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CONTENT

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● «основное значение» – будет присвоено, если средний уровень шума при производстве работ, а также отсутствует мусор на строительной площадке и незначительное количество выхлопов в атмосферу;

● «верхнее значение» – будет присвоено, если низкий уровень шума при производстве работ, отсутствует мусор на строительной площадке и отсутствуют выхлопы в атмосферу.

5. Безопасность производства.

Данный параметр характеризуется безопасностью при выпуске оборудования, безопасностью при эксплуатации оборудования и при выполнении бетонных работ при контурном строительстве.

Уровни варьирования:

● «нижнее значение» – будет присвоено, если не обеспечивается безопасность при выпуске оборудования, при эксплуатации оборудования и при выполнении бетонных работ при контурном строительстве;

● «основное значение» – будет присвоено, если обеспечивается безопасность при выпуске оборудования, при эксплуатации оборудования и выполнении бетонных работ при контурном строительстве с применением дополнительного оборудования и/или возможно при применении иных специальных работ;

● «верхнее значение» – будет присвоено, если обеспечивается высокая безопасность при выпуске оборудования, при эксплуатации оборудования и при выполнении бетонных работ при контурном строительстве, без применения дополнительного оборудования.

Вывод

Полученные параметры на основании опроса экспертов и специалистов в области контурного строительства, с вариативными значениями, позволят сформировать математическую модель для выбора вида контурного строительства.

В заключении хотелось бы отметить, что вышеизложенное позволило сформулировать следующую задачу: формирование математической модели для выбора метода контурного строительства, в зависимости от полученных экспертным опросом параметров.

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JUSTIFICATION OF PARAMETERS OF RUNNING WHEELS OF AN OVERHEAD CRANE OF VARIOUS DESIGNS

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ABSTRACT

The article discusses the issues of substantiating the optimal parameters of traveling wheels of overhead cranes of various designs. The estimation of the optimal taper of bevel running wheels is given and the resistance of movement is determined.

Keywords: running wheel, bridge crane, taper, movement resistance.

Traveling wheels for cranes, crane trolleys are made of various designs, different sizes, from different materials. Some groups of travel wheels are standardized, others are manufactured according to the standards of factories.

As practice shows, in some cases, typical crane equipment does not have the necessary durability [1]. Many scientists have studied the dynamics of hoisting machines [2–4]. The running wheels of bridge cranes need special attention [5, 6]. Crane wheels are the fastest wear element of the crane [7, 8].

On bridge, gantry and other cranes, as a rule, cylindrical double-flange wheels are used. On railway and other cranes in the presence of intersection of rails, in some on bogies of bridge and gantry cranes, single-flange running wheels are used. If the direction of movement of the crane is made using special rollers, then the wheels are made without flanges. These rollers are installed on the opposite side of two racks, or two rollers span one rack. The rolling surfaces of the guide roller can be spherical with a radius of curvature of 250–350 mm. The width of the tread surface of the flangeless running wheel must be at least 60 mm greater than the width of the rail head.

The use of flangeless running wheels with guide rollers significantly reduce friction losses, since the rolling friction of the roller on this rack is less than the sliding friction of the flange on the rail, and, accordingly, the installed power of the motors of the travel mechanism decreases and the service life of the running wheels is significantly increased.

A slope of two bevel gears with total axle rigidity towards the axle was considered by B.S. Kovalsky in work [9], the speeds of wheels 1 and 2 are respectively equal:

$$V_1 = V\left(1 + \frac{k}{D_0}y\right) \quad (1)$$

$$V_2 = V\left(1 - \frac{k}{D_0}y\right) \quad (2)$$

Where V is the speed of the center of the ramp, which is assumed constant, $V=const$. Warning wheel 2, wheel 1 creates a skew of the slope at a speed φ':

$$\varphi' = -\frac{V_1 - V_2}{L} = -\frac{2k}{D_0 L}y \quad (3)$$

Under the law of motion of the center of the slope $y=y(x)$ we have for φ the value $\varphi=y'(x)$ and since $\varphi'=\varphi' V$:

$$y'' + p^2 y = 0 \quad (4)$$

$$p^2 = \frac{2k}{D_0 L} \quad (5)$$

Where

$$y = C_1 \cos px + C_2 sinpx. \quad (6)$$

Which for $y_0=y_0(0)$ gives

$$y = y_0 cos\, px \quad (7)$$

In this case, we have a harmonious movement with a wavelength:

$$l= \frac{2\pi}{p} \quad (8)$$

In the case of a four-wheel overhead crane traveling on two driven bevel wheels and two bevel or cylindrical idle wheels, B.S. Kowalski received [10]:

$$y = y_0 e^{-ax} \cos px, \quad (9)$$

$$\varphi = p y_0 \sin px. \quad (10)$$

Where

$$l = \frac{2\pi}{p}, \quad (11)$$

$$p = \sqrt{\frac{2k}{DL}}, \quad (12)$$

$$n = \frac{p^2 K}{4}. \quad (13)$$

With the base of the crane $K \to 0$, we have $n \to 0$, $e^{-ax} \to 1$ and the dependences (9), (10) and (11) transform into the equations of motion of the ramp.

Dependencies (9) – (13) quite accurately correspond to the movement of the center of the crane and can be used regardless of the direction of movement of the crane. B.S. Kowalski proved that experiments confirm not only the constancy of the movement of the crane on bevel wheels but also the damping nature of the trajectory of the center of the crane, and is the content of the "effect of bevel drive wheels" or "taper effect".

Dependencies (9) and (10) can be supplemented. So, for the speed of the transverse movement of the crane we have:

$$V_y = p y_0 e^{-ax} \sin px. \quad (14)$$

With p=0,1 m^{-1} and y_0 = 0.03 m, Vy = 0.003 should be. To move the idle wheel we have:

$$y_{xx} = y(x) + \varphi(x)\frac{K}{2}. \quad (15)$$

What gives

$$\max y_{xx} = y_0\left[1 + \frac{(pK)^2}{8}\right]. \quad (16)$$

At p=0,1, K=5m, we have y_{xx}=1.04 and the gap that is needed between the rail and the flange of the drive wheel. Some authors recommend accepting the width of idle wheels more than driving ones. R. Garry, on the contrary, recommends reducing the width of idle wheels in order to reduce the skew of the driving wheels, which in some cases causes friction of the flanges.

The given dependences (9) - (13) are sufficient to represent the movement of the crane and can be used in experimental studies. In this regard, let us consider the experiments carried out G. Broughton. For a crane with a lifting capacity of 5 tons, we have: L=25,9 м, K=5,2 м, D=0,76 м, k=0,125.

In this case:

$$p = \sqrt{\frac{2 \cdot 0{,}125}{30 \cdot 1020}} = 0{,}11, \text{ n=0,0163 m},$$

$$l = \frac{2\pi}{p} = \frac{2\pi}{0{,}11} = 57{,}1 \text{ m}.$$

Experimental data l=61 m.

Measurements of the trajectory of the crane movement give y_1=y_0=50 m, y_1=25 m.

These data differ from the law of motion according to (9) and B.S. Kowalski proposed to limit the y_1/y_3 ratio by setting the order.

Some experts talk about the possibility of installing wheels with reverse taper, that is, according to scheme 6 (a), instead of solution 9 (b). At the same time, the results of experiments at the «Elektrostal» plant and studies of the operation of several cranes for 10 years at the «Norilsk Mining and Processing Plant» are used. B.S. Kowalski explains this by the fact that in some cases the cranes work satisfactorily, since the movement of the cranes takes place on a part of the track without the participation of flanges until the gaps between the rails and flanges are exhausted.

In reality, both cases are fundamentally different. In case (a) we have the equation of motion of the slope $y''+p^2y=0$, which indicates a stable motion, in case (b) we will have $y''- p^2y=0$ and unstable motion.

The effects of non-parallelism and curvature of rails and frequencies have been studied by crane designers, although track conditions are significantly affected in many cases.

In order to represent the real crane path by a proper approximation, various "assumptions" are made, for example in S. Bobov the calculated crane path consists of the segments of a way with direct parallel rails returned concerning each other. At work V. Sobolev, the rails of the crane track are represented by sinusoidal arcs of the sin $\pi x/l$ type, where l is the path length of the order of 30–40 m and more.

Since the length of the rails and crane girders requires fairly frequent assessments of the state of the rails, a different solution to the problem is needed. B.S. Kowalski suggested using Fourier rows to represent the crane runway. If y is the lateral displacement of the crane, and y' is the deviation of the rail from the design position, then the equation of motion $y''+p^2y=0$ will be different $y''- p^2y=0$.

Comparison of the solutions of the above equations allows one to estimate the influence of y', and y' is written by the Fourier series according to the instrumental survey of the rail track. Such a solution to the problem was proposed in 1939 by B.S. Kowalski and implemented in V.M. Ivanov, tracks were surveyed at several plants in Donbass, processing was carried out for six members of the Fourier series, different taper values appeared in the solution of the equations.

When choosing the taper k, take into account its influence on the value of the lateral loads of the rail, on the value of the eccentricity of the normal load relative to the rail axis $e=kr/2$ at k=0,08-0,10-0,125 and r=400 mm, we have e=16-20-25 mm for the impact k and the trajectory of the crane. From the research of V.M. Ivanov, the values k=0,08-0,10 follow, previously the value k=0,125 was taken, which significantly increased the lateral movement of the crane, some authors recommend the value k=0,20-0,30, but these values are not entirely accurate.

B.S. Kowalski wrote that the tolerance for the choice of taper can be quite wide, but the tolerance for the taper of paired drive wheels should be as minimal as possible.

With a satisfactory state of the track, high-quality manufacturing of cranes, the latter work well on cylindrical double-flange wheels, however, track defects, the difference in the diameters of the drive wheels in terms of the track are sharply noted in the wheel performance. The correct movement of the cranes is ensured by the flanges, in case of defects in the track and the crane, the intensity of the flanges leads to increased wear, in our practice the bulk of the cylindrical double-flanged wheels are rejected due to the wear of the flanges.

If the difference in the diameters of the driving wheels is m, the wheel skew γ, the ratio of the crane span to its base $\tau=L/k$, then each revolution of the running wheel corresponds to its displacement across the rail by $\delta=m\pi/\tau+\gamma\pi D$. When using cylindrical double-flanged wheels, τ=6.

When $m=10^{-3}D$ and $\gamma=0{,}6\cdot10^{-3}$ we have:

$$\delta = 10^{-3}\pi\frac{D}{6} + 0{,}6\cdot10^{-3}\pi D = 2{,}42\cdot10^{-3}D. \quad (17)$$

For example, with D=630 mm, we have δ=1,52 mm. With a symmetrical installation of the wheel on the rail, the distance from the flange to the wheel is 25/2=12.5 mm, that is, the gap will be exhausted after 12,5/2,42=5 wheel revolutions. In reality, the situation is different, since at the distance between the crane rails the tolerance is 15 mm. In this case, it is possible for the crane to jam on the rails, or when the crane moves, the path increases or decreases due to the deformation of the track and the supporting structure.

Heavy wear of the flanges is a sign of insufficient attention to overcoming wheel skews and the quality of track manufacturing. The tolerances that apply to the laying of rails must be tighter.

The resistance to movement of workshop overhead cranes is generally equal to:

$$W = W_t + W_y. \quad (18)$$

For cranes in an open area, a wind load W_b is added. Resistance on slopes:

$$W_y = iG. \quad (19)$$

Where i is the average slope;

G – is the weight of the crane.

In general, i=0-0,002, sometimes more.

The frictional resistance is:

$$W_t = aw_tG. \quad (20)$$

Where

$$w_t = \frac{2k+fd}{D}. \quad (21)$$

Coefficient α is introduced to take into account the friction of the flanges, α=1,5÷2, sometimes more, f is the reduced coefficient of friction in bearings, usually f=0,001–0,002, d is the bearing diameter, k is the coefficient of rolling friction of the rack wheel.

The value of k depends on the dimensions of the contact plane. In the case of a running wheel on a flat rail, on the basis of Dumas's experiments, $k=0{,}1\cdot b$, with a contact strip width of $2b$.

For an elliptical contact area, it is assumed that $k=0{,}1\cdot\alpha$, where α is the semi-axis of the ellipse oriented along the movement of the crane.

The dimensions of the contact plane are determined by the formula of G. Hertz, which takes into account the magnitude of the load P, the modulus of elasticity.

We have:

$$k = 0{,}82\cdot10^{-4}\sqrt{\frac{PD}{B}}. \quad (22)$$

There are noticeable differences in the assessment of the coefficient of friction f in the wheel bearings, meaning the use of rolling bearings. There are f=0,002 (Huser), 0,01 (NKMZ), 0,015 (VNIIPTMASH), 0,02 (UZTM, STM).

B.S. Kowalski believed that the value f=0,01 is quite convincing and somewhat overestimates the real value of friction. This does not exclude the possibility of a different estimate of f taking into account the special operating conditions of the crane, for example, the ambient temperature.

In general, the movement of the crane is skewed. With bevel wheels, distortions are localized in harmonious movement, or there is a weak contact between the flanges and the rail heads. With cylindrical wheels, considerable friction arises between the flanges and the crane rail and the resistance to crane movement increases markedly. In addition, frictional forces act on the pressure-shifted flange during full sliding. Estimation of additional loads using the flange coefficient α is complicated by the fact that in the expression $w_r=\alpha w_0$, the values of α and w_0 can be varied simultaneously.

Higher f values can decrease a. For the case of four-wheel cranes with bevel drive wheels, a=1 is taken, for cranes on cylindrical wheels, the flange ratio is a=1,5÷1,6 with a central one and a=1,9÷2,0 with a separate drive.

The resistance w_{yk} is taken equal to $w_{yk}=tg\gamma=\gamma$, where γ is the angle of inclination of the rail, which is formed by assembly tolerances and deformations of the supporting structures of the beams. This calculation should be clarified, since the resistance is different for the four wheels of the crane and the value of the resistance:

$$w_{yk} = (\gamma_1 + \gamma_2 + \gamma_3 + \gamma_4)/4. \quad (23)$$

With sufficient rigidity of the bridge, we are talking about raising the center of gravity of a loaded crane, which on average is much less than the maximum.

The preservation of the flanges of the traveling wheels implies additional protection for the crane, in particular this explains the tradition of using double-ribbed traveling wheels. With the reliable operation of the rollers, the running wheels can be made without flanges. This greatly facilitates and reduces the cost of the movement mechanism.

In the case of bevel double-flange travel wheels, the direction of movement of the cranes is selected due to the tapering effect of the two drive wheels with a tapered tread surface. Idle wheels can be cylindrical. Since the flanges are not involved in the work, the service life of the travel wheels increases dramatically. Reduction of lateral loads, reduced dynamics of movement, better adaptability to defects in crane runways allow us to recommend their use in many cases in practice. It is also necessary to take into account a significant reduction in the energy consumption of cranes, a decrease in the loads of the movement mechanism.

The resistance to crane movement is determined by the formula:

$$W_t = (1{,}1 \div 1{,}2) w_t G. \quad (24)$$

Bevel wheels can be flangeless, but horizontal rollers should be used to protect against wheel derailment.

Resistance to movement in this case will be determined by the formula:

$$W_t = w_t G + 2 S w_k. \quad (25)$$

In ideal designs with full realization of the taper effect, the second term in formula (25) is meaningless. But on defective sections of the crane track, it is possible to include rollers in the work, therefore, we keep formula (25) in the same form, bearing in mind that, based on the strength of the rollers, there is a much lower number of revolutions than in the case of cylindrical wheels, about 10 %.

Conclusions. The engine power is selected in accordance with the resistance to the movement of the crane, but when starting the crane, the engine must overcome the mass of the crane or the masses reduced to the running wheels, the rotating drive elements of the travel mechanisms. Permissible approximately double short-term overload of the motor (the motor is checked by the starting torque), with a greater value of inertial load, a motor with a higher power should be used.

In some cases, it is possible to reduce inertial loads by increasing the acceleration time, for example, by correcting the electrical circuit. At the same time, it should be taken into account that in the power flow of the engine-gearbox-travel wheels, a significant part of the inertial load is absorbed by the flywheel moment of the engine, the effect of inertia forces is the less, the further from the engine is the element designed for strength.

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