Skewing of crane during acceleration

Josef Musilek
Institute of Technology and Business in České Budějovice, Department of Civil Engineering, Okružní 517/10, 370 01 České Budějovice, Czech Republic
musilek@mail.vstecb.cz

Abstract. During the work of crane and during its driving on the crane track, the so-called skewing of the crane arises. The physical phenomenon of crane skewing has been interpreted in various ways in the past. This corresponded to various physical models of skewing, which were created by various authors and which were presented in the standards for the design of crane structures and tracks. Current standards define the skewing as the so-called oblique running of the crane along the track with uniform movement. This is a situation where the force on the front guide means returns the obliqued crane when driving to the straight position. Theoretical formulas for the calculation of forces caused by skewing during oblique running were determined by Hannover. These relationships apply to movement of the crane with constant velocity along the crane track. The author of the article derived a model of a crane, which allows to investigate the movement of the crane along the crane track in general, it means both during uniform movement and during its acceleration or deceleration. It is a dynamic model that allows the solution of equations of motion to solve the coordinates corresponding to the degree of freedom and, among other things, to determine the forces between the crane and the crane track. This article presents the results obtained from this model for a specific selected crane during its acceleration. The contact of the front guide wheel is assumed, which is in accordance with the model derived by Hannover and which is also stated in the currently valid standards.

1. Introduction
According to current standards, the skewing of a crane is described as an inclined run of a crane along a crane track, where the front guide means (for example the rim wheel or roller guides of a crane) is in contact with the side of the rail. It is assumed that the crane is inclined to the track and moves in a uniform motion. During this travel, a so-called return force is created on the front guide means, which tries to return the crane back in the straight-ahead direction of travel, thus creating transverse frictional forces under the crane wheels. These forces have a negative impact on the wear of the crane wheel rims or guide rollers. They must also be taken into account when designing the crane structure and the crane track structure. The formulas for calculating these forces were derived by Hannover [1]. Its computational model and formulas have also taken over current standards, such as [2] or [3].

The author of this article created a computational model that investigates the movement of a crane in general and as one of the possible results it is possible to obtain horizontal transverse forces between the crane and the track [4]. It is a dynamic model that investigates the movement of the crane in general due to the driving torques of the crane motors. The model is created from rigid bodies. The bodies are connected by rotation springs. The compliance of the steel structure is simulated by springs and the drive forces are simulated by motors torque controlled by frequency controller. This model is presented in and
is shown on Figure 1. In this article, this model is used to solve a specific crane whose parameters are given in the next chapter. The solution is made for the situation of acceleration of the crane. So the initial situation is, that the crane is standing in skewed position with contact of the wheel rim on the first wheel and due to the torque moment on drive motors the crane start moving.

![Crane model](image)

**Figure 1.** Computational model used for simulation

2. **Crane used for simulation – parameters of the crane**

The crane considered for the simulation is a double girder crane with bridges designed as a BOX. The end carriages are made of RHS profiles. The carrying capacity of the crane is 16000 kg. The crane span is considered to be 12.5 m and the wheelbase of the crane is considered to be 3.15 m. The position of the crab’s hook in relation to the axis of the crane track girder is considered as the minimum range of the hook, which is 1.04 m.

The model further considers the length of the rope, which is twisted between the crab's drum and the load. The length of this rope is considered 1.5 m. The weight of one crane bridge is 6850kg. The weight of one end carriage is 510kg. The weight of a crab traveling on crane bridges is 5000 kg.

The flexibility of the steel structure is considered using equivalent springs. The stiffness of the rotating spring between the bridge and the end carriage is 2.7 x 10^7 N/m, the stiffness simulating the flexibility of the end carriage at its end is 2.2x10^7 N/m.

The slip constant is defined in accordance with the theory according to Hanover. Its size is 2.53x10^6 N on wheels 1 and 4 and 6.6x10^6 N on wheels 2 and 3. The diameter of the crane wheels is 400mm. The angle of the wheel hubs to the side of the rail is 2.8 °. The half gap between the ankle and the side of the rail is 5mm.
The model also considers the effect of driving resistances. The rolling resistance arm is considered to be 0.7 mm. The coefficient of pin friction is considered to be 0.16. The crane travel bearings are rolling ball with a diameter of 62 mm. The crane is driven by two independent asynchronous motors with a gearbox mounted on the travel wheel shaft. The gear ratio of this gearbox is 32 and its efficiency is 0.95. For the dynamic calculation, it is also necessary to know the moment of inertia of the rotor of the bridge, which is 0.011 kg / m². The rated motor torque is 25.9 Nm. It is a four-pole motor and its rated speed is 1460 1 / min at a rated frequency of 50 Hz.

3. Results of simulation

The solution of the equations of motion of the above model is performed by the numerical method Runge-Kutta 4th order. The initial position of the crane is such that the crane is turned to the maximum within the clearances of the crane wheels. The half gap between the wheel trim and the side of the rail is 5mm, so the initial rotation of the crane is:

$$\phi_0 = \frac{10\text{mm}}{3150\text{mm}} = 3,175 \cdot 10^{-3}$$

Due to the above-mentioned rotation of the crane before the start, the first end carriage is in front of the second one before starting, by the value:

$$y_{10} = 19,3m \cdot 3,226 \cdot 10^{-3} = 0,062m$$

The rest of the unknown coordinates calculated are zero.

The Figure 2 the angle between the crane and the crane runway. The figure shows the crane rotation on the crane runway during its motion. It is possible to see that after the end of acceleration the angle becomes constant.

![Figure 2. Angle between the end carriages of the crane and the crane runway](image)

The Figure 3 shows the moment on motor shaft. The picture shows that during the start-up the limit value of the torque on the motor 39N / m was reached, which the frequency converter is able to provide for a short time. Furthermore, the end of the start-up time is clearly visible, when the driving torque of
the motor decreased and then stabilized at a constant value of 10N/m, which corresponds to a uniform movement at constant driving resistances.

Figure 3. Torque moment on drive motor shaft

The Figure 4 shows the skewing forces between the crane wheels and the crane track. The force F2 represents the guiding force between the crane guide means (wheel rim), the forces R1 to R4 are the friction forces between the crane wheels and the rail. The figure shows a large increase in the guide force F2 during the acceleration of the crane and its subsequent decrease and stabilization at a constant value as the crane gets into a uniform movement with a constant angle of inclination.

Figure 4. Lateral forces caused by skewing of the crane
4. Conclusions
The article presents the results of the simulation of a dynamic model, which was created by the author of this article. The simulation was performed for a specific crane and its results are presented in graphical form using graphs, which are the output of the numerical solution of the equations of motion of the model.

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