simulating the crane model with a set of initial conditions for a period of time, and recording the states of the system. The model in this case is described in detail in the paper [6].
The inequality constraints are defined as the physical actuator limitations plus the overshoot. In other words, the physical limitations may never be exceeded and the overshoot must be less than the allowable skew angle margin.
In equations these inequality constraints are formulated as follows:

$$\begin{aligned} G_1(C) &= \max(|x_W|) - \Delta_1 \\ G_2(C) &= \max(|\dot{x}_W|) - \Delta_2 \\ G_3(C) &= \max(|\ddot{x}_W|) - \Delta_3 \\ G_4(C) &= \max(\psi) - \Delta_4 \end{aligned} \qquad (14)$$

Where $\Delta_1, \ldots \Delta_4$ represent the constraints on the wagon position, speed, acceleration and skew angle.
The optimisation process has been conducted for different *cable lengths* and different *inertia*, since these parameters are of influence on the harmonic skew frequency.
From the optimisation results, functions for the gains K_i can be derived, depending on the cable length and load inertia. The gains have not been made a function of the load mass M. The mathematical derivation for a pendulum does not identify any relation of the pendulum time and the load mass.

Effects of Inertia
A homogeneous weight distribution for a 3D rectangular shape is equated with

$$I = \tfrac{1}{12} M \begin{bmatrix} W^2 + H^2 \\ L^2 + H^2 \\ L^2 + W^2 \end{bmatrix} \qquad (15)$$

with length L, width W and height H being the shape dimensions in the x, y and z-direction, respectively.
It may be obvious form a mechanical dynamic standpoint that the inertia is important in the controller configuration. The problem is how the inertia should be estimated for an unknown container. The only clue we would have is the skew harmonic period time, with respect to the mass and the pendulum length.
From the fact that the container dimension appears in the formula for the inertia in equation (15), we can conclude that the variation in load inertia reflects the different sizes of containers (assuming homogeneous weight distribution). The inertia will thus be made a function of the container size.

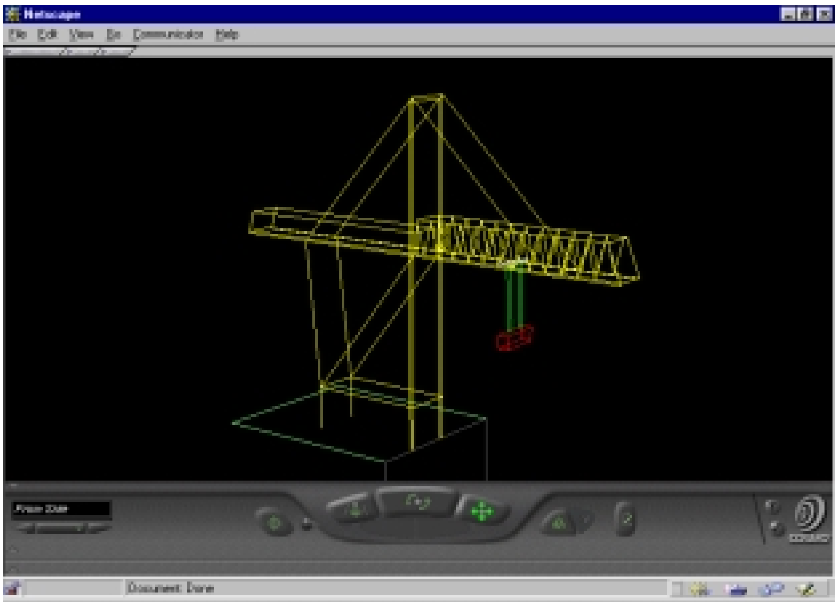

Figure 9 Simulation results are shown in VRML.

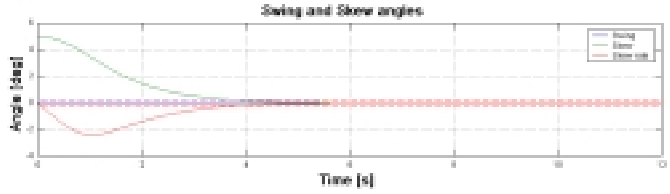

Figure 10 Simulation results of the skew controller:

weight container: 40 ton,
initial sway angle: 0°,
cable length: ℓ=35 m,
initial skew angle: 5°

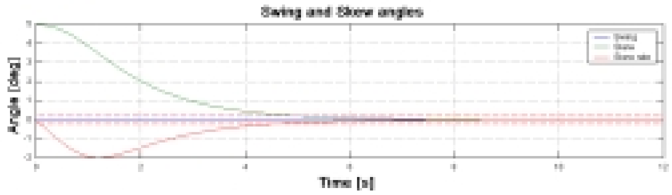

Figure 11 Simulation results of the skew controller:

weight container: 40 ton,
initial sway angle: 0°,
cable length: ℓ=50 m,
initial skew angle: 5°.

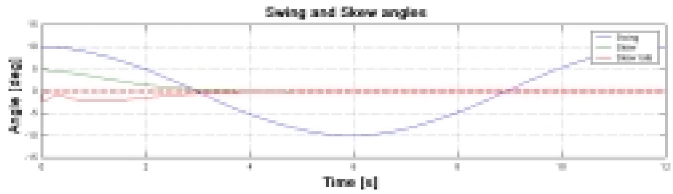

Figure 12 Simulation results of the skew controller:

weight container: 40 ton,
initial sway angle: 10°,
cable length ℓ=35 m,
initial skew angle: 5°.

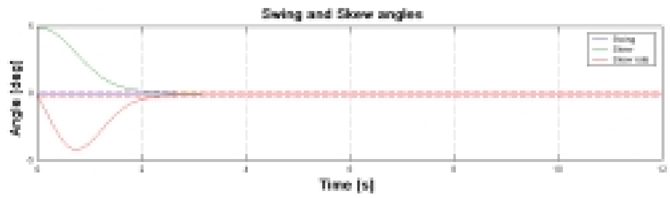

Figure 13 Simulation results of the skew controller:

weight container: 20 ton,
initial sway angle: 0°,
cable length ℓ=15 m,
initial skew angle: 5°.

6. Simulation

The mathematical formulation for the container crane model and the skew controller have been implemented in Matlab. The results of simulation are displayed using a VRML presentation of the crane as shown in Figure 9. In a scenario where the container is moved from the ship to the shore under skew motion, and lowering takes place during the horizontal transport, the controller shows good performance for attenuating skew motion. This simulation setup has been used for studying the behaviour of the system under numerous operating conditions and scenarios.

Simulation results skew control

A controller based on the model shown in Figure 8 is designed and tested. Figure 10 and Figure 11 show the results of a simulation of the skew controller. The figures present the attenuation of a skew angle of 5° for different cable length between 35 m and 50 m and a container weight of 40 ton. From the simulations it follows that an initial skew angle of 5° and a cable length of ℓ=35 m (50 m) is eliminated within 5 s (6 s).

Figure 12 shows the results of a simulation for a combined skew and swing motion. The figure shows how the skew angle is also reduced to zero in a short time even if there is a significant sway motion. The maximum sway angle is here 10° while the skew angle of 5° is eliminated within 4 s for a cable length of ℓ=35 m.
Figure 13 shows the results of a simulation of the skew controller. The figure presents the attenuation of a skew angle of 5° for a cable length of 15 m and a container weight of 20 ton. From the simulation it follows that an initial skew angle of 5° is eliminated within 3 s.

7. Conclusions

The skew attenuation performance of the controller is hardly independent on the load mass, the cable stiffness or the sway motion. When the actuator is not in saturation, the performance is also independent on the amplitude of the skew motion when the controller is activated. Furthermore, when the center of gravity is the restricted boundary region, according to the regulations for container transportation, then the skew control performance is also independent on the location of the center of mass.
The pendulum length and the inertia are both the critical parameters for configuring the skew controller optimally. The pendulum length is measured and an error on it has proven to be of small influence. Also the error on the inertia shows only a small decrease in performance. The inertia however, will still be a guess, since no sensors provide a value for this parameter. The spreader including the empty container are assumed to average to a homogeneous weight distribution. The additional load of the contents of the container is estimated to be homogeneously distributed.
Wind in simulations it has also been shown that a container which experiences a strong wind gust is effectively controlled back to neutral by the controller.
Sensor noise and interruptions have been shown to deteriorate the performance of the control scheme. Generally, the control algorithm will be as good as the sensor inputs. In other words, the controller will not increase the accuracy of the overall system. Simulations have shown that the controller will still perform under noise and interruptions of the sensor skew signal.
A control system for the reduction of the skew motion in a container gantry crane has been presented. This controller is based on a state feedback loop with time variable gains, which are optimized using a model based optimization strategy.
By applying this controller the crane position a container in a fast way without skew (and also without the swing) from any random initial conditions and for any arbitrary mass or load and cable length. The positioning of the container is accurate and within the required 5 cm.

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