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Design rules for tower cranes

塔式起重机设计规范

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Design rules for tower cranes

1 Subject content and scope of application

This Standard specifies the basic criteria and calculation methods that the design calculation for tower cranes should follow. All other calculation methods that are proved to be correct and reliable by theory and practice can also be used.

This Standard is applicable to various types of power-driven tower cranes for various uses.

This Standard is not applicable to the tower cranes converted by auto, tyre and crawler cranes.

2 Normative references

GB 699 *Quality carbon structural steels - Technical conditions*

GB 700 *Carbon structural steels*

GB 755 *Rotating electrical machines - General technical requirements*

GB 985 *Basic forms and sizes of weld grooves for gas welding, manual arc welding and gas-shielded arc welding*

GB 986 *Basic forms and sizes of weld grooves for submerged arc welding*

GB 998 *Basic testing method of low voltage apparatus*

GB 1591 *Low alloy structural steels*

GB 10051.1 *Lifting hooks - Mechanical properties, lifting capacities, stresses and materials*

GB 1231 *Specifications of high strength bolts with large hexagon head, large hexagon nuts, plain washers for steel structure*

GB 1300 *Steel wires for welding*

GB 3077 *Alloy structure steel - Technical requirements*

GB 3098.1 *Mechanical properties of fasteners - Bolts, screws and studs*

GB 3323 *Methods for radiographic inspection and classification of radiographs for fusion welded butt joints in steel*

GB 3632 *Sets of torshear type high strength bolt hexagon nut and plain washer for steel structures*

GB 3811 *Design rules for cranes*

GB 5117 *Carbon steel covered electrodes*

GB 5118 *Low alloy steel electrodes*

GB 5144 *Safety rules for construction tower cranes*

GB 10054 *Builder's hoist - Specification*

GB 10055 *Safety code for builder's hoist*

GB 11352 *Carbon steel castings for general engineering purpose*

JJ 3 *Design rules for construction winches*

JJ 12.1 *Specification for welding quality of construction machinery*

TJ 7 *Code for design of industrial and civil building foundation*

JJ 40 *Fixed fill fluid coupling for tower cranes*

JJ 75 *Lifting equipment - Design requirements of the hook anti-off pawl*

3 Symbols and codes

3.1 Loads

F - Concentrated load, force;

p - Pressure;

M - Bending moment, moment;

T - Torque.

3.2 Limit values for checking

σ - Calculated tensile and compressive stresses;

$[\sigma]$ - Allowable stress of the materials;

σ_s - Yield point of the materials;

σ_b - Tensile strength of the materials;

$\sigma_{0.2}$ - Test stress when the residual strain of the standard tensile test for materials is up to 0.2%;

τ - Calculated shear stress;

$[\tau]$ - Allowable shear stress of the materials;

σ_{rk} - Fatigue strength limit;

E - Elastic modulus of the materials;

$[\lambda]$ - Allowable slenderness ratio of the structural members;

λ - Slenderness ratio of the structural members;

σ_s - Maximum calculated tensile and compressive stress amplitudes;

τ_a - Maximum calculated shear stress amplitude;

$[\sigma_a]$ - Allowable tensile and compressive fatigue stress amplitudes;

$[\tau_a]$ - Allowable shear fatigue stress amplitude.

3.3 Geometric parameters

l, L - Length, distance;

h - Height;

D, d - Diameter;

R, r - Radius;

b - Width;

e - Eccentricity;

I_a - Cross-sectional moment of inertia;

J - Moment of inertia;

W - Section bending modulus of the structural members;

A - Windward area of the structure, cross-sectional area of the structural members;

P - Thread pitch, rope groove pitch;

δ - Thickness;

Δ - Displacement;

θ - Angle;

V - Volume.

3.4 Calculation coefficients

K, k - Dimensionless coefficient;

K_n - Safety factor;

K_f - Load spectral coefficient;

K_s - Structural stress spectral coefficient;

K_m - Mechanism load spectral coefficient;

μ - Friction coefficient, length coefficient of the structural members;

α, β, f - Coefficients;

C_δ - Flexibility;

C_W - Wind coefficient;

C - Wire rope selection coefficient;

C_0 - Reduction factor of unequal end bending moment;

C_H - Transverse load bending moment coefficient;

ω - Structural filling rate;

η - Windshield reduction factor;

ϕ - Stability coefficient of the structural members subjected to axial compression;

ψ - Correction factor of axial compression stability;

ϕ_W - Lateral buckling stability factor of the bending structural members;

ϕ_1 - Hoisting impact factor;

ϕ_2 - Hoisting dynamic load factor;

a. For Item a. in the normal operation above, CALCULATE v_h according to the maximum slewing speed at which the motor is unloaded, and DETERMINE ϕ_{2e}.

b. For Item b. in the normal operation above, CALCULATE v_h according to more than 0.5 times the maximum slewing speed at which the motor is unloaded, and DETERMINE ϕ_{2e}.

4.2.1.2.3 CONSIDER the dynamic load generated when the suspended hoisting item is suddenly completely or partially unloaded. TAKE the unloading impact factor ϕ_3 multiplied by the hoisting load F_Q into consideration. CALCULATE according to the Equation (3).

$$\phi_3 = 1 - 1.5\frac{\Delta m}{m} \tag{3}$$

Where:

Δm - Unloaded hoisting mass, kg;

m - Hoisting mass, kg.

4.2.1.3 Running impact factor

When the tower crane or its hoisting car is running, the collision load generated due to the unevenness of the track is taken into consideration by the running impact factor ϕ_4 multiplied by the deadweight load F_g and the hoisting load F_Q. SELECT ϕ_4 according to Table 7.

Table 7 Running Impact Factor ϕ_4

Running speed, m/s	Impact factor ϕ_4
≤ 1	1.1
> 1	1.2

4.2.1.4 Load caused by the acceleration (deceleration) of the transmission mechanism

a. The load in the load bearing structure or mechanism, due to the acceleration (deceleration) of the transmission mechanism (such as mechanism startup or braking), can be calculated by means of rigid body dynamics. SEE Annex E (reference). During calculation, it is assumed that the hoisting mass is fixed at the end of the boom or on the crane carriage.

b. In order to consider the impacts of the elastic vibration of the system in the process of acceleration (deceleration) on the load increase, the load variation ΔF caused by the driving force of the transmission mechanism shall be

F_W - Wind load acting on the tower crane and the item (F_{W1}, F_{W2}), N;

C_W - Wind coefficient, determined according to Article 4.2.2.1.3;

p_W - Calculated wind pressure, calculated according to Article 4.2.2.1.2, Pa;

A - Windward area perpendicular to the wind direction, calculated according to Article 4.2.2.1.4, m^2.

When determining the load combination, it is assumed that the wind load acts on the most unfavorable position.

4.2.2.1.2 Calculated wind pressure p_W

a. In the open area, the relationship between the wind pressure and the wind speed at a height of 10m from the ground can be calculated according to the Equation (5).

$$p_W = 0.613 v_W^2 \tag{5}$$

Where:

p_W - Calculated wind pressure, Pa;

v_W - Wind speed, m/s.

b. Calculated wind pressure under operating condition can be divided into two categories: p_{W1} and p_{W2}.

p_{W1} refers to the calculated wind pressure under normal operating condition, which is used for selecting the resistance calculation of the motor power as well as the fatigue strength and heat checking of the mechanical parts. p_{W1} = 150Pa.

p_{W2} refers to the maximum calculated wind pressure under operating condition, which is used for calculating the strength, stiffness and stability of the mechanical parts and metal structures, and for checking the overload capacity and overall anti-tipping stability of the transmission. p_{W2} = 250Pa.

In special cases, such as for special purposes or for the areas with stronger wind, higher calculated wind pressure value can be specified according to the agreement between the user and the manufacturing plant.

4.2.2.1.3 Wind coefficient C_W

a. Wind coefficient C_W of the monolithic structure and single component shall be selected according to Table 8.

b. For the spatial structure composed of two parallel plane trusses, the wind coefficient of the monolithic structure can be taken as that of the overall structure, while the total windward area is calculated according to Article 4.2.2.1.4.

c. Wind blows in the diagonal direction of the spatial structure of the rectangular cross section. When the side ratio of the rectangular cross section is less than 2, the wind load is taken as 1.2 times the wind force acting on the long side of the rectangle. When the side ratio of the rectangular cross section is greater than or equal to 2, the wind load is taken as the wind force acting on the long side of the rectangle.

d. The wind load of the spatial structure of the triangular cross section is calculated as 1.25 times the wind force of the projection area perpendicular to the wind direction.

e. For the spatial structure of the triangular cross section (such as horizontal beam boom) whose upper and lower chord members are square steel tubes and whose web members are circular steel tubes, under the action of lateral wind, TAKE the wind coefficient C_W = 1.3.

4.2.2.1.4 Windward area *A*

The windward area is calculated by the projection area of the structural member on the plane perpendicular to the wind direction.

a. The windward area *A* of the monolithic structure can be calculated according to the Equation (6).

$$A = \omega A_1 \tag{6}$$

Where:

A - Windward area of the structure, m^2;

ω - Structural filling rate, selected according to Table 9;

A_1 - Structure outline area, as shown in Figure 1, m^2.

b. For two pieces of same type of parallel structures with equal height, the windshield action of the front piece to the posterior piece shall be taken into account. The total windward area is calculated according to the Equation (7).

Table 8 Wind Coefficient C_W of the Monolithic Structure

Serial No.	Structural style	C_W
1	Plane trusses made of shape steel (filling rate ω = 0.3 to 0.6)	1.6

> 20 to 100	1 100
> 100	1 300

Note: ① In the table, p_{W3} = 800Pa is equivalent to the wind pressure value in Shanghai.

② In special cases, if using in the areas where larger windstorms are likely to occur, higher calculated wind pressure may be specified according to the agreement between the user and the manufacturing plant.

4.2.3.2 Test load F_t

The test load refers to the overload load acting on the tower crane during the performance test, which is divided into static test load F_{st} and dynamic test load F_{dt}.

The static test load F_{st} is calculated according to 1.25 times the rated hoisting capacity; while the dynamic test load F_{dt} is calculated according to 1.1 times the rated hoisting capacity.

During calculation, the dynamic test load F_{dt} shall be multiplied by the dynamic load factor ϕ_6. ϕ_6 is calculated according to the Equation (10).

$$\phi_6 = 0.5\ (1 + \phi_{2n}) \tag{10}$$

Where:

ϕ_{2n} - Hoisting dynamic load factor, SEE Article 4.2.1.2.

4.2.3.3 Collision load F_c

When the tower crane or crane carriage is in collision with the buffer, the collision load F_c acting on the structure is calculated according to the kinetic energy absorbed by the buffer. TAKE the running speed of the tower crane or crane carriage before the moment of collision as 0.7 to 1.0 times the maximum normal operating speed. A small value is taken when a reliable automatic deceleration control device is provided.

The collision load F_c can be calculated according to a rigid body motion model (regardless of the effects of the suspended item) and multiplied by the elastic vibration load factor ϕ_7 to consider the effects of the system's elastic vibration. For the spring buffers commonly used by the tower cranes, ϕ_7 can be taken as 1.25; for other buffers, ϕ_7 can be selected between 1.25 and 1.6 according to its performance.

4.2.3.4 Loads caused by sudden shutdown

The loads caused by sudden shutdown (such as power outage) shall be calculated according to the provisions of Article 4.2.1.4. During calculation, CONSIDER according to the mechanism's most unfavorable operating condition at the moment of shutdown (which is the most unfavorable combination of mechanism acceleration and deceleration as well as hoisting load). At this time, ϕ_5 is selected between 1.5 and 2.

4.2.4 Other loads

Other loads include the installation load, loads acting on the work platforms and channels, and transport load.

4.2.4.1 Installation load

The installation loads at all stages must be calculated according to the structural pattern, installation and disassembly procedures of the tower crane. During calculation, the calculated wind pressure p_{We} is taken as 100Pa.

4.2.4.2 Loads on the work platforms and channels

On the work platforms that might store materials and tools, CALCULATE based on bearing 3,000N of concentrated load;

For the structures that are only used as channels, CALCULATE based on bearing 1,500N of concentrated load;

For the railings, CALCULATE according to 1,000N of horizontal concentrated load.

4.2.4.3 Transport load

The transport load refers to the load acting on the components (such as overall hauling wheel set) of the tower crane during transportation. During calculation, the effects of the actual dynamic load in transport state shall be taken into account.

4.3 Anti-tipping stability

4.3.1 Checking of the operating condition

The anti-tipping stability of the tower cranes shall be checked according to the operating conditions listed in Table 12.

In the process of installation and disassembly, the anti-tipping stability shall check the dangerous states at all stages according to the structural pattern, installation and disassembly procedures of the tower crane.

Under the non-operating condition (storm invasion), the anchoring device or rail clamping device must be used to prevent the tower crane from sliding along the rail under the maximum wind force under the non-operating condition. The anti-sliding condition is as follows:

$$F_{bo} > 1.1F_{W3} + F_{s1} - F_{\mu} \tag{12}$$

Where:

F_{bo} - Braking force (or anchoring force) along the rail direction generated by the rail clamping device and brake wheels, N;

F_{W3} - Maximum wind force along the rail direction under the non-operating condition, N.

When designing the rail clamping device, the friction coefficient of the rail clamp (scratches on the surface and quenched) and the rail is taken as 0.25. The maximum operating force of the manual rail clamping device shall not be greater than 200N.

4.5 Determination of the support reaction

When determining the support reaction of the tower crane, the combined operation of the system composed of the tower crane, chassis, landing legs and foundation shall be taken into account. For the attached and internal climbing tower cranes, the combined operation with the system composed of building supports shall also be taken into account.

4.5.1 Running tower cranes and fixed tower cranes with chassis

4.5.1.1 Calculation of the vertical support reaction

The vertical support reaction of the bearings or wheels (trolleys) is generally calculated according to the four-point support of the rigid chassis. When the support reaction of a wheel is negative, it shall be recalculated according to the three-point support.

4.5.1.2 Calculation of the horizontal support reaction

The absolute value of the horizontal support reaction (generated by the braking of the running mechanism) of the tower crane running on the rails along the rail direction shall not be greater than the adhesive force of the brake wheels and the rails.

The wheels' horizontal support reaction perpendicular to the rail direction shall be calculated according to 0.075 times the maximum vertical support reaction.

4.5.2 Attached tower cranes

For the crane body attached to the building, the support reaction shall be calculated according to the multi-span continuous beam of the elastic support, which is the load of the attachment device.

The support reaction of the first attachment point on the upper part of the crane body (support end of the cantilever on the crane body) is the maximum. This reaction value shall be taken as the calculated load of the attachment device and building support device.

4.5.3 Internal climbing tower cranes

When calculating the support reaction of the internal climbing tower cranes, DETERMINE the simplified mechanical model according to different specific support programs. For the case where the upper and lower support frames are normally used to support the internal climbing tower crane, the upper frame can be considered as mobile rigid hinged support, while the lower frame can be considered as fixed rigid hinged support. However, the torque is supported by the upper frame.

4.6 Rails and foundation

4.6.1 Rails

The rail type used by the rails of the tower cranes is related to the wheel diameter and calculated load. The calculated load of the wheels shall be calculated according to Article 6.4.4.1.

When the running mechanism is at a working level of M_1 to M_3 and a running speed of 15 to 20 m/min, SEE Annex F (reference) for the combination of the wheels and rails.

4.6.2 Rail foundation

The rail foundation can use gravel or concrete foundation according to work needs.

The rail foundation must be able to bear the maximum load under the operating and non-operating conditions.

4.6.3 Fixed foundation

The concrete foundation applied to the fixed tower cranes shall be designed to meet the anti-tipping stability and strength conditions.

The concrete foundation strength shall be calculated according to TJ 7.

4.7 Transport

The transport of the tower cranes must comply with relevant provisions on rail and road transport.

The axial load of the hauling wheel set of the tower crane under overall hauling shall not exceed 1.3×10^5N. Each hauling wheel set shall have a parking brake to ensure reliable braking on the slope of 6%.

4.8 Counterweight

For the upper rotary tower cranes, the counterweight mass shall be determined according to the principle of acting minimum load on the crane body.

For the lower rotary tower cranes, the counterweight mass shall be determined according to the conditions of anti-tipping stability.

4.9 Safety protection

The design and calculation of the tower cranes shall meet the requirements of GB 5144.

5 Structure

5.1 Calculation principle

This standard uses the allowable stress method for calculation. The metal structural members shall be subjected to strength, stability and stiffness calculations, and shall meet its specified requirements. During calculation, the plasticity of the materials shall not be taken into account in general.

The fatigue strength of the structural members and connections shall be calculated in accordance with the provisions of Article 5.7.

5.2 Working levels of the structure

The working levels of the structure are related to the stress status (nominal stress spectral coefficient) and number of stress cycles (stress cycle level) of the structural members.

The load spectrum and the number of operating cycles of the tower crane are the basis for determining the stress spectrum and the number of stress cycles of the structural members. The working levels of the structure and the tower crane are not necessarily the same, depending on the circumstances.

Combination C considers the various combinations of various loads specified in Articles 4.2.1, 4.2.2 and 4.2.3 for the strength and elastic stability calculations of structural members and their connections.

The values, directions and acting positions of various loads shall be combined according to the most unfavorable conditions of the checked structures and connections.

5.3.3 Load combinations during installation, erection and transportation

The installation load specified in Article 4.2.4.1 shall be combined according to the actual situation. When checking the load-bearing member, the allowable stress of the material shall be treated in accordance with the combination C as specified in Article 5.3.2.

The transport load specified in Article 4.2.4.3 shall be combined according to the actual situation. When checking the load-bearing member, the allowable stress of the material shall be treated in accordance with the combination A as specified in Article 5.3.2.

5.4 Materials and their allowable stresses

5.4.1 Materials of the structural members and their allowable stresses

5.4.1.1 Materials of the structural members

The major bearing structural members in the metal structure of the tower crane shall use Q235 plain carbon structural steel complying with GB 700, 20 steel complying with GB 699, 16Mn and 15MnTi steel complying with GB 1591.

For the steel applied to the bearing structural members, the grade of steel and material shall be selected based on the importance of the structure, load characteristics, connection methods, operating temperature and other different conditions.

Killed steel shall be applied to major bearing structural members of the tower crane. The materials shall have the assurance of conformity for the impact toughness at room temperature. For the tower cranes operating in the regions below -20°C, Q235 and 20 steel applied to the major bearing structural members shall have the assurance of conformity for the impact toughness at -20°C; 16Mn and 15MnTi steel shall have the assurance of conformity for the impact toughness at -40°C. The assurance of conformity for the cold bending test is required if necessary.

Steel castings shall use the steel conforming to the provisions of GB 11352.

5.4.1.2 Allowable stresses for the materials of the structural members

TAKE the basic allowable stresses 〔σ〕$_A$, 〔σ〕$_B$ and 〔σ〕$_C$ determined by the corresponding load combinations as the allowable tensile, compressive and bending stresses for the materials of the structural members. The allowable shear stress and allowable end face pressure-bearing stress shall be determined according to Table 19.

a. When the ratio OF the yield point σ_s TO the tensile strength σ_b of the material $\frac{\sigma_s}{\sigma_b}$ is less than 0.7, the safety factor and basic allowable stress corresponding to various load combinations shall be determined according to Table 19.

$$\sigma = \frac{F'_{Nj}}{A_p} < [\sigma] \qquad (18)$$

Where:

σ - Calculated stress, Mpa;

A_p - Net area of the checked cross section, mm^2;

F'_{Nj} - Calculated according to the following equation:

$$F'_{Nj} = F_{Nj}\left(1 - 0.4\frac{n_1}{n}\right)$$

Where:

F_{Nj} - Axial force acting on the connector, N;

n - Number of high strength bolts at one end of the structural member or symmetrically connected with one side;

n_1 - Number of high strength bolts at the checked cross section (at the outermost bolt of the connector).

5.5.2 Strength calculation of the connections

5.5.2.1 Weld connection

The strength of the butt weld subjected to tension (compression) and shear force shall be calculated according to the Equation (19).

$$\sigma_W = \sqrt{\sigma^2 + 2\tau^2} < [\sigma_W] \qquad (19)$$

5.5.2.2 High-strength bolted connections

The high-strength bolted connections are able to bear shear force, tension, moment and combined loads.

5.5.2.2.1 Bear the shear force

In this case, the shear force that the connector is subjected to is transmitted through the friction between the friction surfaces of the connector.

The connection strength of the connector shall be calculated according to the Equation (20).

F_N - External tension that a single bolt is subjected to, calculated according to the Equation (22), N.

$$F_N = \frac{F'_N}{n} \quad (22)$$

Where:

F'_N - External tension that the connector is subjected to, N;

n - Number of bolts in the connector.

b. The bolt strength shall be calculated according to the Equation (23).

$$\sigma_L = \frac{F_1 + k_c F_N}{A_{d_1}} \leqslant (\sigma_L) \quad (23)$$

Where:

σ_L - Calculated tensile stress of the bolt, Mpa;

k_c - Coefficient, related to the stiffness of the bolts and connected parts; during steel connection, the use of metal gaskets or no gaskets, TAKE k_c = 0.2 to 0.3;

A_{d1} - Cross-sectional area calculated by the thread's inner diameter, when the diameters of other parts of the screw are less than the thread's inner diameter or the screw is hollow, A_{d1} refers to the minimum cross-sectional area, mm^2;

(σ_L) - Allowable stress of the bolt, which can be calculated according to the Equation (24), Mpa.

$$(\sigma_L) = \frac{\sigma_s}{K_{n1}} \quad (24)$$

Where:

σ_s - Yield point of the bolt material, selected according to Table 24, Mpa;

K_{n1} - Safety factor, when controlling the preload, TAKE K_{n1} = 1.2 to 1.5; when not controlling the preload, SELECT according to Table 25.

Table 24 Strength of the Bolt Material **Mpa**

Performance level	Tensile strength σ_b	Yield point σ_s
8.8	830 to 1 030	660 to 820
10.9	1 040 to 1 240	940 to 1 110

The stability of the structural members subjected to axial compression shall be checked according to the Equation (27).

$$\sigma = \frac{F_N}{\phi \cdot A_R} \leq (\sigma) \qquad (27)$$

Where:

F_N - Calculated axial pressure, SEE Annex G (reference), N;

A_R - Gross cross-sectional area of the structural member, mm^2;

ϕ - Stability coefficient of the structural member subjected to axial compression selected according to the structural member's maximum slenderness ratio λ or hypothetical slenderness ratio λ_f or maximum conversion slenderness ratio λ_h, SEE Annex A (supplement).

5.6.1.2 Slenderness ratio

a. The slenderness ratio of the structural member shall be calculated according to the Equation (28).

$$\lambda = \frac{l_c}{r} \leq (\lambda) \qquad (28)$$

Where:

l_c - Calculated length of the structural member, SEE Annex H (reference) for the calculation method, mm;

r - Ratio OF the gross cross-sectional area of the structural member TO the slewing radius of an axis, calculated according to the Equation (29), mm;

(λ) - Allowable slenderness ratio of the structural member, SEE Table 26.

$$r = \sqrt{\frac{I_a}{A_R}} \qquad (29)$$

Where:

I_a - Inertia moment of the structural member to the gross cross-sectional area of an axis, mm^4;

A_R - Gross cross-sectional area of the structural member, mm^2.

b. When the yield point σ_s of the material is greater than 345Mpa, the slenderness ratio λ can be approximately replaced by the hypothetical

$$\frac{F_N}{A_R\varphi\psi}+\left[\frac{1}{1-\frac{F_N}{0.9F_{Ex}}}\right]\frac{C_{ox}M_{ox}+C_{Hx}M_{Hx}}{\varphi_W W_x}\leq [\sigma] \quad (32)$$

$$\frac{F_N}{A_R\varphi}\leq [\sigma] \quad (33)$$

Where:

ψ - Correction factor of axial compression stability, its value can be calculated according to the Equation (34), or CHECK the Annex A (supplement);

A_R - Gross cross-sectional area of the structural member, mm^2; it refers to the sum of the normal cross-sectional area of the main leg in case of the truss structure;

C_{ox}, C_{oy} - Reduction factor of unequal end bending moment, calculated according to the Equations (37) and (38);

M_{ox}, M_{oy} - End bending moment of the structural member, N • mm;

C_{Hx}, C_{Hy} - Transverse load moment factor, SEE Annex I (reference);

M_{Hx}, M_{Hy} - Maximum bending moment caused by transverse load in the structural member, N • mm, when M_H and M_o are in reverse direction and $|C_H M_H| < 2C_o M_o$, M_H is taken as zero;

W_x, W_y - Bending modulus of the structural member's cross section, mm^3;

φ_W - Lateral buckling stability factor of the bending structural members, which can be selected according to the provisions of Article 5.6.2.2.

$$\psi=\frac{0.9F_E-F_N}{0.9F_E-\varphi[\sigma_s A_R(1-\varphi)+F_N]} \quad (34)$$

Where:

F_E - Euler critical loads F_{Ex} and F_{Ey}, whichever is smaller, F_{Ex} and F_{Ey} shall be calculated according to the Equations (35) and (36), N.

$$F_{Ex}=\frac{\pi^2 EA_R}{\lambda_x^2} \tag{35}$$

$$F_{Ey}=\frac{\pi^2 EA_R}{\lambda_y^2} \tag{36}$$

$$C_{ox}=0.6+0.4\frac{M'_{ox}}{M_{ox}}\geqslant 0.4 \tag{37}$$

$$C_{oy}=0.6+0.4\frac{M'_{oy}}{M_{oy}}\geqslant 0.4 \tag{38}$$

Where:

$\frac{M'_{ox}}{M_{ox}}$, $\frac{M'_{oy}}{M_{oy}}$ - Ratio of the end bending moments at both ends of the structural member, with respective positive and negative signs, and the absolute value is not greater than 1.

For the space truss structural members and spliced combined structural members, $\varphi\psi$ = 1 is taken during calculation. Furthermore, ADD the additional bending moment $F_N\Delta_o$ generated by the structural member's initial displacement Δ_o caused by the manufacturing error to the M_{ox} or M_{oy} in the second and third items of the Equation (31).

5.6.2.2 Lateral buckling stability checking of the bending structural members

5.6.2.2.1 Lateral buckling stability checking of the bending structural members may not be performed when one of the following conditions is met:

a. When the plates (steel plates or other plates) are laid densely on the compression flange and are firmly connected with it to prevent the lateral displacement of the bending structural member's compression flange;

b. For the bending structural member of the simple support on both ends of the i-shaped cross section, when the ratio OF the free length l_f TO the width b of its compression flange does not exceed the numerical value specified in Table 27, for the bending structural member with no lateral support points in the span, l_f refers to its span; for the bending structural member with lateral support points in the span, l_f refers to the distance between the lateral support points of the compression flange (the bearing of the bending structural member is considered to have lateral supports);

$$F_h = \frac{A_c \sigma}{85}\sqrt{\frac{\sigma_s}{235}} \qquad (39)$$

Where:

A_c - Cross-sectional area of the bending structural member's compression flange, mm^2;

σ - Calculated stress of the bending structural member's compression flange, Mpa;

F_h - Lateral force generated by the compression flange of the bending structural member, N.

5.6.2.2.3 When the bending structural member does not comply with the Article 5.6.2.2.1, the lateral buckling stability must be checked according to the Equation (32), among which φ_W value can be selected or calculated according to Annex A (supplement).

5.6.3 Local stability of the cylindrical shells

For the thin-walled cylindrical shell subjected to axial compression or bending, when the ratio δ / R OF the shell wall thickness δ TO the radius of the shell's middle surface R is not greater than $25\frac{\sigma_s}{E}$, its local stability must be calculated.

a. The critical stress when the cylindrical shell is subjected to axial compression or bending is as follows:

$$\sigma_{c \cdot cr} = 0.2\frac{E\delta}{R} \qquad (40)$$

Where:

$\sigma_{c,cr}$ - Critical stress when the cylindrical shell is subjected to axial compression or bending, when the critical stress calculated according to the Equation (40) exceeds $0.75\sigma_s$, it is allowed to reduce according to the Equation (41), Mpa;

R - Radius of the middle surface of the cylindrical shell, mm;

δ - Wall thickness of the cylindrical shell, mm;

E - Elastic modulus of the material, Mpa.

$$\sigma'_{c \cdot cr} = \sigma_s\left(1 - \frac{\sigma_s}{5.3\sigma_{c \cdot cr}}\right) \qquad (41)$$

b. The local stability of the thin-walled cylindrical shell subjected to axial compression or bending can be checked according to the Equation (42).

$$\frac{F_N}{A_p}+\frac{M}{W_p}\leqslant\frac{\sigma_{c \cdot cr}}{K_n} \tag{42}$$

Where:

F_N - Calculated axial pressure, SEE Annex G (reference), N;

M - Bending moment, N • mm;

A_p - Net cross-sectional area of the cylindrical shell, mm^2;

W_p - Bending modulus of the cylindrical shell's net cross section, mm^3.

c. Stiffening ring

Both ends of the cylindrical shell shall be equipped with the stiffening rings or the structural members with corresponding effects. When the shell length is greater than 10R, it is necessary to set middle stiffening rings. The spacing of the stiffening rings shall not be greater than 10R. The cross-sectional moment of inertia I_{az} of the stiffening ring shall meet the requirements of the Equation (43).

$$L_{az}\geqslant\frac{R\delta^3}{2}\sqrt{\frac{R}{\delta}} \tag{43}$$

Where:

I_{az} - Cross-sectional moment of inertia of the cylindrical shell's stiffening ring, mm^4.

5.7 Calculation of the structural fatigue strength

The calculation of the structural fatigue strength uses the maximum stress method. CHECK according to the Equations (44), (45) and (46).

$$|\sigma_{max}|\leqslant[\sigma_r] \tag{44}$$

$$|\tau_{max}|\leqslant[\tau_r] \tag{45}$$

$$\left(\frac{\sigma_{xmax}}{[\sigma_{rx}]}\right)^2+\left(\frac{\sigma_{ymax}}{[\sigma_{ry}]}\right)^2-\frac{\sigma_{xmax}\sigma_{ymax}}{[\sigma_{rx}][\sigma_{ry}]}+\left(\frac{\tau_{xymax}}{[\tau_{rxy}]}\right)^2\leqslant 1.1 \tag{46}$$

Where:

σ_{max} - Maximum calculated normal stress, SEE Article 5.7.1, MPa;

τ_{max} - Maximum calculated shear stress, SEE Article 5.7.1, MPa;

σ_{xmax} - σ_{max} whose direction is parallel to the x axis, MPa;

σ_{ymax} - σ_{max} whose direction is parallel to the y axis, MPa;

τ_{xymax} - τ_{max} whose plane is perpendicular to the x axis, and whose direction is parallel to the y axis, MPa;

〔σ_r〕 - Absolute value of the allowable tensile (or compressive) fatigue stress, SEE Article 5.7.3, MPa;

〔τ_r〕 - Absolute value of the allowable shear fatigue stress, SEE Article 5.7.3, MPa;

〔σ_{rx}〕, 〔σ_{ry}〕 - 〔σ_r〕 corresponding to σ_{xmax} and σ_{ymax}, MPa;

〔τ_{rxy}〕 - 〔τ_r〕 corresponding to τ_{xymax}, MPa.

The calculation of the structural fatigue strength can also use the stress amplitude method. SEE Annex J (reference).

The cyclic normal stress (tensile or compressive stress) in the positions to be checked, regardless of x direction or y direction, shall meet the Equation (44). All kinds of cyclic shear stress shall meet the Equation (45). The Equation (46) is only used for checking the compound effect of the above three kinds of (or two of them) cyclic stress.

When there is only one kind of cyclic stress in the positions to be checked, or the effect of one kind of cyclic stress on the structural members is significantly greater than other cyclic stresses, the effects of other cyclic stresses can be ignored, and the fatigue checking is only conducted to this kind of cyclic stress. USE the Equation (44) or (45). At this time, for the structural members whose total number of stress cycles is less than 1.6×10^4, the fatigue checking may not be performed.

5.7.1 Calculated stress

The maximum calculated stress σ_{max} or τ_{max} used during calculation refers to the maximum stress of the absolute value on the fatigue calculation point determined by the load combination A specified by Article 5.3.2. During the fatigue calculation, the compressive stress is negative. For the cyclic shear stress with a progressive direction, positive and negative directions must be determined. During calculation, the shear stress corresponding to the negative direction shall take a negative value.

5.7.2 Cycle characteristics of the stresses

Use the following ratio to express the cycle characteristics of the fatigue stress:

$$\chi = \frac{\sigma_{min}}{\sigma_{max}} \quad (47)$$

$$\chi = \frac{\tau_{min}}{\tau_{max}} \quad (48)$$

σ_{min} (or τ_{min}) here represents another limit value corresponding to σ_{max} (or τ_{max}) in a most unfavorable cycle. σ_{min} (or τ_{min}) and σ_{max} (or τ_{max}) are the same as the amount that can be negative (SEE Article 5.7.1). Therefore, the ratio χ can vary from -1 to +1.

When it is necessary to use the Equation (46) for checking, the ratios χ corresponding to three different cyclic stresses are respectively as follows:

$$\chi_x = \frac{\sigma_{xmin}}{\sigma_{xmax}} \quad (49)$$

$$\chi_y = \frac{\sigma_{ymin}}{\sigma_{ymax}} \quad (50)$$

$$\chi_{xy} = \frac{\tau_{xymin}}{\tau_{xymax}} \quad (51)$$

Where: The numerical values of the maximum calculated stress σ_{xmax}, σ_{ymax} and τ_{xymax} are independently determined according to the provisions of Article 5.7.1, respectively. The numerical values of corresponding σ_{xmin}, σ_{ymin} and τ_{xymin} are independently determined according to the provisions of this Article, respectively.

5.7.3 Allowable fatigue stress

The numerical values of the allowable fatigue stress depend on the working levels of the structure, material categories of the structural members, connection type of the connectors (stress concentration level), cycle characteristics of the stresses (SEE Article 5.7.2), etc.

When considering the stress concentration, the non-welding parts of the structural members are divided into three levels of W_0, W_1 and W_2. The welding parts are divided into five levels of K_0, K_1, K_2, K_3 and K_4. SEE Annex B (supplement).

The allowable tensile or compressive fatigue stress of the welds is the same as the base metal.

d. $[\tau_r]$ of the welds

The allowable shear fatigue stress of the welds shall be determined by the Equation (53).

$$[\tau_r] = \frac{[\tau_{rt}]}{\sqrt{2}} \tag{53}$$

$[\tau_{rt}]$ shall be calculated by the $[\sigma_{-1}]$ value corresponding to K_0.

e. Bolts

For the bolts in single shear, the allowable shear fatigue stress $[\tau_r]$ is taken as 0.6 times the allowable tensile fatigue stress $[\sigma_{rt}]$ corresponding to W_2. For the bolts in double or multiple shear, the allowable fatigue stress is taken as 0.8 times the allowable tensile fatigue stress corresponding to W_2.

For the bolts, the allowable compressive fatigue stress can be taken as 2.5 times the allowable shear fatigue stress.

5.8 Rigidity requirements

The rigidity requirements of the tower cranes are divided into static and dynamic rigidity. The static rigidity is characterized by the static displacement value of the structure where the elastic deformation is generated when the specified load is applied to the specified position. The dynamic rigidity of the tower crane as the vibration system is characterized by the lowest natural frequency (referred to as the full load natural frequency) of the system when the length of the suspension of the wire rope winding is equal to the rated hoisting height.

5.8.1 Static rigidity

The horizontal static displacement Δx of the crane body at the junction of the boom shall not be greater than h / 100 under the rated hoisting load of the tower crane, where h refers to the vertical distance FROM the junction of the crane body and the boom TO the plane of action directly supporting the entire crane body for the mobile tower cranes, and refers to the vertical distance FROM the junction of the crane body and the boom TO the highest attachment point for the attached tower cranes. CALCULATE according to the Equation (54).

$$\Delta_x = \frac{1}{1 - \frac{F_N}{F_E}} \Delta_M \tag{54}$$

Where:

F_N - Resultant force of all the vertical forces above the junction of the crane body and the boom [including the conversion force of the crane body's deadweight here, SEE Annex G (reference) for the conversion method], under the rated hoisting load, N;

F_E - Euler critical loads F_{Ex} and F_{Ey}, whichever is smaller, F_{Ex} and F_{Ey} shall be calculated according to the Equation (55), N;

Δ_M - Horizontal displacement at the junction of the crane body and the boom caused by the bending moment M on the centerline of the crane body under rated hoisting load, mm.

$$\begin{cases} F_{Ex} = \dfrac{\pi^2 EI_{ax}}{(\mu_1 \mu_2 h)^2} \\ F_{Ex} = \dfrac{\pi^2 EI_{ay}}{(\mu_1 \cdot \mu_2 h)^2} \end{cases} \tag{55}$$

Where:

μ_1, μ_2 - SEE Annex H (reference).

5.8.2 Dynamic rigidity

Do NOT check the dynamic rigidity of the tower cranes in general. When the user or the design itself has a special request, checking can be performed.

5.9 Construction requirements

5.9.1 General principle

a. The structural rod's center of gravity line shall coincide with the centerline of the structural geometry as far as possible. The center of gravity line of the attachment weld and the bolt set shall coincide with the rod's center of gravity line as far as possible.

b. The structural design of the major bearing structure shall be as simple as possible, and the force shall be clear, so as to reduce the stress concentration.

c. The structures (or structural members) shall be constructed so that they can be easily made, transported, staggered, installed and maintained. The structure of the outdoor operation must be protected from water.

d. For the welded structural members with direct access to dynamic load and complex stress, unless taking measures to reduce or eliminate the internal welding stress, otherwise the thicknesses of the carbon steel and low alloy steel in use shall not be greater than 50mm and 36mm, respectively.

e. It is permissible for the major bearing structural members to use different connection methods at different junctions, but the mixture of different connection methods are not allowed at the same junction.

f. For the large structures requiring splicing on site, it is advisable to give priority to the high-strength bolted connections.

g. The joint of the welded beam's transverse stiffening rib and the flange plate shall be chamfered, with a width of about *b* / 3 (but not greater than 40mm) and a height of about *b* / 2 (but not greater than 60mm). SEE Figure 6. *b* refers to the width of the stiffening rib.

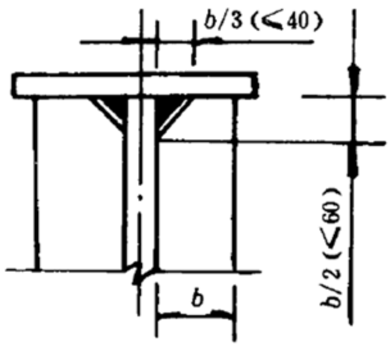


Figure 6

h. For the bearing structure, the thickness of the steel plates and open profiles shall not be less than 4mm, while the thickness of the closed profiles shall not be less than 3mm.

i. The thickness of the truss gusset plate is generally determined according to the magnitude of the internal force of the connecting rod, but not less than 5mm.

5.9.2 Welded connection

5.9.2.1 Butt welding

The groove type of the butt welding shall conform to the provisions of GB 985 and GB 986.

For the butt welding with unequal plate thickness or width in the major bearing structure, when the difference between the plate thicknesses or widths of the

6.2.1.1 Motor primaries

The static power of the mechanism is calculated by the hoisting load, rated hoisting speed and mechanism efficiency. The motor primaries is performed according to the static power of the mechanism and the electrical continuity. The electrical continuity of the mechanism is shown in Annex M (reference).

In the absence of special requirements, the motor capacity and the mechanism control system shall be such that the average acceleration of the item at the time of commencement of the mechanism is not greater than $0.8 m/s^2$.

6.2.1.2 Motor capacity checking

The motor overload and heating must be verified. The verification method is described in Annex N (reference) and Annex O (reference).

6.2.1.3 Brake selection

The driving device of the hoisting mechanism shall be provided with at least one supporting brake which shall be normally closed. The brake wheel must be mounted on the rigid coupling shaft of the transmission mechanism.

The safety factor of braking shall not be less than 1.5.

In the absence of special requirements, the item lifting and lowering deceleration caused by braking shall not be greater than $0.8 m/s^2$.

It is recommended to use the support and control braking together. The control braking can be electrical, such as regenerative braking, reverse braking, energy consumption braking, eddy current brake, etc. It can also be mechanical. The control braking is only used to consume kinetic energy for safe deceleration of the items. When used in conjunction with the control braking, the minimum brake safety factor of the support brake should still meet the above requirements.

6.2.2 Luffing mechanism

6.2.2.1 Luffing jib mechanism

6.2.2.1.1 Equivalent luffing resistance

The equivalent luffing resistance torque refers to the root-mean-square value calculated from the corresponding hoisting capacity in terms of the luffing resistance torque at each of the different amplitude positions and the luffing time of the corresponding luffing interval in overall luffing process under normal operating condition. The luffing resistance torque is generated by the unbalanced hoisting load and the deadweight load of the boom system, the wind force acting on the boom system, the horizontal inertial force and the centrifugal force of the hoisting item, the inertia force of the boom system, the resistance caused by the slope, and the frictional resistance of the boom system during luffing.

6.2.2.1.2 Motor primaries

For the motor with the luffing jib mechanism, the equivalent power is calculated according to the maximum value of the root-mean-square value's equivalent resistance torque of various operating conditions under normal operating condition, and the motor primaries is performed according to the equivalent power and the electrical continuity of the mechanism. The electrical continuity of the mechanism is shown in Annex M (reference).

The motor capacity and mechanism control system shall ensure that the maximum acceleration (deceleration) of the horizontal movement of the boom head during luffing shall not be greater than $0.6m/s^2$.

6.2.2.1.3 Motor capacity checking

The motor overload and heating must be verified. The verification method is described in Annex N (reference) and Annex O (reference).

6.2.2.1.4 Brake selection

Normally closed brake shall be used. For the balanced luffing mechanism, the braking safety factor is respectively taken as 1.25 and 1.15 under the operating and non-operating conditions. For the unbalanced luffing mechanism, a support brake and a stopper or two support brakes shall be mounted. The principle of selecting the brake safety factor of the support brake is the same as that of the lifting mechanism. When two support brakes are mounted, each brake safety factor shall not be less than 1.25. Furthermore, the stopper shall be capable of safely supporting the boom and hoisting load when supporting the brake failure. An interlocking protection device shall be provided between the stopper and the luffing jib mechanism so that the luffing mechanism cannot be actuated before the supporting of the stopper is removed.

The brake deceleration shall not exceed the numerical values in Article 6.2.2.1.2.

6.2.2.2 Luffing mechanism of the trolleys

6.2.2.2.1 Luffing static resistance

The luffing static resistance of the luffing mechanism of wire rope traction trolley includes the frictional resistance, equivalent ramp resistance (calculated according to 1% slope), and resistance caused by hoisting wire rope and traction wire rope bypassing the guide pulley. The frictional resistance includes the resistance of the wheel rolling along the rail, the frictional resistance within the wheel bearing, and the additional frictional resistance between the wheel rim and the side of the rail. The latter is generally considered by multiplying the sum of the two basic frictional resistances above by the additional coefficient 1.5.

6.2.2.2.2 Maximum traction of the stable luffing

The maximum traction of the stable luffing includes the frictional resistance (as specified in Article 6.2.2.2.1), maximum wind resistance under normal operating condition (calculated by the wind pressure p_{W2}), maximum ramp resistance during full load operation (calculated according to 2% slope), resistance caused by hoisting wire rope and traction wire rope bypassing the guide pulley, and traction required by the traction wire rope to maintain a certain degree of sag.

The traction wire rope of the luffing mechanism of wire rope traction trolley uses the method in Article 6.4.4.2.2 to select the wire rope according to the maximum traction of the stable luffing, and performs checking according to the maximum torque of the motor.

6.2.2.2.3 Motor primaries

The static power of the mechanism is calculated according to the luffing static resistance, luffing speed and mechanism efficiency. The motor primaries is performed according to the static power and electrical continuity of the mechanism. The electrical continuity of the mechanism is shown in Annex M (reference).

The motor capacity and mechanism control system shall ensure that the acceleration (deceleration) of the luffing trolley shall not be greater than $0.5 m/s^2$.

6.2.2.2.4 Motor capacity checking

The motor overload and heating must be verified. The verification method is described in Annex N (reference) and Annex O (reference).

6.2.2.2.5 Brake selection

The brake torque of brake for the luffing mechanism of the wire rope traction trolley plus the running frictional resistance torque (excluding the frictional resistance torque of the rim and the side of the rail) shall be able to stop the luffing trolley in an unfavorable situation within the required time (the time required is determined by the operating conditions of the crane). For the luffing mechanism of the overhead unbalanced trolley, the brake safety factor shall not are used, it is advisable to slow down before braking. The brake deceleration shall not exceed the numerical value in Article 6.2.3.2. In the meantime, it is also necessary to ensure that the slewing parts are free to slew under the wind force at a wind speed of greater than 20m/s.

6.2.3.5 Limit torque coupling

Each self-locking slewing transmission mechanism shall be equipped with a limit torque coupling. The limit torque of the limit torque coupling shall ensure that the slewing mechanism does not slip in the process of normal startup or braking and only starts to slip in case of overload. The limit torque value is generally taken as 1.1 times the torque value transmitted to the coupling shaft after subtracting the inertia torque of the parts on the motor shaft from the maximum torque of the motor.

6.2.4 Running mechanism

6.2.4.1 Running static resistance

The running static resistance includes the frictional resistance, ramp resistance (the slope is taken as 0.5%) and cable towing resistance. For the regular travelling tower cranes, the effects of the equivalent wind resistance shall also be taken into account. The frictional resistance includes the resistance of the wheel rolling along the rail, frictional resistance within the wheel bearing, and additional frictional resistance between the wheel rim and the side of the rail head. The latter is generally considered by multiplying the sum of the two basic frictional resistances above by the additional coefficient 1.5.

6.2.4.2 Motor primaries

The static power of the mechanism is calculated according to the running static resistance, running speed and mechanism efficiency. The motor primaries is performed according to the static power and electrical continuity of the mechanism. The electrical continuity of the mechanism is shown in Annex M (reference).

In the absence of special requirements, the motor capacity and the mechanism control system shall ensure that the average linear acceleration (deceleration) value of the tower crane is not greater than 0.05m/s^2 to 0.07m/s^2.

6.2.4.3 Motor capacity checking

The motor overload and heating must be verified. The verification method is described in Annex N (reference) and Annex O (reference).

6.2.4.4 Brake selection

The brake torque of the running mechanism's brake plus the running frictional resistance (excluding the frictional resistance of the rim and the side of the rail head) shall be able to stop the tower crane at full load and under the unfavorable conditions of downwind and downhill within the required time (which can be calculated according to the deceleration specified in Article 6.2.4.2).

Normally closed brakes shall be used. It is advisable to slow down before braking.

6.2.4.5 Slip checking

During the startup or braking of the running mechanism, the drive wheel should not slip normally. During checking, the adhesion coefficient of the tower crane wheels and the rail is taken as 0.12.

For the running mechanism using the hydraulic coupling with limited torque, the slip checking may not be performed during startup.

6.2.5 Jacking mechanism

The self-elevating tower cranes shall give priority to the hydraulic jacking mechanism. The mechanical jacking mechanism can also be adopted.

6.2.5.1 Jacking load

The jacking force is calculated from the total gravity of the jacking components and the frictional resistance between the relative moving parts in the jacking process. During jacking, the calculated wind pressure is 100Pa as specified.

According to the approximate calculation, the frictional resistance can be taken as 0.1 to 0.2 times the total gravity of the jacking components.

6.2.5.2 Jacking speed

The jacking speed is suitable to be taken as 0.005m/s to 0.013m/s (0.3m/min to 0.8m/min). For large tower cranes, TAKE a smaller value.

The lowering speed shall not be greater than the jacking speed.

6.2.5.3 Hydraulic jacking mechanism

a. Fuel tank capacity

The fuel tank capacity of the hydraulic system can be determined according to the Equation (58).

The maximum operating load T_{2max} is used for static strength calculation of the parts, which can be determined according to the following methods:

a. Hoisting mechanism

When hoisting items from the ground, for the part outside the shaft connected with the motor, TAKE the load acting on the part, which is generated by the product of the hoisting load and the dynamic load factor ϕ_2; for the part on the shaft connected with the motor, TAKE 2.0 to 2.5 times the motor's rated torque acting on this part. During the ascending startup and descending braking of the suspended items, the load of the parts can be calculated according to the Article E2 in Annex E (reference).

b. Other mechanisms

For the running, slewing and luffing mechanisms, the maximum operating load T_{2max} can be calculated according to the Article E2 in Annex E (reference).

6.3.2.4 Special loads

The special loads are used for checking the static strength of some parts.

a. Buffer collision load

The dynamic load caused to the running mechanism during buffer collision shall be estimated according to the Equation (62).

$$T_{3max} = 0.25 \frac{R}{i} \sum F_{vmax} \tag{62}$$

Where:

T_{3max} - Torque acting on the drive shaft of the running mechanism during buffer collision, N • m;

R - Wheel radius, m;

i - Total transmission ratio of the mechanism;

$\sum F_{vmax}$ - The sum of the maximum calculated wheel pressures on the drive wheels of the transmission mechanism, N.

b. Test load

According to the provisions of Article 4.2.3.2, the greater one of the product of the dynamic test load and the dynamic load factor ϕ_6, and the static test load, shall be taken as the special load of the computing mechanism.

6.3.2.5 When a mechanism is applied to the self-installation operation of the tower crane, the load caused by this operation to this mechanism shall be checked.

6.3.3 Number of stress cycles

When calculating the fatigue strength of the transmission parts, the number of stress cycles within the design life shall be calculated. When the stress variation is smaller than 10% of the maximum stress of the absolute value, their number of stress cycles shall not be calculated.

Total number of stress cycles N_s is:

$$N_s = N_h t_z \tag{63}$$

Where:

N_h - Number of stress cycles per hour;

t_z - Total operating time of the parts, selected according to Table 30, h.

The parts of the mechanism shall generally be calculated according to the design life of the mechanism. In some cases, the design life of certain parts is lower than the specified value of this mechanism's utilization level for economic reasons or subject to technical limitations.

Number of stress cycles per hour N_h shall be approximately calculated according to the following two cases.

a. The transmission part whose number of stress cycles is only related to the number of operating stresses:

$$N_h = N_a N_p \tag{64}$$

Where:

N_p - Number of operating stresses per operating hour;

N_a - Number of stress cycles that the part experiences in each operating cycle.

b. The transmission part whose number of stress cycles is related to the slewing speed:

$$N_h = \frac{60 n_m \cdot N_l}{i_m} \tag{65}$$

Where:

n_m - Motor's slewing speed, r/min;

i_m - Transmission ratio from the motor to the calculated part;

N_b - Number of stress cycles that each part undergoes.

6.3.4 Calculated stress

The load for the calculated stress on the hazardous point of the calculated parts shall comply with the provisions of Article 6.3.2. USE usual method of material mechanics for calculation. The composite stress shall be synthesized according to the appropriate strength theory.

6.3.5 Strength limit

During static strength calculation, for the materials with better plasticity, the yield point σ_s of the material is used as the strength limit of the parts.

When the ratio OF the yield point σ_s TO the tensile strength σ_b of the material is greater than 0.7, in order to reduce the risk of brittle fracture caused by accidentally exceeding the yield point of the material, the hypothetical yield point calculated according to the Equations (66) and (67) as specified is used as the strength limit of the parts.

$$\sigma_{SF} = \frac{\sigma_s + 0.7\sigma_b}{2} \qquad (66)$$

$$\tau_{SF} = \frac{\sigma_{SF}}{\sqrt{3}} \qquad (67)$$

6.3.6 Fatigue strength limit

The fatigue strength limit σ_{rk} of the parts is obtained by the test or calculation. 90% of the parts shall remain effective under this stress.

The magnitude of the part's fatigue strength limit depends on:

a. Stress cycle characteristics χ ($\chi = \frac{\sigma_{min}}{\sigma_{max}}$);

b. Mass of the material;

c. Shape of the part;

d. Dimensions of the part;

e. Surface status of the part.

l_s - Fulcrum spacing of the shaft, mm.

6.3.9 Abrasion checking

For parts that are in frequent friction during motion, the abrasion loss of the friction surface during the use period shall be within the allowable range. Relevant physical quantities affecting the abrasion loss are usually checked to ensure that they do not exceed the allowable value. For instance, for the brakes, clutches and sliding supports, the pressure intensity per unit area *p* of the coverage and the characteristic coefficient *pv* (the product of *p* and relative motion velocity *v* of the friction surface) shall be checked to ensure that they do not exceed the allowable values. The allowable physical quantities of the commonly used materials of the friction surfaces shall comply with the provisions in Annex R of GB 3811 [SEE Annex R (reference) of this Standard]. When the hydraulic push rod brakes or other brakes are used as the speed control device, the friction surfaces shall use the abrasion and high temperature resistant materials. The brake wheels shall have good heat dissipation conditions, and shall also be conducted with the heat dissipation checking, so as to ensure that their temperature rise is within the allowable range.

6.4 Parts

6.4.1 Hook

The design calculation and selection of the hooks shall meet the provisions of GB 1005.1.

Each hook shall be equipped with an anti-off pawl. The design of the anti-off pawls shall comply with the provisions of JJ 75.

6.4.2 Wire rope

6.4.2.1 Selection of the wire rope's structural type

The hoisting wire ropes shall be preferred to use the non-rotating wire ropes.

In the environments with larger corrosion, the galvanized wire ropes shall be used.

6.4.2.2 Calculation and selection of the wire rope's diameter

The calculation and selection of the wire rope's diameter shall be performed according to the following two methods. During design, SELECT any one of the methods according to the specific circumstances.

a. The diameter of the wire rope shall be selected according to the maximum operating static tension of the wire rope and the safety factor related to the working level of the mechanism belonging to the wire rope, which is suitable for the running wire ropes and the wire ropes for tensioning. The breaking force of the selected wire rope shall meet the Equation (71).

$$F_{ro} \geq F_{rmax} K_{nr} \tag{71}$$

Where:

F_{ro} - Breaking force of the selected wire rope, N;

F_{max} - Maximum operating static tension of the wire rope, N;

K_{nr} - Minimum safety factor of the wire rope, selected according to Table 34.

Table 34 K_{nr} Values

Working levels of the mechanism	M_1	M_2	M_3	M_4	M_5	M_6
Safety factor K_{nr}	3.5	4	4.5	5	5.5	6

Note: The safety factor of the wire ropes for tensioning shall not be less than 3.5.

b. The diameter of the wire rope can be determined by the maximum operating static tension of the wire rope according to the Equation (72), which is suitable for the running wire ropes.

$$d_{min} = C\sqrt{F_{max}} \tag{72}$$

Where:

d_{min} - Minimum diameter of the wire rope, mm;

C - Selection factor of the wire rope, mm / $\sqrt{N}$.

The values of the selection factor C are taken according to the working levels of the mechanism, which shall be selected according to Table 35. The values of the selection factor C of the wire rope with a fiber core and the structural types of 6 × 19 and 6 × 37 are listed in the table. When the structural type and the strength limit of the wire rope are different, the calculation of the selection factor C value is shown in Annex S (reference).

c. Under any circumstances, the actual diameter of the stressed wire rope shall not be less than 6mm.

6.4.2.3 Allowable deflection angle of the wire rope

F_{vmin} - Minimum wheel pressure generated by the basic load and additional load during normal operation, N.

When determining F_{vmax} and F_{vmin}, the coefficients ϕ_1 to ϕ_7 in the Article 4.2 of this Standard shall all be taken as 1.

6.4.4.2 Fatigue calculation of the wheel tread

The contact fatigue strength shall be calculated according to the Hertz equation. It can be divided into line contact and point contact according to different contact situations of the wheels and the rail.

a. Line contact, calculated according to the Equation (75).

$$F_v \leq k_1 D_w l_t C_1 C_2 \tag{75}$$

k_1 - Allowable line contact stress constant associated with the material, k_1 of the steel wheels shall be selected according to Table 37, MPa;

D_w - Wheel diameter, mm;

l_t - Effective contact length of the wheels and the rail, mm;

C_1 - Rotating speed factor, selected according to Table 38;

C_2 - Working level factor, selected according to Table 39.

b. Point contact, calculated according to the Equation (76).

$$F_v \leq k_2 \frac{R_W^2}{k_r^3} C_1 C_2 \tag{76}$$

Where:

k_2 - Allowable point contact stress constant associated with the material, k_2 of the steel wheels shall be selected according to Table 37;

R_W - Curvature radius, the greater value of the curvature radius of the wheel and the top radius of the rail;

k_r - The coefficient determined by the ratio OF the curvature radius of the top of rail TO the curvature radius of the wheel ($\frac{r_W}{R_W}$), selected according to Table 40.

6.4.5.4 Mounting bolts of the slewing bearing

a. The mounting bolts and nuts shall select high strength bolts and nuts above Level 8.8.

b. The mounting bolts shall be conducted with the static strength calculation and fatigue strength calculation. SEE Annex U (reference) for the calculation method.

c. When the upper and lower supports connected with the slewing bearing use the threaded hole installation, the minimum screw-in depth of the bolts on the upper and lower supports shall not be less than the specified value. SEE Table U5 in Annex U (reference).

6.4.6 Reducer selection

Selection of standard model reducers shall check the input power and output torque according to the working level and load of the mechanism as well as the allowable bearing capacity of the reducer at the corresponding working level, so as to ensure the design life and intensity in use.

The maximum radial load on the shaft end of the reducer shall also be checked if necessary.

6.4.6.1 Reducer checking of the hoisting mechanism

The maximum static power of the hoisting mechanism shall be less than or equal to the allowable input power of the reducer at a given working level.

The maximum output torque of the reducer shall be applied to the reel according to the maximum operating load, and shall also consider the effects of the pulley efficiency on the calculation. The calculated value shall be less than the maximum allowable output torque of the selected reducer at the given working level.

6.4.6.2 Reducer checking of the slewing and running mechanisms

The motor directly drives the reducer. The maximum output torque of the reducer shall be calculated according to the maximum torque of the motor, and shall not exceed the maximum allowable output torque of the reducer at the given working level.

The motor drives the reducer by means of a hydraulic coupling, an electromagnetic coupling, a belt drive or the like. If the transmission torque of these devices is greater than the maximum torque of the motor, the maximum output torque of the reducer shall be checked according to the maximum torque of the motor. On the contrary, the checking shall be performed according to the transmission torque of these devices.

6.4.6.3 Reducer checking of the luffing mechanism

a. When the primary reducer is selected for the trolley luffing mechanism, the output torque of the reducer shall be calculated according to the luffing static resistance torque in Article 6.2.2.2.1.

Then, the maximum output torque of the reducer shall be calculated according to the maximum output torque of the motor. This torque shall not be greater than the maximum allowable output torque of the selected reducer at the given working level.

b. The selection method and requirements of the luffing jib mechanism's reducer are the same as the hoisting mechanism.

6.4.7 Coupling selection

6.4.7.1 Mechanical coupling selection

The mechanical couplings can generally be selected from the couplings in standard specifications. Then, CHECK the transmitted torque to ensure that it meets the Equation (78).

$$T_c \leq T_t \tag{78}$$

Where:

T_t - Torque given in the coupling specification table, N • m;

T_c - Calculated torque of the coupling, calculated according to the Equation (79), N • m.

$$T_c = K_{nt} T_{1max} \tag{79}$$

Where:

K_{nt} - Safety factor of the coupling, for the hoisting and unbalanced luffing mechanisms, K_{nt} = 1.5; for other mechanisms, K_{nt} = 1.35;

T_{1max} - Calculated basic load of the coupling, calculated according to the Article E1 in Annex E (reference), N • m.

6.4.7.2 Hydraulic coupling selection

The torque-limited (yox) and general (yop) hydraulic couplings are normally adopted.

For the running device with the rated running speed of less than 0.7m/s, it is allowed not to check the capacity of its buffering device absorbing the kinetic energy.

The buffer housing and stop as well as the fixture shall be designed according to the maximum impact force that occurs when the tower crane collides at the rated speed. K_{n3} in Table 33 shall be taken as the strength safety factor.

7 Electric

7.1 Control system and maneuvering gear

7.1.1 The control system of each mechanism shall meet the requirements for the operating speed of each mechanism.

7.1.2 The control system of the hoisting mechanism shall be designed based on the following principles:

a. When the load is up to 125% of the rated load, the items can still be hoisted from the ground.

b. When the voltage value is 0.9 times the rated voltage, lift up the maximum hoisting capacity, the items are not allowed to slip regardless of the control handle in any position.

c. In the descending state, regardless of the control handle in any position, unless the design has allowed, the descending speed shall not exceed 120% of the rated descent speed.

7.1.3 The control system shall ensure that the mechanism is activated and the braking acceleration is required.

7.1.4 The hoisting mechanism with the mechanical or electromagnetic shift, its control circuit shall take protective measures to prevent the driver misuse.

7.1.5 The maneuvering gear is preferentially used with a linkage console. The console shall be arranged in accordance with the provisions of GB 5144 and shall be provided with an emergency stop button capable of stopping all motions.

7.1.6 For portable maneuvering gears, the control circuit voltage shall not exceed 48V, and must be equipped with an emergency button capable of cutting off the total power supply. This device should have good insulation properties and the protection grade of not lower than IP 44.

7.2 Power supply

7.2.1 The tower cranes use 380V, 50Hz three-phase AC power supply. The three-phase AC power supply with other parameters can also be used according to the user requirements.

7.2.2 The power supply shall ensure that the tower crane's voltage variation of does not exceed ±10% of its rated value during normal operation.

7.2.3 The power supply shall set up the circuit switch that can easily turn on and cut off the power supply of the complete machine.

7.2.4 Control voltage shall be selected from the following levels:

AC: 380V, 220V, 127V, 48V, 36V, 24V;

DC: 220V, 110V, 48V, 24V, 12V.

7.2.5 For the running tower cranes, the power cords shall be introduced by a cable roll that can be rolled on its own. The radius of the inner ring of the cable reel shall be greater than the allowable bending radius of the power cable. Under normal circumstances, when the outer diameter of the cable is less than 21.5mm, the diameter of the reel's inner ring shall be at least 10 times the outer diameter of the cable. When the outer diameter of the cable is greater than 21.5mm, the diameter of the reel's inner ring shall be at least 12.5 times the outer diameter of the cable. The collecting ring of the cable reel shall meet the requirements of the corresponding voltage level and current capacity. Each slip ring shall be at least equipped with a pair of carbon brush. The contact voltage drop of the carbon brush shall be less than 1.5V. The current density is generally taken as $15A/cm^2$ to $20A/cm^2$. The protection grade of the collecting ring shall not be lower than IP 44.

7.2.6 The maximum tensile stress that the cable copper core is subjected to shall not exceed 20MPa.

7.3 Motor selection

7.3.1 Basic requirements of the motors

The motor selected by each mechanism must comply with GB 755 and the standards related to the special motors.

The working system levels of the motor shall conform to the corresponding requirements of the mechanism.

The protection grade of the motor enclosure shall not be lower than IP 23. Each motor whose protection grade is lower than IP 44 must be equipped with a protective cover, without affecting the heat dissipation of the motor.

7.4.2.3 The resistors for the tower cranes are repetitive short-term working system, with a cycle period of 60s. The electrical continuities are divided into 4.4%, 6.2%, 8.8%, 12.5%, 17.5%, 25%, 35%, 50%, 70% and 100%. The capacity of the resistance at all levels shall be selected according to different access time.

7.4.2.4 The same resistance element has different allowable current values at different electrical continuities. This value shall be not less than the operating current of the motor. In order to reduce the number of resistors, the allowable current value of the selected resistance elements at individual levels can be 5% less than the motor operating current.

7.4.2.5 The protection grade of the resistance box shall not be lower than IP 23. The lead wire shall be firmly connected to withstand vibration without becoming loose.

7.4.3 Contactor

The contactor shall be selected according to its operating current and expected service life based on the selection curve provided by the manufacturing plant. For the frequently used contactors along with more complex operating conditions, it is advisable to select the capacity at a higher level for application.

7.4.4 Fuse

7.4.4.1 SELECT the fuses with the corresponding voltage levels based on the circuit voltage. It is advisable to select the switch or spiral fuses.

7.4.4.2 SELECT the melt rated current based on the rated current of the circuit and the required operate time of protection according to the protection characteristics curve provided by the manufacturing plant.

As for the fuses for preventing the motors from short circuit, the melt rated current can respectively be calculated according to the Equations (82) and (83).

a. For protecting single motor

$$I_R = k_i I_e \qquad (82)$$

Where:

I_R - Melt rated current, A;

I_e - Rated current of the motor, A;

k_i - Coefficient, taken as 1.5 to 2.5, and taken as the greater value during heavy-load full-pressure direct startup.

7.5.4 Wire laying

7.5.4.1 The connecting wires in the distribution boxes can be laid in the trunkings or use the rear X-type wiring. Both ends of each wire shall be marked with the serial number consistent with that in the electrical schematic diagram.

7.5.4.2 The external connecting wires can be laid in the trunkings and metal pipes. The cables can be laid directly. Where there is mechanical damage, chemical corrosion and oil erosion, the protective measures shall be taken on the cables.

7.5.4.3 For the fix-laid cable, the bending radius shall not be less than 5 times the outer diameter of the cable. In addition to the cable reel, the bending radius of the mobile cable shall not be less than 8 times the outer diameter of the cable. It is appropriate to use the cable mesh suspension method to fix the cables suspending on the crane body. A cable mesh shall be set every 20m.

7.5.4.4 A halfway junction box shall be set when the cable needs to be lengthened. The junction box shall be secured for easy maintenance. Standard terminal blocks shall be used in the junction box. The terminal capacity shall meet the requirements of the wire's current carrying capacity, and shall have the same serial number as the wire. The protection grade of the junction box is IP 44.

7.5.5 Voltage loss

7.5.5.1 For AC power supply, at peak current, the voltage loss from the self-powered transformer's low-voltage bus to any motor terminal shall not exceed 15% of the rated voltage.

7.5.5.2 The peak current shall be calculated according to the following methods:

a. Peak current calculation of single mechanism:

$$I_p = k_e I_e \tag{84}$$

Where:

I_p - Peak current, A;

k_e - Motor starting current multiple, for the cage motors, TAKE the given value of the sample; for the winding motors, 2 can be taken when there is no exact data;

I_e - Rated current of the motor, A.

7.6.1.2 The arrangement of the components in the distribution box shall ensure that there is sufficient dismantling distance and the safety distance required by the electrical components. In the meantime, the heat, arc, vibration and magnetic field generated by the action of the components shall not affect the normal functions of other components, and shall not cause malfunction.

7.6.1.3 The metal shell of the distribution box must be grounded.

7.6.2 Installation requirements

7.6.2.1 A clearance of at least 400mm must be maintained before the distribution box for easy maintenance.

7.6.2.2 Each distribution box must be equipped with a facility for easy hoisting (rings, etc.).

7.7 Protective devices

7.7.1 Short circuit and overcurrent protection

7.7.1.1 The total power supply circuit must be equipped with an automatic air switch for short circuit protection. For the tower crane with a lifting torque of above 800kN • m, each mechanism shall be provided with a separate automatic air switch for short circuit protection. Each phase of the automatic air switch shall be configured with an instantaneous overcurrent release, and its setting shall be greater than the peak current of the control object.

If some low-power motors use fuses for short circuit protection, the rated current of the melt shall be selected according to Article 7.4.4.2.

7.7.1.2 When an overcurrent relay using instantaneous operation is used for overcurrent protection of a single winding motor, its operating current shall be set according to 2.5 to 3 times the rated current of the motor. When an overcurrent relay using inverse time operation is used for protecting a winding motor, the relay should not operate when the motor starts, and there should be good protection characteristics during overcurrent.

7.7.1.3 The frequently started and reversed cage motors shall not use thermal relays for motor overload protection. It is advisable to select the motors with overheating protection (embedded with thermo-sensitive elements in the motor stator windings).

7.7.2 Under-voltage, overvoltage and voltage loss protection

7.7.2.1 In order to ensure normal operation, the under-voltage and overvoltage alarm devices shall be set. When the voltage is lower than $0.85V_e$

and higher than $1.1V_e$ (V_e refers to the rated supply voltage), the alarm shall be given or the power supply circuit shall be cut off automatically.

7.7.2.2 The power supply circuit must be equipped with voltage loss protection. When the power supply is interrupted, the total circuit can be automatically disconnected.

7.7.3 Zero position protection

Each mechanism shall be equipped with the zero position protection. When starting operation and recovering power supply after voltage loss, the handle of each mechanism's maneuvering gear must be set to zero before the motor can start up.

7.7.4 Phase dislocation and loss protection of the power supply

The power circuit shall be equipped with the device for phase dislocation and loss protection.

7.7.5 Others

It is necessary to set up an emergency switch or device that can quickly disconnect the main power supply in an emergency situation where the driver is easy to operate.

The DC shunt motors shall be equipped with magnetic loss protection.

7.8 Ground connection

7.8.1 The normally uncharged metal shells, metal piping, low-voltage sides of the safety lighting transformers of all electrical equipment are required to be grounded in a reliable way. When the electrical equipment is directly attached to the metal structural member, and there is a reliable electrical contact, it is unnecessary to install another electrical connecting wire.

7.8.2 Three-phase four-wire power supply systems shall be grounded repeatedly, which is to reconnect one or more grounding devices on the part with the ground.

7.8.3 Under normal circumstances, it can be considered that there is reliable electrical connection between the wheel and rail of the running mechanism, along with the ground connection through the rail. If there is no reliable electrical connection between the wheel and the rail due to the non-conductive dust deposition, you should have a special grounding wire or take other measures to make it in good contact.

7.8.4 The minimum cross-section of the copper wire for the grounding of a single electrical device is:

Exposed bare wires, $4mm^2$;

Insulated wires, $1.5mm^2$.

7.8.5 The connection of the grounding wire to the device can be screwed or welded. Screw connection shall take anti-loose and anti-rust measures.

7.8.6 The grounding wires are strictly forbidden to be used as the current-carrying zero lines.

7.9 Lighting, signals and communication

7.9.1 Lighting

7.9.1.1 ENSURE that there is moderate lighting in the cabs. The lighting of the crane body and other parts shall be set according to the requirements of the users.

7.9.1.2 The power supply voltage of the fixed lighting device shall not exceed 220V. It is strictly forbidden to use metal structures as the circuits of the lighting circuits.

7.9.1.3 The power supply voltage of the portable lighting device shall not exceed 48V. AC power supply is strictly prohibited using autotransformer step-down. Primary coils for other step-down transformers shall be controlled by the bipolar switch or automatic air switch.

7.9.2 Signals

7.9.2.1 The cabs shall at least have the following signal indications:

a. Signal indication on the on-off status of the total power supply;

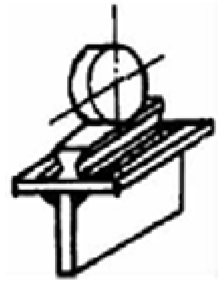
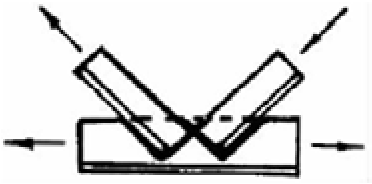
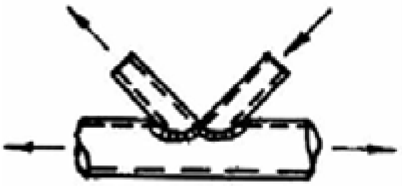
b. Alarm or signal indication when exceeding the hoisting capacity and hoisting torque.

7.9.2.2 The signals can be light, sound or instrumentation indications. The signal device shall be located in the sight of the operators.

7.9.3 Communication

The communication equipment shall be selected according to the requirements of the users. Radio intercom, carrier telephones and amplifiers are optional.

7.10 Others

	Weld between the flange plate and web bearing the concentrated load K-shaped groove weld Double-sided i-shaped groove weld	K_3 K_4
	USE I-shaped slope welds for rod member connection of the truss joint	K_4
	For the truss made of tubes, USE I-shaped slope welds for joint connection	K_4

Note: ① The welded joints applied in Table B1 shall meet the following technical requirements:

a. Before the butt joints are welded, the base metal at the weld root shall be processed and finished. The backing welds shall be applied. The end of the weld shall not protrude or recess the continuous surface of the structural member.

b. For the T-shaped or cross connectors of the K-shaped slope, the clearance of the incomplete penetration in the blunt edge shall not be greater than 0.2 times the height of the welding foot, and shall not exceed 3mm.

c. All welds of the joints shall be conducted with the appearance inspection, and shall comply with the provisions on conforming welds in JJ 12.1.

② Besides meeting the requirements in Note ①, if the joint welds can meet the following technical conditions at the same time, the corresponding stress concentration level (except for Level K_0) can be one level lower than that specified in the table (indicating that K_i can be degraded to K_{i-1}):

a. In the welding area, the transition shall be smooth without defects, and shall be processed if necessary. The direction of the processing traces shall be parallel to the direction of the maximum tensile stress.

b. All welds shall be conducted with the appearance inspection. The butt welds shall also be conducted with the non-destructive testing of internal quality. Furthermore, they shall conform to the provisions on top quality welds in JJ 12.1.

Annex D

(Reference)

Estimation method of the hoisting impact factor ϕ_1

D1 When the items are hoisted from the ground, the steel structure (boom, crane body, etc.) of the tower cranes will generate elastic vibration. Since the structure mass is involved in the vibration, the additional inertial impact's dynamic load is generated in the structural member. The impact acceleration a is caused by hoisting the items from the ground, and its value can establish the following relationship with ϕ_2:

$$a = 0.1\ (\phi_2 - 1)\ g \tag{D1}$$

Where:

ϕ_2 - Hoisting dynamic load factor, determined according to the Article 4.2.1.2;

g - Acceleration of gravity.

SET $k_a = \dfrac{a}{g}$

Then $\phi_1 = 1 \pm k_a$

When designing different structural members of the same tower crane, ϕ_1 value will be different. It should be noted, however, that the load of the structural member to be checked should be increased after taking ϕ_1 into account. TAKE the schematic diagram shown in Figure D1 as an example to explain the defining principle of ϕ_1.

a. For the A-A section, when the mass m_3 is in vibration, its gravity F_{g3} shall be multiplied by $\phi_1 = 1 + k_a$;

b. For the B-B section, when the mass m_3 is in vibration, its gravity F_{g3} shall be multiplied by $\phi_1 = 1 - k_a$; while the gravity F_{g1} of the mass m_1 shall be multiplied by $\phi_1 = 1 + k_a$.

Where:

T_s (F_s) - Drive torque (force) converted to the calculation element, N • m (N);

T_r (F_r) - Static resistance torque (force) converted to the calculation element, N • m (N);

T_1 (F_1) - Maximum torque (force) on the calculation element, N • m (N);

$|T_b|$ ($|F_b|$) - Brake torque (force) converted to the calculation element, N • m (N);

a_s - Acceleration during startup, m/s^2;

a_b - Deceleration during braking, m/s^2.

The load on the calculation element:

$T_1 = T_r + J_2 \cdot a$ or $F_1 = F_r + m_2 a$

E2 Calculated load considering the elastic vibration

The rigid body dynamics method does not reflect the effects of the system flexibility. Since the components of the transmission system are not absolutely rigid, when the external force changes, the system will produce elastic vibration so that the components are subject to greater dynamic load. The elastic vibration load can be analyzed and calculated through the equivalent elastic vibration mechanics model. The dynamic load factor ϕ_5 is introduced to the preliminary design and simplified calculation for consideration. The load of the calculation load in Figure E1 can be calculated according to the Equation (E4).

$$T_2 = T_{(i)} + \phi_5 \Delta T$$
or
$$F_2 = F_{(i)} + \phi_5 \Delta F \qquad (E4)$$

Where:

T_2 (F_2) - Maximum load acting on the calculation element, N • m (N);

$T_{(i)}$ ($F_{(i)}$) - Load acting on the calculation element before the acceleration (deceleration) of the system, N • m (N), which shall be calculated according to the rigid body dynamics model;

ΔT (ΔF) - Variation of the load acting on the calculation element before and after the acceleration (deceleration) of the system, N • m (N), which shall be calculated according to the rigid body dynamics model, $\Delta T = T_{(f)} - T_{(i)}$ or $\Delta F = F_{(f)} - F_{(i)}$;

Annex F

(Reference)

Selection of the rail and wheel combination

F1 When selecting the rail and wheel combination of the tower crane, the calculated load of the wheels shall be determined according to the Article 6.4.4.1. The rail and wheel combination shall be selected from Table F1 or F2 according to this calculated load.

Table F1 Allowable Load Under the Condition of the Tensile Strength of the Wheel Material σ_b = 600MPa

kN

Wheel diameter mm	Rail type							
	P_{18}	P_{24}	P_{33}	P_{38}	P_{43}	P_{50}	QU_{70}	QU_{80}
250	50	45	65	80	-	-	-	-
315	65	60	85	105	110	110	-	-
400	-	80	110	136	140	140	180	200
500	-	-	-	170	180	180	230	250
630	-	-	-	-	230	230	290	320
710	-	-	-	-	-	260	330	360

Table F2 Allowable Load Under the Condition of the Tensile Strength of the Wheel Material $\sigma_b \geq$ 800MPa

kN

Wheel diameter mm	Rail type							
	P_{18}	P_{24}	P_{33}	P_{38}	P_{43}	P_{50}	QU_{70}	QU_{80}
250	70	65	90	110	-	-	-	-
315	90	85	115	145	150	150	-	-
400	-	110	150	190	195	195	250	275
500	-	-	-	240	250	250	320	350
630	-	-	-	-	320	320	405	450
710	-	-	-	-	-	365	460	505

Annex H

(Reference)

Calculated length factor μ

H1 The calculated length l_c of the structural member can be calculated according to the Equation (H1).

$$l_c = \mu_1 \mu_2 \mu_3 \mu_4 l \quad \text{(H1)}$$

Where:

l - Actual geometric length of the part to be calculated in the structural member, or actual geometric length of the part of the structural member as defined by the value of the calculated length factor μ, m;

μ_1 - Calculated length factor related to the support mode of the structural member (not necessarily the same in two planes), SEE Article H2;

μ_2 - Calculated length factor of the variable cross-sectional structural member, SEE Article H3;

μ_3 - Calculated length factor of the compressive structural member considering the action of non-directional force, SEE Article H4;

μ_4 - Calculated length factor of the stressed structural member considering the action of double axial force, SEE Article H5.

H2 Calculated length factor μ_1 related to the support mode of the structural member

H2.1 If the structural member has only one attachment support point in its shaft portion, and the support flexibility of the attachment support point may not be taken into account, the values of the calculated length factor μ_1 can be looked up in Table H1.

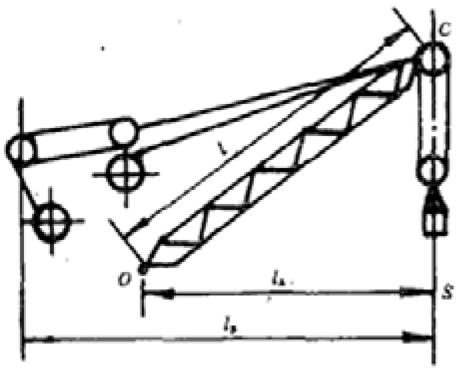


Figure H1

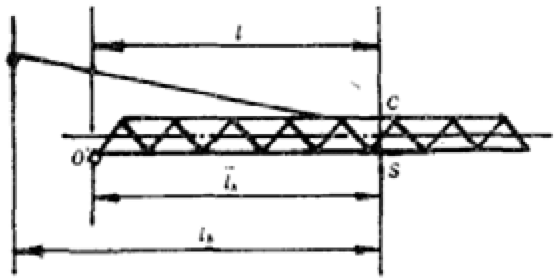


Figure H2

b. For the trolley luffing boom with double hoisting points (Figure H3) and the trolley luffing boom with double and single hoisting points (the central section of the boom is equipped with a hinge point, as shown in Figure H4), when considering the effects of the non-directional force of two boom cables at the same time, if the length of the *OA* segment (l_1) is taken as the geometric dimensions of the structural member, it is advisable to look up the numerators in Table H6 or H7; if the length of the *AB* segment (l_2) is taken as the geometric dimensions of the structural member, it is advisable to look up the denominators in Table H6 or H7. Since the effects of the variable cross section and the double cable of the boom have been taken into account in Tables H6 and H7, μ_2 = 1 and μ_4 = 1 shall be taken after referencing the data in these three tables.

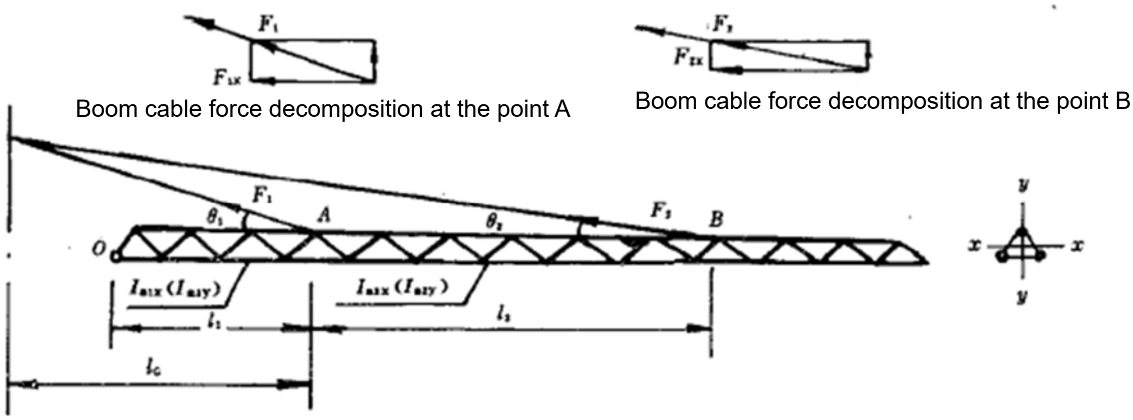


Figure H3

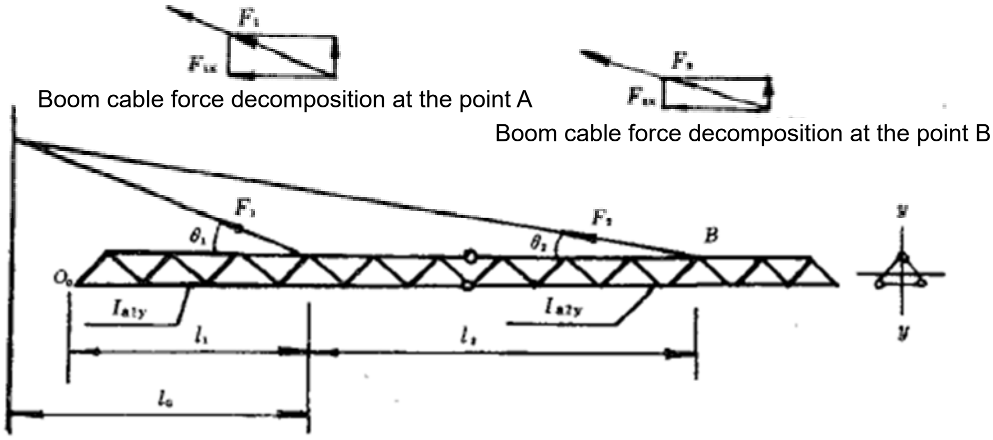


Figure H4

In the Figure H3, Figure H4, Table H6 and Table H7:

l_1, l_2 - The lengths of two segments of boom (*OA* and *AB* segments), m;

I_{a1x}, I_{a2x} - The cross-sectional inertia moments generated when two segments of boom (*OA* and *AB* segments) bypass the horizontal axis (which is *x*-*x*) of the centroid of section, m^4;

I_{a1y}, I_{a2y} - The cross-sectional inertia moments generated when two segments of boom (*OA* and *AB* segments) bypass the vertical axis (which is *y*-*y*), m^4;

F_{1x}, F_{2x} - Components of the tensions F_1 and F_2 in the boom cable in the horizontal direction, $F_{1x} = F_1 \cdot \cos\theta_1$, $F_{2x} = F_2 \cdot \cos\theta_2$, N;

θ_1, θ_2 - Angle between the boom cable and the boom centerline;

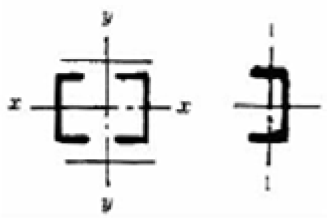
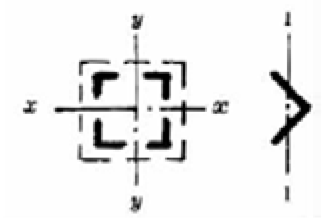
F_1, F_2 - Internal forces in the boom cable under the calculated operating condition, N.

H4.2 For the compressive structural members subjected to the external force of directional force, or the compressive structural members ignoring this favorable factor despite of subjected to the non-directional force, TAKE $\mu_3 = 1$.

H5 Consider the calculated length factor μ_4 of the compressive structural member subjected to the double axial force

When the trolley luffing boom with double hoisting points as shown in Figure H3 is checked to be stable in the hoisting plane, the effects of the integrity and double axial force of the boom functioning at the same time shall be generally taken into account. If the length of the *OA* segment (l_1) is taken as the geometric dimensions of the structural member, it is advisable to look up the numerators in Table H8. If the length of the *AB* segment (l_2) is taken as the geometric

Table H9 Calculation Equation for the Conversion Slenderness Ratio λ_h of the Lattice Structural Members

Item	Section form of the structural member	Batten category	Calculation equation	Symbol meaning
1	a) 	Batten plate	$\lambda_{hy}=\sqrt{\lambda_y^2+\lambda_1^2}$	λ_y - Slenderness ratio of the entire structural member to the imaginary axis λ_1 - Slenderness ratio of the single limb to the 1-1 axis, its calculated length takes the net distance between the batten plates
2		Lacing bar	$\lambda_{hy}=\sqrt{\lambda_y^2+27\frac{A_R}{A_{R1}}}$	A_R - The sum of the gross cross-sectional areas of the chord members cut out by the cross section of the structural member A_{R1} - The sum of the gross cross-sectional areas of the oblique lacing bars cut out by the cross section of the structural member
3	b) 	Batten plate	$\lambda_{hx}=\sqrt{\lambda_x^2+\lambda_1^2}$ $\lambda_{hy}=\sqrt{\lambda_y^2+\lambda_1^2}$	λ_1 - Slenderness ratio of the single limb to the minimum stiffness axis 1-1, its calculated length takes the net distance between the batten plates
4		Lacing bar	$\lambda_{hx}=\sqrt{\lambda_x^2+40\frac{A_R}{A_{R1X}}}$ $\lambda_{hy}=\sqrt{\lambda_y^2+40\frac{A_R}{A_{R1Y}}}$	A_{R_1X} - The sum of the gross cross-sectional areas of various oblique lacing bars in the plane perpendicular to the x-x axis cut out by the cross section of the structural member A_{R_1Y} - The sum of the gross cross-sectional areas of various oblique lacing bars in the plane perpendicular to the y-y axis cut out by the cross section of the structural member

$$K_H = 1.0 - \frac{\Delta_H F_E}{M_H} = 1.0 - \frac{\Delta_H \pi^2 E I_a}{M_H (\mu l)^2} \quad (I3)$$

When calculating the K_H, the structural member has already converted into a uniform cross-sectional hypothetical structural member. Its calculated length $l_c = \mu l$ has also been converted from the length coefficient μ ($\mu = \mu_1 \bullet \mu_2$) (in case of simple supports on both ends or cantilever support, it is advisable to look up the Table H5; in other cases, it is advisable to derive according to the energy method). In order to take I_{ae} into account when calculating Δ_H, the Equation (I3) can be rewritten into:

$$K_H = 1.0 - \frac{\Delta_H \pi^2 E I_{ae}}{M_H (\mu_1 l)^2} \quad (I4)$$

Where: $I_{ae} = I_{amax} / \mu_2^2$

For instance: if there is a bending structural member with one free end and one fixed end (SEE Figure I1).

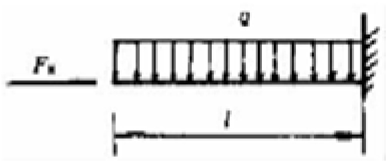


Figure I1

$M_H = \frac{q}{2} l^2$; $\Delta_H = \frac{q l^4}{8 E I_{ae}}$ $\mu = 2$

If this is an unequal cross-sectional structural member, USE the equivalent moment of inertia I_{ae} for equivalence. Then

$$K_H = 1.0 - \frac{q l^4}{8 E I_{ae}} \frac{\pi^2 E I_{ae}}{4 l^2} \frac{2}{q l^2} = 1 - \frac{\pi^2}{16} = 0.38$$

In order to facilitate the calculation, TAKE $C_H = 1 - 0.4 \frac{F_N}{F_E}$, which is the serial number 3 in Table I1.

Annex J

(Reference)

Stress amplitude method for the fatigue strength calculation of the structure

J1 The fatigue strength of the structure shall be checked according to the Equations (J1), (J2) and (J3).

$$|\sigma_a| \leqslant (\sigma_a) \quad (J1)$$

$$|\tau_a| \leqslant (\tau_a) \quad (J2)$$

$$\left(\frac{\sigma_{ax}}{(\sigma_{ax})}\right)^2 + \left(\frac{\sigma_{ay}}{(\sigma_{ay})}\right)^2 - \frac{\sigma_{ax}\sigma_{ay}}{(\sigma_{ax})\ (\sigma_{ay})} + \left(\frac{\tau_{axy}}{(\tau_{axy})}\right)^2 \leqslant 1.1 \quad (J3)$$

Where:

σ_a - Maximum calculated tensile (compressive) stress amplitude, SEE Article J2, MPa;

τ_a - Maximum calculated shear stress amplitude, SEE Article J2, MPa;

σ_{ax} - σ_a whose direction is parallel to the x axis, MPa;

σ_{ay} - σ_a whose direction is parallel to the y axis, MPa;

τ_{axy} - τ_a whose plane is perpendicular to the x axis, and whose direction is parallel to the y axis, MPa;

(σ_a) - Allowable tensile (or compressive) fatigue stress amplitude, SEE Article J3, MPa;

(τ_a) - Allowable shear fatigue stress amplitude, SEE Article J3, MPa;

(σ_{ax}), (σ_{ay}) - (σ_a) corresponding to σ_{ax} and σ_{ay}, MPa;

(τ_{axy}) - (τ_a) corresponding to τ_{axy}, MPa.

The cyclic normal stress (tensile or compressive stress) in the positions to be checked, regardless of x direction or y direction, shall meet the Equation (J1). All kinds of cyclic shear stress shall meet the Equation (J2). The Equation (J3) is only used for checking the compound effect of the above three kinds of (or two of them) cyclic stress.

When there is only one kind of cyclic stress in the positions to be checked, or the effect of one kind of cyclic stress on the structural members is significantly greater than other cyclic stresses, the effects of other cyclic stresses can be ignored, and the fatigue checking is only conducted to this kind of cyclic stress. USE the Equation (J1) or (J2). At this time, for the structural members whose total number of stress cycles is less than 1.6×10^4, the fatigue checking may not be performed.

J2 Calculated stress amplitude

The maximum calculated stress σ_{max} or τ_{max} shall be determined according to the provisions of Article 5.7.1. The maximum calculated stress amplitude shall be calculated according to the Equations (J4) and (J5).

$$\sigma_a = 0.5\ (\sigma_{max} - \sigma_{min}) \qquad (J4)$$

$$\tau_a = 0.5\ (\tau_{max} - \tau_{min}) \qquad (J5)$$

J3 Allowable fatigue stress amplitude

The numerical values depend on the working levels of the structure, material categories of the structural members, connection type of the connectors (stress concentration level), cycle characteristics of the stresses (SEE Article 5.7.2), etc.

a. $[\sigma_a]$ of the base metal

When σ_{max} refers to the tensile stress, TAKE $[\sigma_a]$ as $[\sigma_{at}]$ (allowable tensile fatigue stress amplitude). When σ_{max} refers to the compressive stress, TAKE $[\sigma_a]$ as $[\sigma_{ac}]$ (allowable compressive fatigue stress amplitude). $[\sigma_{at}]$ and $[\sigma_{ac}]$ shall be calculated according to the equations in Table J1.

When the numerical value of $[\sigma_{at}]$ or $[\sigma_{ac}]$ is greater than $0.375\sigma_b (1 - \chi)$, TAKE $[\sigma_{at}] = 0.375\sigma_b (1 - \chi)$ or $[\sigma_{ac}] = 0.375\sigma_b (1 - \chi)$.

b. $[\tau_a]$ of the base metal

The allowable shear fatigue stress amplitude shall be determined by the Equation (J6).

$$[\tau_a] = \frac{[\sigma_{at}]}{\sqrt{3}} \qquad (J6)$$

Annex N

(Reference)

Motor overload checking

N1 Motor of the hoisting mechanism

The pull-in torque of the motor shall meet the Equation (N1).

$$T_s \geq K_u \frac{9.55 F_Q v_h}{n_m \eta} \tag{N1}$$

Where:

T_s - Pull-in torque of the motor during reference electrical continuity, N • m;

K_u - Coefficient, considering the voltage loss, allowable deviation of the maximum torque, overload value of the test load, mechanism acceleration and other factors.

Winding asynchronous motors: K_u = 2.0 to 2.2,

Cage asynchronous motors (including the multi-speed motor, the speed values are different): K_u = 2.2 to 2.4;

DC motors: K_u = 1.4;

F_Q - Rated hoisting load, N;

v_h - Rated hoisting speed, m/s;

η - Total efficiency of the transmission mechanism;

n_m - Slewing speed of the motor, r/min.

N2 Motors of the running and trolley luffing mechanisms

$$T_s \geq (F_r + F_{W2}) \frac{D_W}{2 i \eta Z_m} \tag{N2}$$

Where:

T_s - Pull-in torque of the motor during reference electrical continuity, N • m;

F_r - Running static resistance, for the entire machine, determined according to the Article 6.2.4.1; for the trolleys, determined according to the Article 6.2.2.2.1, N.

D_W - Wheel diameter, m;

i - Total transmission ratio;

Z_m - Number of motors.

N3 Motor of the slewing mechanism

$$T_s \geqslant \frac{K_u (T_\mu + T_{s1} + T_{W2})}{i\eta Z_m} \tag{N3}$$

Where:

T_s - Pull-in torque of the motor during reference electrical continuity, N • m;

K_u - Coefficient, winding asynchronous motors: K_u = 1.5 to 1.54; cage asynchronous motors: K_u = 1.6 to 1.63 (the lower limit is taken for the no-load and light-load startups); DC motors: K_u = 1;

T_μ - Torque of the slewing frictional resistance, N • m;

T_{s1} - Slope resistance torque, SEE Article 4.2.1.6, N • m;

T_{W2} - Wind resistance torque generated by the calculated wind pressure P_{W2}, N • m.

N4 Motor of the luffing jib mechanism

$$T_s = K_u T_r \tag{N4}$$

Where:

K_u - Coefficient, winding asynchronous motors: K_u = 1.54; cage asynchronous motors: K_u = 1.63; DC motors: K_u = 1;

T_r - The sum of the static resistance torques converted by the deadweight loads of the boom and balanced system, hoisting load of the unbalanced section, wind load generated by the calculated wind pressure p_{W2} and frictional resistance of each rotational hinge point of the boom system to the motor shaft in case of considering the efficiency of the transmission system, N • m.

Annex O

(Reference)

Heat checking of the asynchronous motors

It is advisable to use the average loss method. The equivalent torque method or equivalent current method can also be used for accurate motor heat checking according to the different types of motor. The following methods can also be adopted for approximate calculation.

O1 Approximate method for heat checking

$$P_n \geq \frac{T_{re} \cdot n_m}{9550 \ \eta K_z} \tag{O1}$$

Where:

P_n - Rated power of the motor, its working system is S_3, and the electrical continuity $JC\%$ is the same as the value of the actual mechanism, kW;

T_{re} - Equivalent average resistance torque for the most unfavorable operating cycle, calculated according to the Equation (O2), N • m;

n_m - Slewing speed of the motor, r/min;

K_z - Coefficient, determined according to the Article O2.

$$T_{re} = \sqrt{\frac{\sum T_{ri}^2 t_i}{\sum t_i}} \tag{O2}$$

Where:

T_{ri} - Corresponding motor resistance torque within the operating time t_i, including the starting and braking torques, N • m;

t_i - Each operating time in the operating cycle of the mechanism, excluding the downtime, s.

When the data in the Equation (O2) cannot be determined, it is advisable to take the approximation:

$$T_{re} = T_r \cdot K_G \tag{O3}$$

Annex P

(Reference)

Stability checking of the oil cylinders

P1 The stability of the oil cylinders can be checked according to the Equation (P1).

$$F_N \leqslant \frac{F_{cr}}{K_{nh}} \qquad (P1)$$

Where:

F_N - Axial load of the oil cylinder, calculated according to Article 6.2.5.1, N;

F_{cr} - Critical load of the oil cylinder, calculated according to the Equation (P2), N;

K_{nh} - Safety factor, K_{nh} = 2.5 to 3.5.

$$F_{cr} = \frac{k_s \cdot \pi^2 E I_a}{l^2} \qquad (P2)$$

Where:

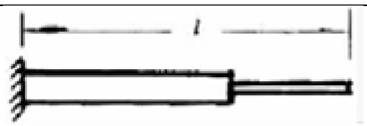
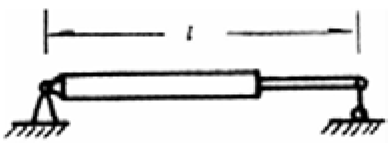
k_s - Support condition factor, determined according to Table P1;

E - Elastic modulus of the material, MPa;

I_a - Inertia moment of the piston rod's cross section, mm^4;

l - Calculated length of the oil cylinder, determined according to Table P1, mm.

Table P1 Support Condition Factor k_s of the Oil Cylinder

Support mode and calculated length l	Description	k_s
	Fixed on one end, while free on the other end	0.25
	Hinged connection on both ends	1

Annex Q

(Reference)

Method of determining the fatigue strength limit σ_{rk}

Q1 Fatigue strength limit of the polished test rod

The maximum stress when the intact rate of 90% is remained after the polished test rod is subjected to an infinite number (number of cycles is greater than 10^7) of symmetrical cycles under the rotational bending test conditions is called fatigue strength limit, which is expressed by σ_{-1}.

When the fatigue strength limit of the material is not given, the fatigue strength limit value for the symmetrical cycle of the structural steel can be calculated according to the Equations (Q1), (Q2) and (Q3).

Tensile and compressive: $$\sigma_{-1t} = 0.23(\sigma_b + \sigma_s) \quad \text{(Q1)}$$

Bending: $$\sigma_{-1b} = 0.27(\sigma_b + \sigma_s) \quad \text{(Q2)}$$

Twisting: $$\tau_{-1} = \frac{\sigma_{-1}}{\sqrt{3}} \quad \text{(Q3)}$$

Q2 Effects of the shape, surface status and dimensions

The transmission parts are more complex than the test rods in shape, along with rough surface and larger dimensions. Therefore, the fatigue strength $(\sigma_{-1})_G$, $(\tau_{-1})_G$ of the transmission part is less than the fatigue strength σ_{-1}, τ_{-1} of the polished test rod. The effects of the shape, surface status and dimensions on the fatigue strength is considered through corresponding coefficients according to the Equations (Q4) and (Q5):

$$(\sigma_{-1})_G = \frac{\varepsilon_\sigma \beta}{K_\sigma}\sigma_{-1} \quad \text{(Q4)}$$

$$(\tau_{-1})_G = \frac{\varepsilon_\tau \beta}{K_\tau}\tau_{-1} \quad \text{(Q5)}$$

Where:

K_σ, K_τ - Shape coefficient;

β - Surface status coefficient;

ε_σ, ε_τ - Dimension coefficient.

The shear stress caused by the transverse force does not need to be considered for stress concentration.

Q2.1 The values of the shape coefficients K_σ and K_τ are rarely derived directly from the fatigue test. Most of the curves are the data of the theoretical stress concentration factors a_σ and a_τ. At this time, the sensitivity coefficient q of the material is converted into the effective stress concentration factor, which is:

$$K_\sigma = 1 + q_\sigma (\alpha_\sigma - 1), \quad K_\tau = 1 + q_\tau (\alpha_\tau - 1) \quad \text{(Q6)}$$

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} \quad \text{(Q7)}$$

Where:

r - Fillet radius of the notch (such as groove and hole), mm;

$\sqrt{a}$ - Material constant, looked up from Figure Q1, $\sqrt{mm}$.

Q2.2 Surface status coefficient β

The curves of the most common surface status coefficients β_1 and β_2 are listed in Figure Q2. The curves 1 to 5 indicate the surface processing coefficient β_1. The curves 6 to 7 indicate the corrosion coefficient β_2.

The numerical values of the surface strengthening coefficient β_3 can be looked up in Table Q1. If β_1, β_2 and β_3 show up at the same time, the continued multiplication shall not be used. If only processed with cutting, β is equal to β_1. If processed with cutting before strengthening, β is equal to β_2. Whether to be strengthened or not in the corrosion environment, β is equal to β_3.

Table Q1 Values of the Surface Strengthening Coefficient β_3

Strengthening method	Heart intensity MPa	β_3		
		Smooth specimen	Stress-concentrated specimen	
			$K_\sigma < 1.5$	$K_\sigma \geq 1.8$ to 2.0
High frequency quenching	600 to 800	1.5 to 1.7	1.6 to 1.7	2.4 to 2.8
	800 to 1 000	1.3 to 1.55	1.4 to 1.5	2.1 to 2.4
Nitriding	900 to 1 000	1.1 to 1.25	1.5 to 1.7	1.7 to 2.1
Carburization	400 to 600	1.8 to 2.0	3.0	3.5
	700 to 800	1.4 to 1.5	2.3	2.7

Annex T

(Reference)

Calculation method of the reel wall thickness

This calculation method is applicable to the wall thickness calculation of the steel plate welding reels.

T1 Wall thickness calculation of the reel

The reel length of the tower crane is generally designed to be less than 3 times the diameter of the reel. It can be seen from the force analysis that: since the reel wall is subjected to smaller bending and torsional stresses, the calculation shall be performed according to the compressive stress and local bending stress in the position where the wire rope is wound out.

a. The compressive stress σ_p of the reel wall in the position where the wire rope is wound out shall be calculated according to the Equation (T1).

$$\sigma_c = 0.5 \frac{F_{max}}{\delta p_d} \quad \text{(T1)}$$

Where:

σ_c - Compressive stress in the position where the wire rope is wound out, MPa;

δ - Wall thickness of the reel, mm;

p_d - Axial winding pitch, mm;

for the grooved reels:

$p_d = p$ (p refers to the rope pitch)

for the smooth reels:

$p_d = 1.01d$ (d refers to the diameter of the wire rope)

b. The local bending stress of the reel wall in the position where the wire rope is wound out shall be calculated according to the Equation (T2).

$$\sigma_{be} = 0.96 F_{max} \frac{1}{\sqrt{D_d \delta^3}} \quad \text{(T2)}$$

The wall thickness of the side plate shall be checked according to the Equation (T5).

$$\sigma_{be} = 1.44\left(1 - \frac{2}{3}\frac{D_m}{D_d}\right)\frac{F_N}{\delta_1^2} \leqslant [\sigma] \quad (T5)$$

Where:

σ_{be} - Bending stress of the reel's side plate, MPa;

F_N - Calculated axial force, N, generally taken as $0.1F_{rmax}$;

$[\sigma]$ - Allowable stress of the material, calculated according to the Equation (T4), MPa;

D_m, D_d, δ_1 - Geometric dimensions, SEE Table T1, mm.

T3 Few notes

a. This calculation method is also applicable to the wall thickness calculation of multi-layer rolled steel plate welding reels.

b. The effect of bending and torsion shall also be taken into account when the reel length is greater than three times the diameter of the reel.

c. For the large-size reel with a diameter of greater than or equal to 1,200mm and a length *l* of greater than $2D_d$, especially the steel plate welded large-size thin-walled reel, the reel wall stability shall be checked.

Annex U

(Reference)

Calculation method of the slewing bearing selection

U1 Calculation of the equivalent external load

According to the structural type of slewing bearing, the calculations shall be respectively performed according to the following equations:

a. Single-row and double-row ball slewing bearing

$$F_{eq} = F_v + \frac{4.37M}{D_0} + 3.44F_h \qquad \text{(U1)}$$

b. Single-row cross roller slewing bearing

$$F_{eq} = F_v + \frac{4.5M}{D_0} + 2.5F_h \qquad \text{(U2)}$$

c. Three-row roller slewing bearing

$$F_{eq} = F_v + \frac{4.5M}{D_0} \qquad \text{(U3)}$$

Where:

F_{eq} - Equivalent external load of the slewing bearing, N;

F_v - Axial force acting on the center of the slewing bearing, N;

F_h - Horizontal force acting on the center of the slewing bearing, N;

M - Overturning moment acting on the center of the slewing bearing, N • m;

D_0 - Center circle diameter of the roller, for the double-row ball slewing bearing and three-row roller slewing bearing, D_0 refers to the center circle diameter of the roller in the lower row, m.

U2 Calculation of the equivalent static capacity of the slewing bearing

a. Single-row ball type

$$F_0 = f_0 d_0^2 Z_0 \sin\theta \tag{U4}$$

b. Single-row cross roller type

$$F_0 = f_0 d_0^2 l_0 \frac{Z_0}{2} \sin\theta \tag{U5}$$

c. Double-row ball type

$$F_0 = f_0 d_0^2 Z_0 \tag{U6}$$

d. Three-row roller type

$$F_0 = f_0 d_0 l_0 Z_0 \tag{U7}$$

Where:

F_0 - Equivalent static capacity of the slewing bearing, N;

d_0 - Diameter of the steel ball or roller, for the conical rollers, d_0 refers to the medium diameter, mm;

θ - Nominal contact angle, for the single-row ball type, θ is equal to 50°; for the cross roller type, θ is equal to 45°;

f_0 - Static capacity coefficient, determined according to the raceway surface hardness, when the surface hardness is not less than HRC58, f_0 is equal to 49 for the single-row and double-row ball type; f_0 is equal to 98 for the cross roller and three-row roller type, N/mm;

l_0 - Contact length of the roller, for the single-row cross roller type, the roller is cylindrical, and l_0 is equal to $0.75d_0$; for the three-row roller type or the cross roller type using the tapered roller or drum-shaped roller, l_0 is taken as $0.85d_0$, mm;

Z_0 - Number of steel balls or rollers, for double-row ball type or three-row roller type, Z_0 refers to the number of rollers on the upper row, Z_0 is calculated according to the Equation (U8).

$$Z_0 = \frac{\pi D_0 1000 - d_0}{d_0 + b} \tag{U8}$$

A_S - Effective cross-sectional area of the thread, the A_S value of the standard bolt can be selected according to Table U2, mm^2.

$$F_N = \frac{K_g M}{D_1 n} - \frac{F_V}{n} \quad \text{(U12)}$$

Where:

K_g - Load distribution coefficient,

when K_g = 4.37, used for the single-row and double-row ball type,

when K_g = 4.10, used for the cross roller and three-row roller type;

n - Number of the mounting bolts on the inner ring of the slewing bearing;

D_1 - Distribution circle diameter of the mounting bolt hole on the inner ring of the slewing bearing, m.

b. Fatigue strength calculation

The fatigue strength of the mounting bolts for the slewing bearing shall meet

$$\sigma_a \leqslant \frac{\sigma_{ak}}{K_{na}} \quad \text{(U13)}$$

Where:

σ_{ak} - Fatigue limit stress amplitude of the bolt, selected according to Table U3, MPa;

K_{na} - Fatigue strength safety factor, SEE Table U4;

σ_a - Alternating stress amplitude of the bolt, calculated according to the Equation (U14), MPa.

$$\sigma_a = \frac{0.07 F_{max}}{A_{d1}} \quad \text{(U14)}$$

Where:

A_{d1} - Minimum cross-sectional area of the bolt, mm^2.

Annex V

(Reference)

Current carrying capacity of the wire

V1 Calculation equation for the current carrying capacity of the wire

$$I_z = K_a K_t K_j I_g \tag{V1}$$

Where:

I_z - Current carrying capacity of the wire, A;

K_a - Correction coefficient for the parallel laying of multiple cables or perforated wires, generally taken as 0.9 for the perforated wires, and taken as 0.8 for the cables;

K_t - Correction coefficient of the ambient temperature, SEE Table V1 for the commonly used values, the K_t value can be calculated according to the Equation (V2);

K_j - Correction coefficient of the electrical continuity with repeated short-term working system, the operating cycle time is taken as 10min, SEE Table V2 for the commonly used values, the the K_j value can also be calculated according to the Equation (V3);

I_g - Reference value for the current carrying capacity of the wire and cable, SEE Table V3 for the commonly used values, A.

$$K_t = \sqrt{\frac{t_1 - t_0}{t_1 - t_2}} \tag{V2}$$

Where:

t_1 - Maximum operating temperature of the wire core, °C;

t_0 - Operating ambient temperature, °C;

t_2 - Rated operating ambient temperature, 25°C (or 45°C).

$$K_j = \sqrt{\frac{1 - e^{-\frac{600}{\tau}}}{1 - e^{-\frac{600JC}{\tau}}}} \tag{V3}$$

Additional information:

This Standard was proposed by the Ministry of Development of the People's Republic of China (now known as the Ministry of Housing and Urban-Rural Development of the People's Republic of China).

This Standard shall be under the jurisdiction of the Beijing Institute of Construction Machinery of the Technical Committee for Standardization of Machinery and Vehicles under the Ministry of Development.

Beijing Institute of Construction Machinery under the Ministry of Development and Changsha Institute of Construction Machinery under the Ministry of Development were responsible for drafting this Standard.

Main drafters of this Standard: Xie Yaoting (chief editor), Hu Zhaolin (chief editor), Chen Jianyuan, Gu Dimin, Chen Zhenyi, Jia Zhiquan, Li Yishen, Zhang Guorui, Ye Yuanhua, Tang Mingqing, Ding Yi, Luo Shijun, Mei Kun, Zhou Zhong, Cong Jiaxu, Zhang Desheng, Meng Xiaoping, Gao Yongxing, Huang Gongcai, Zheng Dianyi, Hu Yuankai, Zhu Yiwei, Lin Gang and Fang Yanlong.

This Standard shall be interpreted by Beijing Institute of Construction Machinery under the Ministry of Development and Changsha Institute of Construction Machinery under the Ministry of Development.

END

------ The End ------