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Fatigue life prediction for tower cranes under moving load

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Abstract For the tower crane, the luffing trolley and cargoes are equivalent to concentrated force. This study established a moving load model. The transient dynamic analysis method was used to carry out dynamic simulation, analyze and obtain the vibration characteristics of the boom under luffing and lifting, and calculate the fatigue life of the key components of the boom based on the S-N curve. The analysis results showed that the greater the luffing speed, the greater the arm end vibration amplitude. The lifting speed has influence on the amplitude and frequency of vibration. Under the worst working conditions, the minimum service life of the top chord 638 of the most dangerous component was 17.96 years, which can meet the requirements of the safety regulations for tower cranes. At the same time, the damage degree of the dangerous component during luffing was greater than that during lifting.

1. Introduction

The tower crane is a kind of construction machinery that completes cargo transportation through luffing and lifting. It has the advantages of wide adaptability, large turning radius, high lifting height, etc. To improve production efficiency, the operating speed and lifting capacity of tower cranes are increasing. Thus, the mechanical movement may cause greater deformation and vibration of the structure. The resulting alternating load will not only cause fatigue damage to the structure, but also cause discomfort to operators.

Many scholars have conducted extensive research work on the vibration safety of cranes. Chen et al. [1] applied the computational fluid dynamics (CFD) method to analyze the shape coefficient, angle wind coefficient and wind pressure height change coefficient of the tower crane body and boom. Fu et al. [2] calculated the vibration response of the boom tower crane under static wind load and pulsating wind load. The results showed that the tower crane had large vibration under the pulsating wind load. The connection between the fourth boom and the fifth boom of the tower crane was vulnerable to maximum stress. Ma et al. [3] used the S-N curve to calculate the wind-induced vibration fatigue damage of the tower structure. Klinger [4] analyzed the fatigue damage problem caused by wind vibration.

Li et al. [5] believed that the forced vibration of the structure is the response under the excitation force of a fixed period. However, the luffing and lifting of the crane will also cause the boom to vibrate in the vertical direction. Tao et al. [6] simulated the lifting and slewing of tower cranes through transient analysis, obtained the stress-time history of key units, and studied the time-varying reliability considering fatigue cumulative damage under the most dangerous working conditions. Based on the wavelet multi-resolution analysis method, Zhao et al. [7] used a vibration signal on the beam to identify the damage of cracked beam according to the dynamic response characteristics of simply supported beam under moving load. Dong et al. [8] simplified the tower crane as a cantilever beam that vibrated in the vertical plane of the boom and equivalent the luffing trolley and cargo to a moving mass. Based on the Euler Bernoulli beam theory, the vibration differential equation of the moving mass cantilever system was established, and the influence law of luffing and lifting on the structural vibration of the whole machine was analyzed. Chen et al. [9] simulated the influence of the trolley traveling on the boom of the ship unloader and obtained the dynamic response of the boom head and midspan position at different trolley traveling speeds.

When the crane luffing trolley moves on the boom, it forms an interactive time-varying vibration system with the whole structure. The structural vibration under the excitation of moving objects is complex, which is the main factor that causes crane metal structure to bear cyclic stress. Wei et al. [10] pointed out that the influence of moving load was usually ignored in the existing fatigue assessment of crane metal structures, resulting in large error of life prediction. However, this kind of vibration did not attract the attention of engineers, and the calculation and prevention standards for this kind of vibration were not mentioned in the crane design specifications. Frequent vibration is bound to affect the overall working effect and structural fatigue of tower crane. In this paper, the luffing trolley and cargo were equivalent to the moving concentrated force. On this basis, the moving load model was established. The transient dynamic analysis method was used to carry out dynamic simulation, analyze the vibration characteristics of the boom under luffing and lifting, and calculate the fatigue damage and fatigue life of the key components of the boom under severe conditions based on the S-N curve. It is of great significance for further vibration control.

2. Finite element model of tower cranes

The research object in this paper was QTZ25 tower crane, which consisted of tower body, boom, balance arm, tie rod, tower top and auxiliary structures. These structures are composed of steel pipes, angle steel, channel steels and square steels of different sizes. The maximum operating range was 30 m, the maximum independent lifting height was 26 m, and the maximum lifting weight was 25000 N. The material of the main components of the tower crane was Q235. Q235 has good toughness and strength. Also, it is easy to weld. Thus, it is commonly used for welding, riveting, or bolted steel structural components. It is widely used in wide fields such as construction machinery, transmission towers, bridges. Owing to the calculation scale, a finite element model was established after reasonable simplification of the tower crane. Beam 188 was selected for the tower body, boom and balance arm. Bar element link 180 was selected for the tie rod. Each connection point for structural components of the tower body, boom, balance arm and tower cap was taken as a node. The finite element model formed is shown in Fig. 1. There were 337 nodes and 840 elements in total [11].

3. Vibration characteristics of jib under moving load

3.1 The influence of translation motion on vibration characteristics of jib

The luffing process of tower crane is that the luffing trolley moves from one end of the boom to the other, which will cause

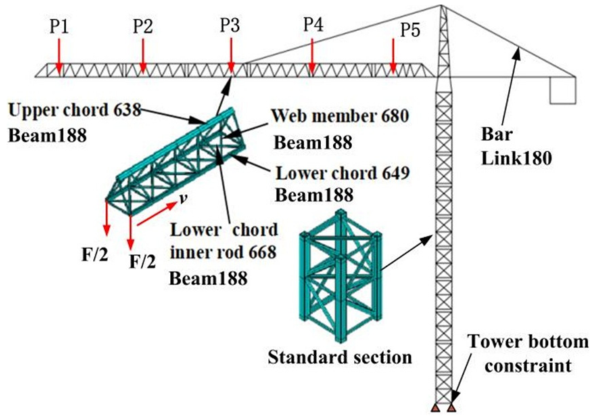


Fig. 1. Finite element model of tower crane.

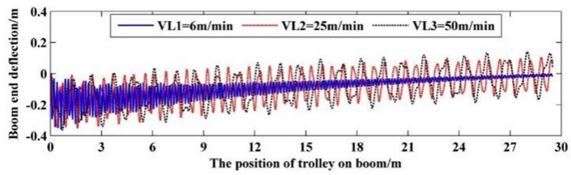


Fig. 2. The influence of moving speed on boom end vibration.

vibration and related chain reaction of the tower crane. In order to accurately reflect the vibration characteristics of tower crane under moving load during luffing. Meanwhile, the bending and torsional deformation of the tower body of a small lifting height tower crane is much smaller than the bending deformation of the lifting arm. The moving load was simplified as a concentrated force, which can improve computational efficiency. The load was symmetrically applied to the two lower chords of the lifting arm and moved from the end of the lifting arm towards the direction of the tower body (as Fig. 1). This paper established a moving concentrated force model in ANSYS, simplified the trolley and lifting load into a single moving concentrated force, used the transient dynamic analysis method, loaded the corresponding nodes on the boom at a certain time interval, and simulated the whole process of moving load from the lower chord at the end of the boom to approaching the tower body. In order to ensure the accuracy of the simulation results, the length of the setting unit of the lower outer chord of the boom was 0.03 m. According to the requirements of the tower crane specification, the load under the worst working condition was 25000 N. The load horizontal movement speed VL was 6 m/min, 25 m/min, 50 m/min, and the load time interval was 0.3 s, 0.072 s, 0.036 s.

Fig. 2 shows the vibration characteristics of the boom end when the load moved from the boom end to the tower. Analysis showed that different moving speeds impacted the vibration amplitude and frequency of the boom. It can be seen from the figure that when the load was at the end of the boom, there was minor difference in the amplitude of the boom end at different moving speeds, which were 0.209 m, 0.229 m and 0.243 m, respectively. When the load was at the root of the

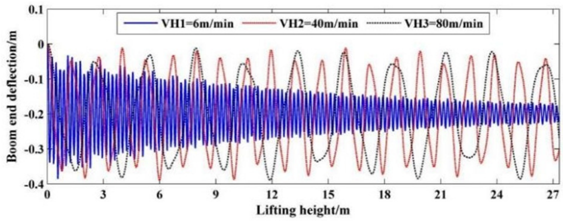


Fig. 3. The influence of lifting speed on boom vibration.

tower boom, the amplitude of the boom end was 0.012 m, 0.145 m and 0.201 m at different moving speeds. It indicated that the greater the load moving speed, the greater the amplitude of the boom end vibration. When the moving speed was small, the vibration amplitude of the boom end decreased obviously and the vibration frequency was high as the trolley gradually approached the tower body. When the moving speed was large, the vibration amplitude of the boom end decreased slightly as the trolley gradually approached the tower body, but the change was not significant.

3.2 The influence of lifting motion on vibration characteristics of jib

The lifting speed is the main reason for the boom vibration. In order to analyze the influence of the lifting speed on the boom vibration during the lifting process, this paper simulated and calculated the vibration characteristics of the boom end point when the load lifting speed VH was 6 m/min, 40 m/min and 80 m/min, respectively. The vibration curve is shown in Fig. 3. The analysis showed that the lifting speed had influence on the amplitude and frequency of vibration. In the initial stage of lifting, the lifting speed had little influence on the change of boom end vibration deflection and amplitude. With the increase of lifting height, the boom end vibration deflection, vibration amplitude and vibration period increased with the increase of lifting speed. When the lifting speed was small, the amplitude of the boom end decreased gradually with the increase of the lifting height. When the lifting speed was large, the vibration amplitude had minor changes.

The lifting speed of the load is the main parameter that affects the dynamic stress of the boom. Generally speaking, the dynamic stress increases with the increase of lifting speed. During the lifting process, the state parameters (deflection, bending moment, shear force, acceleration, etc.) of each section on the lifting arm are constantly changing, and research pays particular attention to the changes in the parameters at the end of the lifting arm. Obviously, the dynamic deflection of the arm end is greater than its static deflection. The impact coefficient $\boldsymbol{I}_{\mathrm{u}}$ is introduced here in to reflect the variation of deflection at a certain position of the lifting arm under moving load.

$$I_u = \frac{R_d(x) - R_s(x)}{R_s(x)}. \tag{1}$$

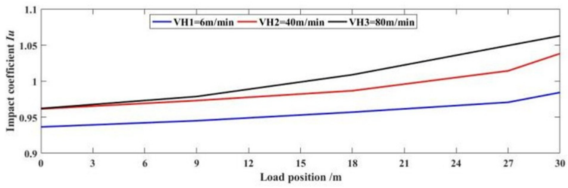


Fig. 4. Trend chart of maximum impact coefficient on the boom under different lifting speeds and loads.

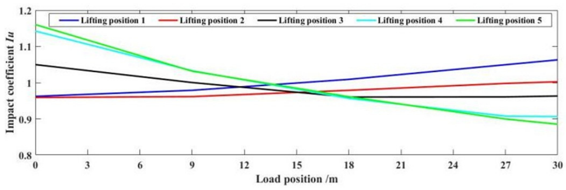


Fig. 5. Trend chart of maximum impact coefficient on the boom at different lifting positions.

There, $R_d(x)$ and $R_s(x)$ respectively represent the maximum dynamic deflection and static deflection at position x of the lifting arm during the moving load process. For tower cranes, the maximum dynamic deflection and static deflection are located at the end of the boom. Their static deflection can be obtained based on static analysis.

Fig. 4 shows the trend of the maximum impact coefficient with respect to the lifting speed when lifting a heavy object at the boom end with a load of 25000 N. From the figure, the maximum impact coefficient increases with the increase of lifting speed. At the same time, the impact coefficient at each position on the boom is not the same. Although the boom is a cantilever structure, the maximum impact coefficient at the end of the entire boom is not the highest due to the load being placed at the end of the boom. The closer it is to the tower body, the greater the impact coefficient.

To study the influence of different lifting positions on the impact coefficient, five loading points were selected on the lifting arm in this paper, as shown in the Fig. 1. Each loading point had a load of 25000 N at the same lifting speed, and the variation trend of the maximum impact coefficient on the lifting arm is analyzed as shown in Fig. 5. Positions 1 and 2 are closer to the end of the boom. Due to the load, the maximum impact coefficient of the boom end is smaller. The closer it is to the tower body, the greater the impact coefficient. Due to the connection between the tower top and the lifting arm with a pull rod, the impact coefficient of the arm end is larger when the lifting load is at positions 3, 4 and 5. The closer the arm end is to the tower body, the smaller the impact coefficient.

Fig. 6 shows the trend of the maximum impact coefficient at each position on the boom as a function of load when lifting a heavy object at the same lifting speed at position 1. From the figure, it can be seen that the impact coefficient of the boom end slightly increases when the load is small. Overall, the load has minor impact on the impact coefficient.

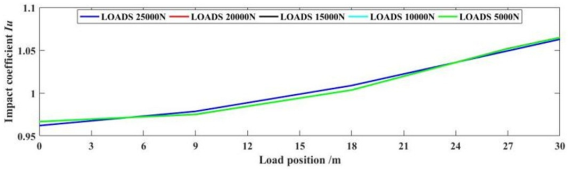


Fig. 6. Trend chart of maximum impact coefficient on jib with different lifting weights.

4. Fatigue life prediction of key components of tower crane

4.1 Linear cumulative fatigue damage theory

The most representative of the linear fatigue cumulative damage theory is the Miner damage theory, which assumes that the fatigue damage degree of the structure or component is linearly related to the number of stress cycles: $D=\sum_{i=1}^{l}\frac{n_i}{N_i}=1$. *l* is the stress level series of variable amplitude load; n_i is the number of cycles under the level *i* stress; N_i is the fatigue life under the level *i* stress. According to the linear damage theory, the damage of stress σ_i to the material is D/N_i, and the total damage to the material after N_i times is $\frac{n_i D}{N_i}$. When the damage of all levels of stress to the material accumulates to the critical value *D*, the material is destroyed [12]. Due to the inherent flaws of Miner's law, it cannot provide accurate fatigue life estimation, and the predicted results have significant dispersion. However, using Miner's law to estimate fatigue life is simple and convenient, and has been widely used in practical engineering applications.

4.2 S-N curve of material

The main indicators characterizing the fatigue resistance of materials include fatigue limit, fatigue strength and fatigue life [13]. The fatigue performance of materials can be represented by the curve between the fatigue limit and the number of cycles *N*, also known as the *S-N* curve. It is generally obtained through fatigue testing. For steel and its alloys, the *S-N* curve usually has a horizontal asymptote, as shown in Fig. 7. When the life *N* approaches infinity, the corresponding stress is the fatigue limit S_r of the material. In theory, when $S < S_r$, the component can withstand infinite cycles without damage.

The *S-N* curve is usually expressed as $NS^m = C$ in the form of power function. m and *C* are fatigue parameters of materials, which are generally obtained from constant amplitude fatigue tests.

4.3 Fatigue life prediction of key components of tower crane

The fundamentals of applying the fatigue cumulative damage theory to the fatigue analysis in time domain is: first, the time domain analysis of the boom vibration response under the moving load of the tower crane is carried out to determine the stress response time history of the parts (dangerous components) that need to check the fatigue life, and the stress cycle amplitude and the corresponding number of cycles are counted to calculate the corresponding fatigue cumulative damage [14, 15]. The basic idea is shown in Fig. 8.

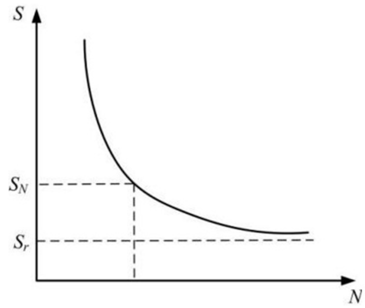


Fig. 7. The S-N curves of material.

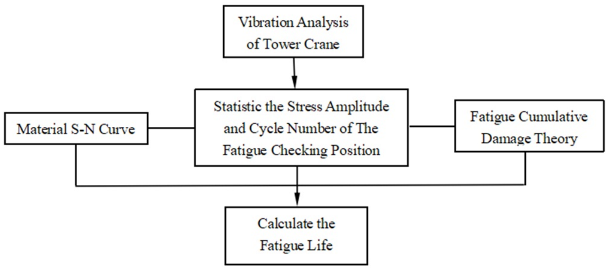


Fig. 8. Fatigue life time domain analysis steps.

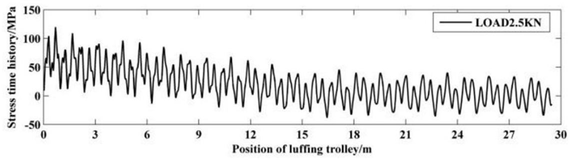
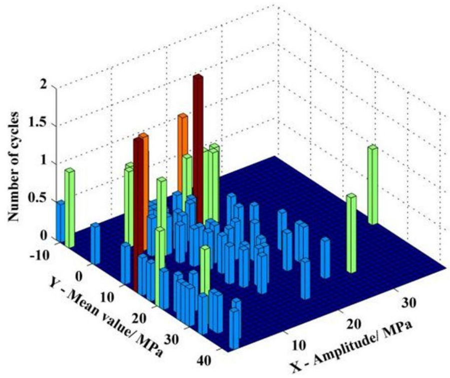
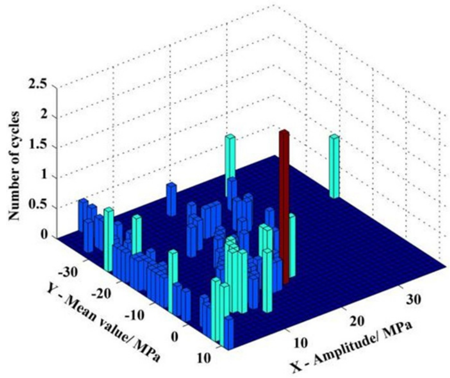


Fig. 9. Stress time history of top chord 638.

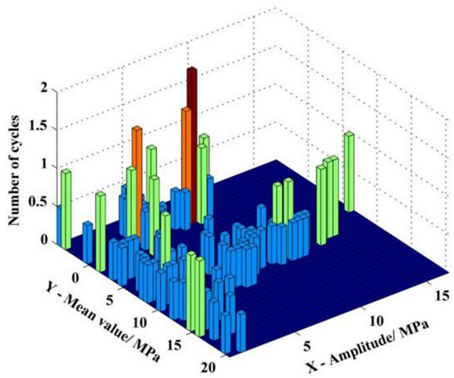
The fatigue failure of tower crane first occurred at the connection between the pull rod and the boom [16], which was close to the standard section of the boom on the left side of the pull rod point. According to the dynamic characteristics analysis, under the rated lifting capacity, the greater the luffing speed and lifting speed, the greater the stress time history amplitude of dangerous components on the boom. Therefore, the rated lifting capacity is 2.5 KN, the maximum luffing speed is 50 m/min, and the maximum lifting speed is 80 m/min. Fig. 9 shows the stress time history of top chord 638 during luffing. Fig. 10 shows the stress cycle amplitude distribution S_j (j = 1, 2, ..., n) of several dangerous components in the standard section obtained by the rain flow method. Without considering the influence of average stress fatigue strength, according to Miner cumulative damage criterion, the fatigue cumulative damage of the stress response time history is $D=\frac{n\bar{S}^m}{C}$.



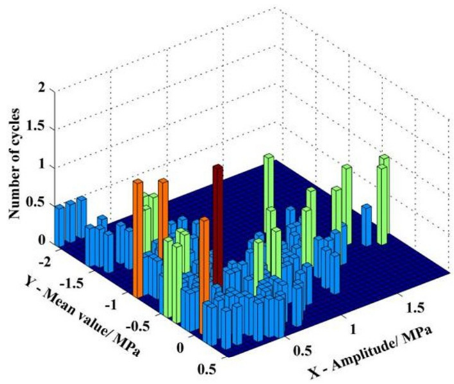
(a) Stress amplitude distribution of upper chord 638



(b) Stress amplitude distribution of lower chord 649



(c) Stress amplitude distribution of web member 680



(d) Stress amplitude distribution of inner rod 668

Fig. 10. Rainfall stress distribution of key components.

Table 1. Fatigue damage and fatigue life of main components.

Key members		Damage degree	Service life (year)
Translation motion	Upper chord 638	4.9096×10^{-8}	17.96
	Lower chord 649	3.8837×10^{-8}	22.76
	Web member 680	4.3061×10^{-8}	20.48
	Inner rod 668	2.6674×10^{-8}	33.06
Lifting motion	Upper chord 638	4.2976×10^{-8}	20.52
	Lower chord 649	3.6367×10^{-8}	25.25
	Web member 680	4.0234×10^{-8}	21.92
	Inner rod 668	2.4187×10^{-8}	36.46
Static load	Upper chord 638	5.7217×10^{-8}	15.41
	Lower chord 649	4.2024×10^{-8}	20.96
	Web member 680	4.6603×10^{-8}	18.92
	Inner rod 668	3.2782×10^{-8}	26.90

$\bar{S}^m = \frac{1}{n}\sum_{j=1}^{n} S_j^m$. n is the number of cycles and C is the fatigue parameter.

The main material of this type of tower crane is Q235, the design strength is 235 MPa, and the material fatigue parameter in code for design of steel structures [17] is $C = 4.26 \times 10^{13}$, $m = 3$. According to the stress amplitude statistics obtained by the rain flow method, the fatigue damage degree and fatigue life of key components in a complete movement process are shown in Table 1. According to the continuous working mode of 150 cycles per day for 28 days per month and 9 months per year.

It can be seen from Table 1 that under the worst working conditions, the maximum damage degree of the top chord 638 of the most dangerous component was 4.9096×10^{-8} during each luffing process. The service life was 17.96 years, and the damage degree of other components was less than this component. The damage degree of upper chord 638 was 4.2976×10^{-8} in each lifting process, with a service life of 20.52 years, meeting the requirements of tower crane safety regulations. From the calculation results, the damage degree of dangerous components during luffing was greater than during lifting, resulting in a service life less than that during lifting. Compared with the fatigue life under static load, considering the impact of moving load vibration, the fatigue life of several key components has been reduced to varying degrees, among which the life of Web member 680 has been reduced by 26.22 %.

5. Conclusions

In this paper, the tower crane was taken as the research object. The luffing trolley and cargo were equivalent to a concentrated force, and a moving load model was established. Using the transient dynamic analysis method, the vibration characteristics of the boom under luffing and lifting were analyzed. The fatigue life simulation of the key components of the boom was carried out based on the S-N curve. The following conclusions are:

1) The luffing speed can influence the vibration amplitude and frequency of the boom. The greater the moving speed, the greater the vibration amplitude of the arm end. As the trolley moves closer to the tower, the vibration amplitude at the boom end becomes significantly smaller when the moving speed is small, and the vibration frequency is high. When the moving speed is large, the vibration amplitude at the boom end does not change significantly.

2) Lifting speed has influence on the vibration amplitude and frequency. In the initial stage of lifting, the lifting speed has little influence on the change of boom end vibration deflection and amplitude. With the increase of lifting height, the boom end vibration deflection, vibration amplitude and vibration period increase with the increase of lifting speed.

3) During each amplitude change process of the luffing movement, the top chord 638 of the most dangerous component has the highest damage degree, with a service life of 17.96 years. The damage degree of other components is less than that of the component. The service life of the upper chord 638 is 20.52 years for each lifting process of the lifting movement. All meet the requirements of tower crane safety regulations. During luffing, the damage degree of dangerous components is greater than that during lifting, resulting in a service life less than that during lifting.

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