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Design procedure of reach stacker control system

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Abstract. Reach stackers are efficient and maneuverable machinery for an overload of overall containers in cargo terminals and ports today. Such machinery design determines the practical interest of the engineering industry specialists. However, the lack of information in open sources about structural or kinematic diagrams of equipment control mechanisms and recommendations for their calculation does not allow an effective approach to the creation and update of reach stackers.

We developed the design procedure of the reach stacker control system, which considers the kinematic and geometric parameters of the container spreader, as a result of the conducted research. It can be used in the design stages of mechanisms for spreader lateral displacement, extension of the spreader sections, spreader inclination in the vertical plane, container fixation by spreader, spreader rotation, lifting-lowering and telescoping of the boom, as well as during the real operation modes of reach stackers in cargo terminals and ports.

Keywords: reach stacker, container, spreader, mechanism, hydraulic cylinder, force, moment.

INTRODUCTION

Cargo terminals and ports still use gantry and portal cranes, as well as large lift capacity forklifts for the overload of overall containers [1–4]. But they are gradually being replaced by more maneuverable reach stackers, which provide a growing level of container traffic.

The reach stacker is a special type of forklift that has some external features of the hydraulic truck crane by the telescopic boom presence, at its end fixed the device for gripping containers (spreader). The main parameters of the reach stacker are the lift capacity, the container tier numbers into which they are stacked, the wheelbase of the tractor and the working weight [5–7]. Additional characteristics include the ability to work with different spreader types, engine power, transmission type, and running equipment features.

The reach stacker design of different producers is the same type, although some models have certain design features, which consist of changing the relative placement of the cabin and the boom, as well as the boom shape. This allows them to be used for loading containers into the holds of watercraft, working with containers placed on railway platforms or special trailers (at the same time, operations are carried out in the reach stacker side part).

The reach stacker contains a long-base wheeled machine usually, on its frame the boom with the spreader and the cabin are placed in the longitudinal axis plane (Fig. 1).

The cabin can be moved to the machine front in some models. The spreader boom and mechanisms control are carried out by the hydraulic system. The container gripe by the spreader is carried out by rotary locks that are fixed in the container fittings.

Fig. 1. Overload operations by reach stacker

The production of reach stackers is well established in European and Asian countries and the USA. The most famous producers are Ferrari (Italy), Liebherr (Germany), Kalmar (Sweden), Terex (USA) and others.

The significant number of functional and economic advantages of reach stackers in comparison with the existing equipment for container overload causes practical interest among project organizations, producers and specialists in the industry of modern mechanical engineering and logistics systems [2, 3–6]. However, there are no structural or kinematic diagrams of the working equipment control mechanisms and recommendations for their calculation in the available information reference and literary sources.

PURPOSE OF THE PAPER

The purpose of the paper is to develop the design procedure of the reach stacker control system, taking into the weight, kinematic and geometric parameters of the working equipment and the gripping device.

RESEARCH RESULTS

The analysis of the existing structures of reach stackers made it possible to identify several mechanisms in the control system of the working equipment and spreader. These included the spreader lateral displacement, extension of the spreader sections, spreader inclination in the vertical plane, container fixation by spreader, spreader rotation, lifting-lowering and telescoping of the boom.

The spreader lateral displacement mechanism is designed for the possibility of accurately placing containers on top of each other when forming tiers at warehouse sites [8].

We assume for the calculation that the spreader displacement occurs with the gripped container (Fig. 2).

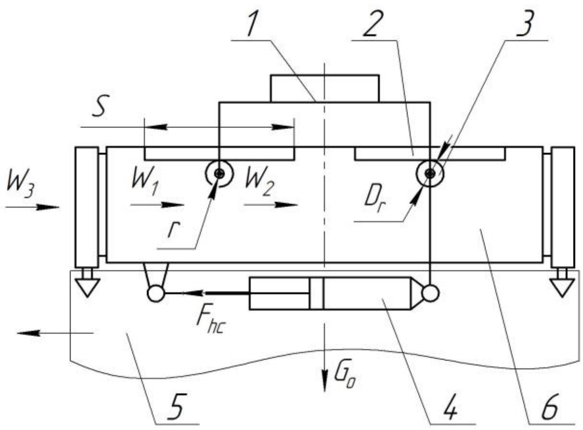

Fig. 2. Diagram of spreader lateral displacement mechanism: *1* – upper frame; *2* – guide; *3* – roller; *4* – hydraulic cylinder; *5* – container; *6* – main frame with retractable sections

The spreader lateral displacement is carried out by one hydraulic cylinder, its rod connected to the main frame of gripper 6, and the case to the upper frame 1 through an elongated bracket. Rollers 3 are made on the upper frame, on which the main frame 6 with retractable sections is attached, on its end parts L-shaped guides 2 are made. The spreader

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movable part rolls along the rollers and moves to the side by the required amount when the hydraulic cylinder rod is extended and retracted ($S_1 \pm 800$ mm in both directions usually).

The next condition must be met to ensure the possibility of spreader lateral displacement

$$F_{hc} \geq W_n,$$

where F_{hc} – force on the control hydraulic cylinder rod; W_n – total resistance to spreader lateral displacement.

The total resistance to spreader lateral displacement is determined by equation

$$W_n = W_1 + W_2 + W_3,$$

where W_1 – resistance when rolling spreader on rollers; W_2 – frictional resistance in roller pins; W_3 – resistance from inertial forces.

We determine the resistance when rolling the spreader on the rollers by equation

$$W_1 = \frac{2 \cdot G_0 \cdot f}{D_r},$$

where G_0 – total load on the rollers, $G_0 = \left(m_{fr} + m_{r.s} + m_{cont}\right) g$, here m_{fr}, $m_{r.s}$, m_{cont} – mass of the main frame, retractable sections and container respectively [9]; f – friction coefficient of the guide roller rolling, taken for the «steel – steel» pair [10]; D_r – roller diameter.

We determine the frictional resistance in roller pins by expression

$$W_2 = \frac{2 \cdot G_0 \cdot f_p \cdot r}{D_r},$$

where f_p – friction coefficient of rolling in the roller pin [11]; r – radius of the roller pin.

We determine the resistance from the inertia forces by equation

$$W_3 = \frac{G_0 \cdot a}{g},$$

where a – spreader acceleration at the beginning of the lateral displacement.

The amount of the required force on the control hydraulic cylinder rod F_{hc} is taken depending on the obtained value W_n.

We determine the diameter of the hydraulic cylinder piston by expression [12]

$$D_p = 2 \cdot \sqrt{\frac{F_{hc}}{\pi \cdot p_{nom} \cdot \eta_{hc}}}, \qquad (1)$$

where p_{nom} – nominal pressure in the reach stacker hydraulic system; η_{hc} – overall efficiency of the hydraulic cylinder, $\eta_{hc} = 0,98$.

The stroke of the hydraulic cylinder rod is taken based on the desired value of the spreader lateral displacement, i.e., $L_{rod} = 2S_1$ [10]. The hydraulic cylinder with the diameter of the piston D_p and rod d_{rod} is accepted according to current standards.

The mechanism for extending spreader sections (Fig. 3) is intended for the possibility of gripping containers 20’...40’ type or other.

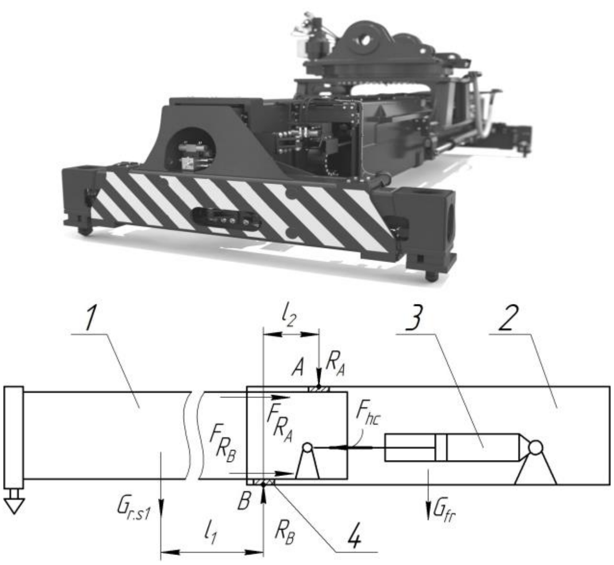

Fig. 3. Container spreader and design diagram of mechanism for extending sections: *1* – retractable section; *2* – main frame; *3* – hydraulic cylinder; *4* – support (slide plate)

The section extension drive is made in the form of hydraulic cylinders 3, mounted on the gripper main frame 2, which pushes the sections to the desired outreach. The calculation diagram of the mechanism for extending sections conventionally depicts the extension of one section, since the extension of another is similar and carried out synchronously.

Let's determine the reactions in the supports of section 4 R_A and R_B. To do this, we will draw up the equations of the active reactions relative to point *A*:

$$\sum M_A = 0; \quad R_B \cdot l_2 - G_{r.s1} \cdot (l_1 + l_2) = 0;$$

$$R_B = \frac{G_{r.s1} \cdot (l_1 + l_2)}{l_2},$$

where $G_{r.s1}$ – weight of the retractable section, determined by $G_{r.s1} = m_{r.s1} \cdot g$, here $m_{r.s1}$ – mass of the retractable section [9]; l_1, l_2 – arms of active forces.

Let's find the reaction R_A:

$$\sum M_B = 0; \quad R_A \cdot l_2 - G_{r.s1} \cdot l_1 = 0;$$

$$R_A = \frac{G_{r.s1} \cdot l_1}{l_2}.$$

The required force on the hydraulic cylinder rod is

$$F_{hc} = \left(F_{R_A} + F_{R_B}\right) \cdot k_s,$$

where F_{R_A}, F_{R_B} – friction forces that arise in the supports *A* and *B* respectively during the section extension are determined by

$$F_{R_A} = R_A \cdot f; \quad F_{R_B} = R_B \cdot f,$$

where f – friction coefficient of sliding for the «section – sliding plate» pair, depending on the adopted materials, there may be «steel – bronze», «steel – kaprolon» pair [10]; k_s – safety factor, $k_s = 1{,}1 \ldots 1{,}25$.

The calculated piston diameter of the hydraulic cylinder for the extending section is determined by equation (1).

The stroke of the rod is taken based on the dependence

$$L_{rod} = \frac{L_{cont}^{40'} - L_{cont}^{20'}}{2},$$

where $L_{cont}^{40'}$, $L_{cont}^{20'}$ – length of 40' and 20' type containers respectively or other types that the spreader is designed to handle.

It is impractical to accept the stroke of the rod according to the standard, since high accuracy of rotary locks positioning for the retractable sections relative to the container fittings is required. The hydraulic cylinder with the diameter of the piston D_p and rod d_{rod} is accepted according to the current standards.

The spreader rotation mechanism is designed to facilitate the process of gripping the container and its precise setting when formed tiers or when setting on trailers [13, 14].

Considering this, we will calculate the rotation mechanism of the reach stacker gripping device by the kinematic diagram (Fig. 4) taking into account the mass of the gripped container with cargo for the designed equipment of the machine.

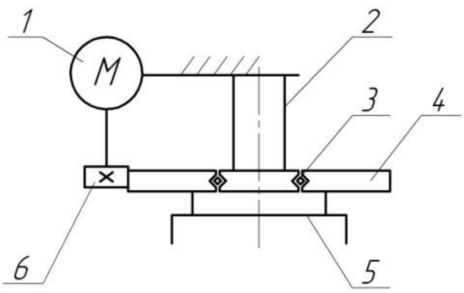

Fig. 4. Kinematic diagram of spreader rotation mechanism: *1* – hydraulic motor; *2* – adapter frame; *3* – bearing unit; *4* – gear ring; *5* – upper frame (spreader rotating part); *6* – gear

The moment of resistance to the spreader rotation during start-up (when working with the maximum-size container for the projectable stacker) is determined by expression

$$M = M_1 + M_2,$$

where M_1 – moment of resistance to the spreader rotation, which is created by inertia

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forces (dynamic moment); M_2 – rotation resistance moment with the container, which is created by friction forces in the mechanism.

We determine the rotation resistance moment which is created by friction forces in the mechanism by equation

$$M_1 = 0,5 \cdot \mu \cdot D \cdot \frac{\sum N}{\cos\beta},$$

where μ – reduced friction coefficient in the rolling bearings of the support bearing unit; D – diameter of the rolling raceway for the support bearing unit; β – inclination angle of the transverse axis of the rollers to the vertical; $\sum N$ – total load on the rollers.

The total load on the rollers is

$$\sum N = \frac{G_r}{\cos\beta} \cdot \left(1 - \frac{2\varphi}{\pi} + 8 \cdot \frac{e \cdot \sin\varphi}{\pi \cdot D}\right),$$

where G_r – resultant of external loads [9]

$$G_r = \left(m_{fr} + m_{r.s} + m_{dr} + m_{cont}\right) g,$$

where m_{fr} – mass of the main spreader frame; $m_{r.s}$ – mass of retractable section; m_{dr} – mass of drive elements; m_{cont} – mass of the container with cargo according to the nominal lift capacity of the designed reach stacker; e – distance from the rotation axis to the resultant of external loads

$$e = M_t / G_r,$$

where M_t – torque

$$M_t = m_{cont} \cdot l_1 \cdot g,$$

where l_1 – distance from the rotation axis to the maximum displacement point of the cargo gravity center in the container, approximately can be taken $l_1 = \frac{l_{cont}^{\max}}{4}$, here $l_{cont}^{\max}$ – container maximum length of the fit type.

We define the parameter φ using an expression $\varphi = \arccos\frac{D}{4 \cdot e}$ [1].

The moment of resistance to the spreader rotation with the container, which is created by inertia forces

$$M_2 = I \cdot E,$$

where E – angular acceleration

$$E = \frac{2a}{l_{cont}^{\max}},$$

where a – minimum angular acceleration of the spreader; I – inertia moment of the spreader rotating masses [13]

$$I = \gamma \cdot \beta \cdot \left(m_{cont} \cdot \frac{l_{cont}^{\max}}{4} + m_{r.s} \cdot \frac{l_{spr}^{\max}}{4}\right),$$

where $l_{spr}^{\max}$ – spreader width when the sections are retractable to the maximum; value $\frac{l_{cont}^{\max}}{4}$ and $\frac{l_{spr}^{\max}}{4}$ set the distance from the rotation axis of the spreader to the mass centers of the corresponding elements; γ – inertia coefficient of the rotation part; β – inertia coefficient of the turning mechanism.

The maximum torque applied to the rotation axis of the spreader is within the limits

$$M' = (0,85...0,92)M.$$

The braking torque of the spreader rotation

$$M_{br} = \omega^2 \cdot \frac{1+r}{2 \cdot r \cdot \beta'} \cdot I \cdot g,$$

where r – coefficient of averaging acceleration and braking moments; β' – spreader rotation angle, $\beta' = 195$ degrees usually [5].

We determine the hydraulic motor power of the spreader rotation mechanism by equation

$$N_{hm.rot} = \frac{M' \cdot \omega}{10^3 \eta_{rot}},$$

where ω – angular speed of the spreader rotating part; η_{rot} – efficiency of the turning mechanism depending on the unit design.

The hydraulic motor-reducer with the required technical characteristics is accepted after the power calculation $N_{hm.rot}$.

Time to turn the spreader with the container at an angle β' [14]

$$t_{rot} = \frac{g \cdot I \cdot \omega^2}{248 \cdot N_{hm} \cdot \eta_{rot}} \cdot \left(1,37 + \eta_{rot}^2\right) + \frac{\beta'}{\omega},$$

where N_{hm} – power of the hydraulic motor-reducer for the spreader turning mechanism.

The gear ratio of the «gear ring – gear of the hydraulic motor» stage [11].

$$U_r = \frac{n_{hm}}{n_{spr}},$$

where n_{hm} – shaft speed of the hydraulic motor-reducer; n_{spr} – shaft speed of the spreader rotating part.

The teeth number of the gear ring and gear of the hydraulic motor and other geometric parameters, acting torques, etc. determined by kinematic calculations next.

The lifting-lowering and telescoping mechanism of boom sections is designed to increase the lifting height when increasing the tiers of containers [2]. The design diagram of the sections telescoping mechanism and the boom lifting-lowering is presented in Fig. 5.

We determine the necessary force on the rod of the boom telescoping hydraulic cylinder

$$F_{hc}^{T} = \left(G_{b.s2} + G_{cont}\right) \cdot \sin\alpha,$$

where α – maximum angle of telescopic boom inclination, usually is $\alpha = 60°$; $G_{b.s2}$, G_{cont} – weight of the retractable boom section and a container respectively, $G_{b.s2} = m_{b.s2} \cdot g$, $G_{cont} = m_{cont} \cdot g$ [9].

Two hydraulic cylinders are used in the boom telescoping mechanism, so the required force on one rod will be $F_{hc}^{T1} = \dfrac{F_{hc}^{T}}{2}$.

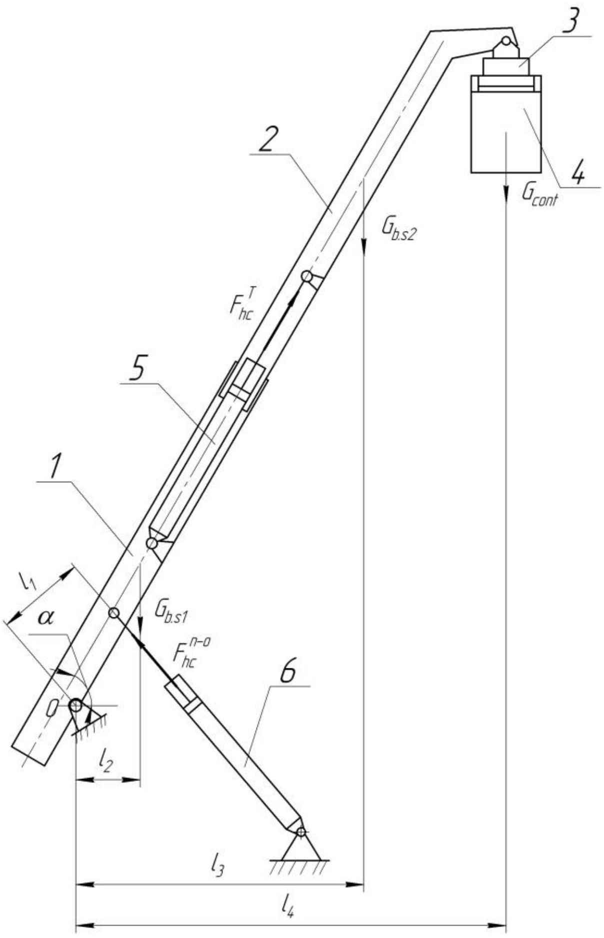

Fig. 5. Design diagram of boom lifting-lowering and telescoping mechanism: *1* – main boom section; *2* – retractable boom section; *3* – spreader; *4* – container; *5* – boom telescoping hydraulic cylinder; *6* – boom lifting-lowering hydraulic cylinder

We will formulate the equation of the moments for the acting forces relative to point *O* to determine the required force on the rod of the boom lifting-lowering hydraulic cylinder

$$\sum M_O = 0;$$

$$F_{hc}^{n-o} \cdot l_1 - G_{b.s1} \cdot l_2 - G_{b.s2} \cdot l_3 - G_{cont} \cdot l_4 = 0.$$

Where from

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$$F_{hc}^{n-o} = \frac{G_{b.s1} \cdot l_2 + G_{b.s2} \cdot l_3 + G_{cont} \cdot l_4}{l_1},$$

where $G_{b.s1}$ – weight of the boom main section, $G_{b.s1} = m_{b.s1} \cdot g$; l_1, l_2, l_3, l_4 – arms of force action relative to point O respectively.

Two hydraulic cylinders are used in the boom lifting-lowering mechanism, so required force on one rod will be $F_{hc}^{n-o.1} = \frac{F_{hc}^{n-o}}{2}$.

The calculated piston diameter of the telescoping and lifting-lowering hydraulic cylinder is determined by equation (1). The stroke of the hydraulic cylinders rod for lifting and lowering the boom can be determined graphically after designing the mechanism, considering the need to transfer the boom from the horizontal position (transport) to the working (α angle position). The stroke of the hydraulic cylinders rod for telescoping the boom is determined graphically after the arrangement of the mechanism, considering the required lifting height of the container (number of tiers) [12, 15]. Hydraulic cylinders with the diameter of the piston D_p and rod d_{rod} are accepted according to current standards.

CONCLUSIONS

Thus, the proposed design procedure allows you to perform calculations of the reach stacker control system for the working equipment and the gripping device. This allows you to reasonably choose the drive devices, taking into the kinematic and geometric parameters of the spreader for moving and stacking large overall containers. It can be used at mechanisms construction stages, as well as during the real operation modes of handling equipment in cargo terminals and ports.

At the same time, it is taken into the weight characteristics of the gripped container with cargo and the reach stacker design elements. However, the separate task is to check the performance of the main reach stacker parts and units from load action during overload operations with overall containers.

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Методика розрахунку системи керування річстакером

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Анотація. Річстакери сьогодні є ефективною та маневреною технікою для перевантажування габаритних контейнерів у вантажних терміналах і портах. Очевидним є те, що проєктування такої техніки зумовлює практичний інтерес фахівців машинобудівної галузі. Однак відсутність у відкритих джерелах інформації про конструктивні чи кінематичні схеми механізмів керування обладнанням та рекомендацій щодо їх розрахунку не дозволяє здійснити ефективний підхід до створення й модернізації річстакерів.

В результаті проведених досліджень нами розроблено методику розрахунку системи керування річстакером, що враховує кінематичні та геометричні параметри роботи контейнерного спредера, і може використовуватися як на стадіях конструювання механізмів для бічного зміщення спредера, висування секцій спредера, нахилу спредера у вертикальній площині, фіксації контейнера спредером, повороту спредера, підйому/опускання та телескопування стріли, так і за режимів реальної експлуатації річстакерів у вантажних терміналах і портах.

Ключові слова: річстакер, контейнер, спредер, механізм, гідроциліндр, зусилля, момент.