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DESIGN ASPECTS OF A MECHANISM FOR AUTOMATED CONTAINER HANDLING

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1 Introduction

In the Rotterdam harbour area several container terminals are in operation. The newest one is almost fully automated. Only the terminal crane that lifts the containers from the quay to the ship (or reversed), is man-operated. The internal transport of a container to and from the storage yard is done by an AGV (automatic guided vehicle). The storage yard consists of several lanes in which the containers are stacked up to three high. Each lane has its own ASC (automatic stacking crane, gantry type), which brings one container at a time from the AGV pickup position to the storage position (or reversed). Two major problems are involved with the design of this ASC, which is basically a huge XYZ-manipulator:

- Position accuracy. In the ASC the container is hoisted at cables. A mechanism is to be added to guide the vertical motion in order to prevent the horizontal swinging.
- Space occupation. The guiding mechanism should not collide with the containers already in the stack. The most critical situation arises, when the smaller container type (20 feet) shall be guided to ground level and when full stacks surround this place, see fig. 1 left .

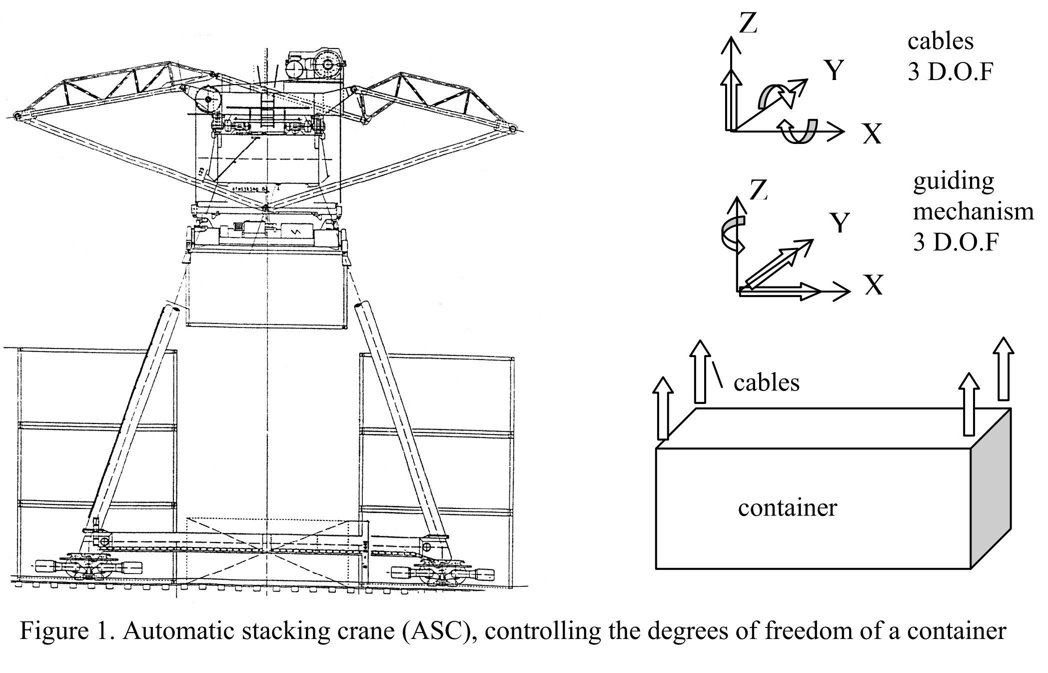


Figure 1. Automatic stacking crane (ASC), controlling the degrees of freedom of a container

2 Problem definition

The design problem described here focuses on the method of swinging prevention. Basically there are three principles, see fig. 2.

- By applying slanted hoisting cables. A container hanging at slanted ropes can be regarded as a planar four-bar mechanism. The best swinging prevention occurs when the momentaneous centre of motion lies near the bottom of the container. It works only in the mechanism plane and will be effective in or around one particular height.
- By adding damping (or friction) to the horizontal motion of the spreader (= gripper for containers). An alternative solution of this type uses guiding cables that can be wounded up. The trolley on the crane has extension arms to keep the guiding cables more or less horizontal. Moving down the guiding cables can passively be released with friction. Hoisting requires a driving motor for winding up the cables. Friction is effective to avoid swinging, but also costs energy and destroys the position accuracy. Controlled winding (4-quadrant operation) does not need to have these disadvantages, but this is technically much more complex and expensive.
- By a guiding mechanism. Such a mechanism needs to guarantee sufficient stiffness in X- and Y-direction (the two horizontal directions). A planar mechanism producing a straight line can be applied, assumed that a coplanar construction can provide the required stiffness in the out-of-plane direction. Figure 1, right part, explains the analysis of the degrees of freedom. The accepted handling method with a spreader, connected to the top corner fittings, eliminates not only the vertical degree of freedom of the container, but also the two rotations out of the horizontal plane. The guiding mechanism should determine the other three DOF's: horizontal translation (X and Y) and rotation around the vertical axis.

The method with friction cables could favourably be combined with the slanted cable method. This has been done in earlier ASC designs. The rotation around the Z-axis is however still possible.
The method with the guiding mechanism is simple from the design point of view: the more stiffness the more accurate the position of the container. A light construction is of course prevailed, but this is not a very critical point. The heavy container and spreader (50,000 kg) will probably dominate the dynamic behaviour of the manipulating motion. Space occupation will however certainly play a role. The attention is further drawn to the design of a particular type of such a mechanism.

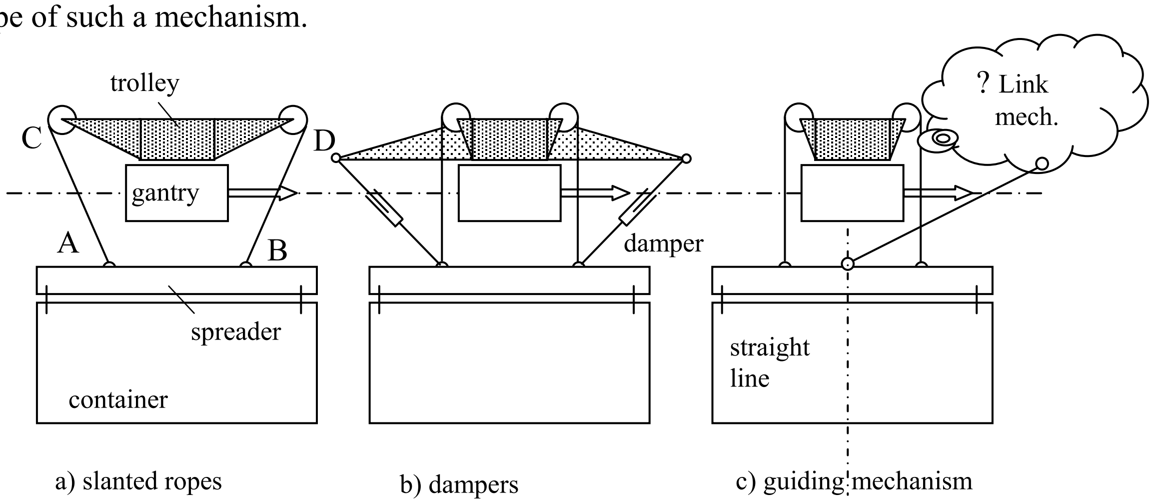
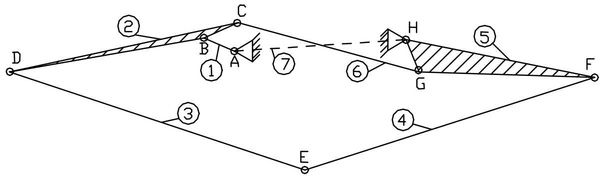


Figure 2. Methods of swinging prevention



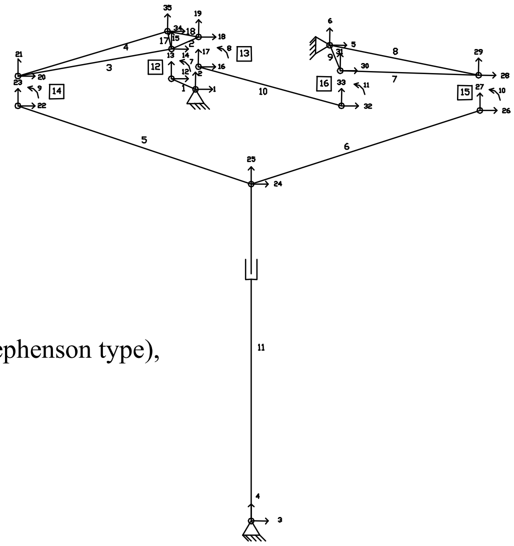


Figure 3. Proposed guiding mechanism (planar 6-bar, Stephenson type),
Kinematic model and static force model

3 Guiding mechanism

In literature many planar mechanisms are known that have a coupler point generating a straight line exactly or approximately. Here a specific type will be regarded: the six-bar linkage according to Stephenson, see fig. 3.
Obviously the original idea is derived from a symmetrical five-bar linkage, the two driving links rotated simultaneously in opposite direction. In that case an exact straight line in space (3D) is generated. The mechanism studied here has the two driving arms BD and HF (B is a fixed point here) connected by a sixth link, thus forming a four-bar subchain BCGH. The starting point of the design study is the mechanism in a kinematically optimized form. This means that the original five-bar mechanism is not longer precisely symmetrical and that the guiding point moves only approximated along a straight line.
The mechanism can be applied favourably in the driving direction of the ASC. Now the arms occupy in the best possible manner the space gap of a three-container stack.

4 Design demands

The overall position accuracy of the container depends on the accuracy of the whole ASC, not only on the guiding mechanism. In several situations the accuracy is important:

- When placing a container on top of another container in the yard. The result should be a stable stack. A relative low accuracy, estimated 5 cm, can be accepted.
- When coupling the spreader to the container (retrieving). The spreader has slanted guiding corners, which adapt the spreader position to the container position during the final downward motion. This would allow the spreader to have initially a deviation of about 5 cm in the X- or Y-direction. The final position accuracy, as required to enter the twistlock pins into the corner castings, will be then about 1 cm.
A stiff guiding mechanism would make the coupling system kinematically overdetermined. To avoid a fine X-Y motion control of the whole gantry crane the spreader device has hydraulic cylinders, which allow some extra horizontal motion of the spreader. During the coupling procedure the cylinders are released.
- When placing a container on an AGV. This vehicle has slanted guiding corners, which allow it to have an X-Y deviation of 5 cm (or worse). The cylinders of the spreader can be released during the final part of the motion.

- When coupling with the container on the AGV. Now the guiding corners of the spreader cause the adaptation, comparable with retrieving a container in the yard.

The required horizontal position accuracy (overall) of the gantry crane can be estimated finally to about 5 cm. It is reasonable then that the guiding mechanism shall contribute to it by a much smaller amount, preferably not more than 1 cm. This measure will be requested both for kinematic and dynamic accuracy.

5 Kinematic analysis

The existing 6-bar mechanism has been analyzed to verify its kinematic motion. A kinematic model was made for a standard mechanism computer program (RUNMEC). Figure 4, left part, shows the mechanism in some positions, making clear that no collision occurs between the arms DE or EF and the neighbour containers.
The straight line, generated by point E (hinge with the spreader), has a deviation of maximally 1 cm over a length of 8 m, see fig. 4 right part. On first sight this mechanism works satisfactorily, there is no need to improve the position accuracy.
A discussion point is the kinematic transfer quality. Point E is driven along its path by the cables, the mechanism follows passively. In [Klein Breteler, 1995] it was stated that in such cases the first order transfer function s' (geometric velocity) is a suitable measure for this kinematic transfer quality. The value s'=0 will lock the mechanism for whatever definition of the independent variable of the passive mechanism. In case of the exactly symmetric mechanism this situation can be recognized easily: the bars BD and DE or GF and FE are then in-line. For this particular mechanism the (complements of) the angles BDE and GFE can be

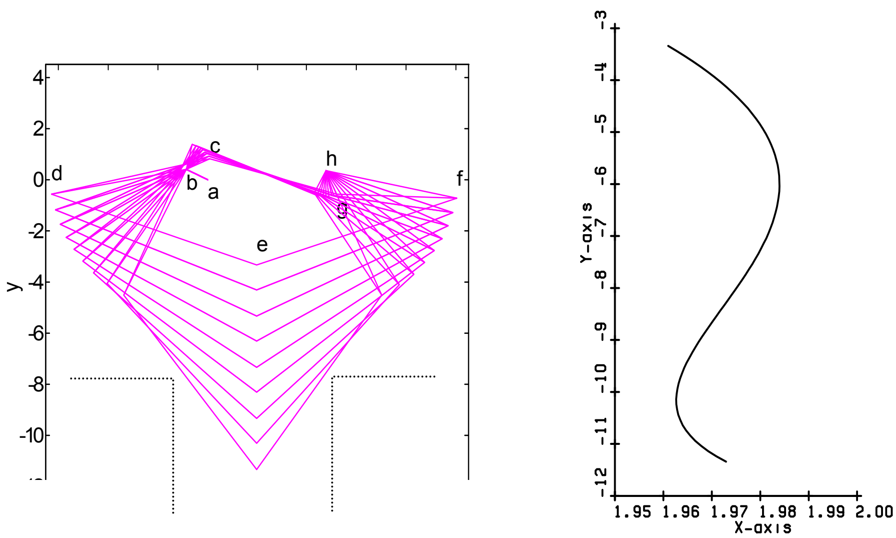


Figure 4. Kinematic motion and kinematic accuracy of the straight line (point e)

regarded as pressure angles. They look reasonable, that means not more then about 45°. It can be expected that the optimized mechanism, which is nearly symmetric, has no transfer problems either. The conclusion is that the current kinematic scheme is acceptable. The mechanism with the given dimensions can make a three-high stack without collision problems.

6 Dynamic analysis

Static displacements are caused by static forces, which are here inertia force and wind force. As a worst case situation the following assumptions were made:

- The biggest container type (40 feet) container, fully loaded, is to be displaced. The estimated mass of container and spreader is 50,000 kg. The mass centre lies eccentric at 2 m distance of the geometric centre of the container.
- Maximum acceleration/deceleration 0,5 m/s^2.
- Maximum wind force of 11 Beaufort (30 m/s).

These assumptions introduce static maximum forces of 28,500 N in lane driving direction (X-axis) or 42,000 N in trolley driving direction (Y-axis). The last mentioned force causes also a moment of 50,000 Nm around the Z-axis.

Vibration amplitudes can be predicted when the mechanical system can be reduced to a single mass-spring system. This is only meaningful under some conditions like:

- A substantial portion of the total mass is located at the output link (this is certainly the case here). The (lowest) natural frequency is needed then to know.
- The system is relatively stiff (high rate of motion period and vibration period).

For a typical step motion (dwell-rise-dwell) the motion period can be understood as the time to move from the begin to the end position. For all smooth motions the response lies then in a narrow band of rest-amplitudes [Koster, 1973]. A worst case situation can be defined assuming such a characteristic step motion. Here for instance: a step h_m=1m is to be taken in t_m=5s (this would be achieved with a constant acceleration and deceleration of 0,16 m/s^2). The diagram specifies then the natural frequency that the elastic mechanism must have minimally. With the assumed characteristic motion the rest amplitude should be 0,01 of h_m. The rate of the vibration period and the step time, as read from the diagram on the horizontal axis, is 0.2. The required period of the natural frequency is then 0.2x5s=1s. The demanded lowest natural frequency is thus 1 Hz.

The dynamic demands should be applied to all positions of containers in the yard. This means that all vertical positions of the guiding point should be considered. For simplification three vertical positions were considered further: upper, middle and lower.
It is very likely that the X- and Y-displacement of the ASC with maximum acceleration will occur only with the container in the upper position (safety reasons). The worst case condition is thus to be applied mainly to the upper position.

Proceeding with the design the following steps have been taken (using the RUNMEC program for the first three steps) [Wiersma, 1997].

- A planar model for force analysis was made towards a construction with truss bars (only normal forces), see figure 3 right part. Compared with the kinematic model, the triangle BCD was modified such that realistic internal forces are expected.
- The worst case static forces (only X-direction) were applied in many positions and the highest normal force in any bar was determined. The result was –104 kN in bar 3.
- Normal stiffness was chosen such that the maximal horizontal deviation of the guided point was about 1 cm. The same stiffness was applied to all beams. Some attempts were necessary to satisfy this condition.
- A beam profile was selected that has sufficient normal stiffness and bending stiffness against buckling. Each bar in the planar model will be represented in the coplanar model

by two identical bars, which together have the chosen normal stiffness. Material choice: steel, square cross-section of 100 mm and 4 mm wall thickness.

- A constructive design was made based on the chosen bar profile. The crossbars in this spatial construction have the same profile.
- The construction was entered in a standard computer program for finite elements (ANSYS). The three positions were treated as three different constructions.
- The worst case static load was applied to the X- and Y-direction individually, and the static displacements were determined. Some improvements to the crossbar layout were made to minimize their number.
- Finally the vibration modes and natural frequencies were analyzed with ANSYS. Picture 5 gives an impression of the amplitudes concerning the lowest natural frequency, 1.32Hz (upper) and 0.55 Hz (lower) respectively. This is acceptable.

7 Conclusion

Obviously a much lighter construction then the present one is possible. Compared with free swinging the lowest natural frequency can be increased about four times by applying the guiding mechanism. For smooth motions of the crane the swinging amplitude of the container will be about 1 cm in worst case circumstances.

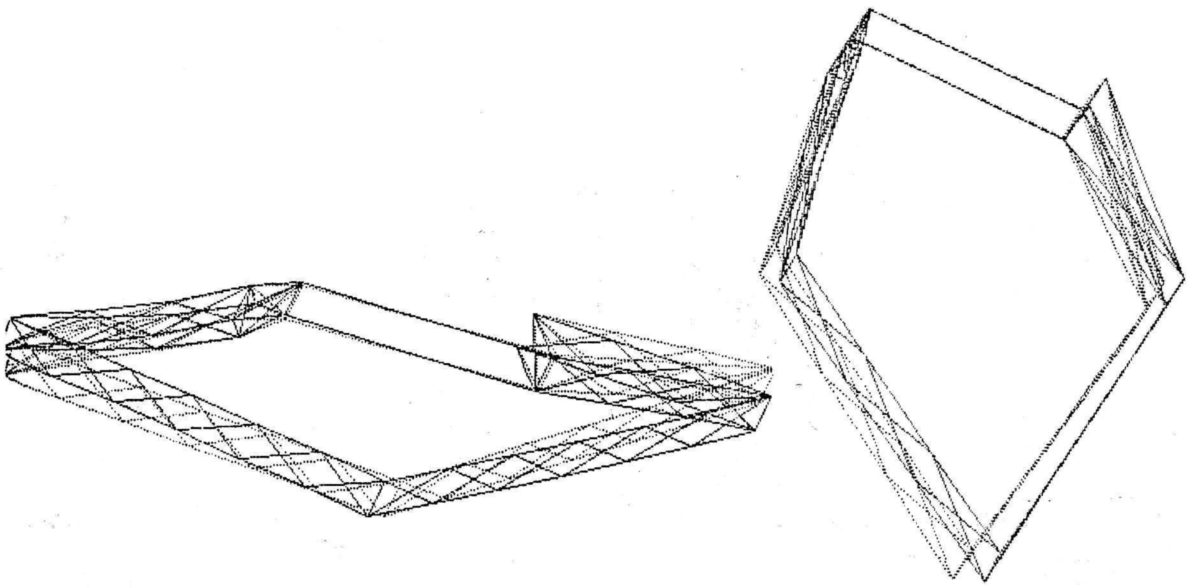


Figure 5 Vibration modes in the upper and lower position of the guiding mechanism

References:

Klein Breteler, A.J., Prediction of jamming in multi-link mechanisms designed to perform rectilinear motion. Proceedings of 9th world congress of IFToMM, Polytecnico di Milano, 1995, pp 188-192.

Wiersma, T.L., Link mechanism for a stacking crane (in Dutch). Report 97.3.TT.4971, University of Technology Delft/Faculty WbMT, The Netherlands, 1997.

Koster, M.P., Vibrations of cam mechanisms and their consequences on the design. Thesis University of Technology Eindhoven, The Netherlands, 1973.