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Fatigue Life Assessment of Screw Blades in Screw Sand Washing Machine under Extreme Load

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Abstract The aim of this study is to estimate the fatigue life of the screw blades in the screw sand washing machine under the extreme load (i.e., the load caused by the full load condition). Firstly, the extreme load is taken into consideration in the fatigue life assessment of screw blades by means of the finite element analysis (FEA). Next, the *P-S-N* curve is fitted by the fully reversed rotating bending testing on standard specimens and theory deduction. Then, fatigue loadings are generated according to the maximum and minimum stresses of the root of screw blades under different thicknesses. Finally, the service life of screw blades is assessed based on the *P-S-N* curve, fatigue loadings, and Soderberg mean stress correction method. In particular, the effects of the surface finish factor, fatigue notch factor, and residual stress on the fatigue life of screw blades are considered. The results show that the stress concentration is at the root of screw blades; the screw blades with the thickness of 10 mm, whose service life is around 35 years, are the optimum in terms of the screw sand washing machine proposed in this study; the safety factor of screw blades should be 2, considering the influence of the fatigue load.

Keywords: *marine sand desalination, fatigue life, fatigue loading, finite element analysis (FEA), P-S-N curve, screw blade*

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1. Introduction

In recent years, with the excessive consumption of land-based sand sources and the swift increase of environmental pressure, marine sand is widely utilized for the fine aggregates of construction industries in an attempt to substitute land-based sand worldwide [1]. Nowadays, marine sand mining has already become the second most important marine mining source after oil source [1]. However, marine sand contains a great number of chloride ions, which will cause premature corrosion in steel bars embedded in concrete [2]. In architectural industries, the chloride ion content in the fine aggregate should be restricted strictly. Therefore, various mechanical wash-sand systems are used to get rid of chloride ions from marine sand in many countries, achieving the qualified marine sand used for the aggregates of construction industries [3]. Because of the excellent performance of desalting marine sand, the screw sand washing machine is the core element of different categories of mechanical wash-sand systems in the marine sand desalination field [4]. Consequently, this study takes the screw sand washing machine as the research object.

The sand washing process of the screw sand washing machine is shown in Fig. 1. To begin with, marine sand is conveyed into the screw sand washing machine via the belt conveyor or the rotating wheel sand washing machine [3,4]. Secondly, with the help of the motor and reducer, the screw structure has the ability to run at a constant rotational speed. More importantly, the functions of the screw structure are to agitate marine sand and water and transport marine sand. In most mechanical sand washing systems, ozone water is employed to eliminate chloride ions from marine sand [3]. In fact, ozone water is produced by a kind of special equipment, and then ozone water is poured into the screw sand washing machine via shower nozzles. Eventually, with the continuous operation of the screw sand washing machine, desalted marine sand is acquired.

Engineering practices have proven that the fatigue failure of screw blades is the main failure mode of the screw sand washing machine. To ensure the reliable operation of the screw sand washing machine and verify the design rationality of screw blades, the research on the fatigue life prediction of screw blades is executed in this study. In practice, the fatigue life estimation of screw blades could be beneficial to accurately control the fatigue strength of screw blades on the basis of actual engineering requirements. Additionally, it could provide a variety of theoretical guidance for determining the reasonable maintenance period of the screw sand washing machine. In conclusion, the investigation reported in this paper has certain theoretical and engineering values.

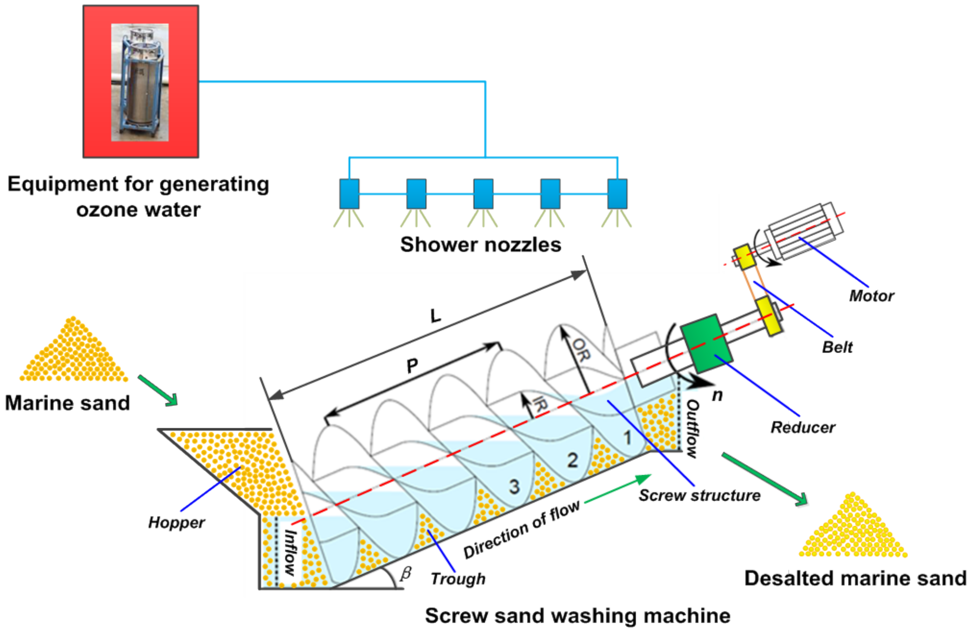

Figure 1. Sand washing process of screw sand washing machine

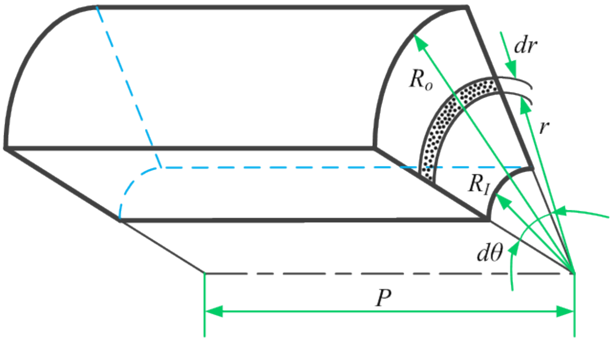

Figure 2. Marine sand in a pitch

2. Mechanical Model Establishment of Screw Blades

2.1. Simplification of Theoretical Model

A marine sand sector in a pitch used for calculating the axial forces acting on an individual screw blade surface is illustrated in Figure 2 [5]. In Figure 2, R_o is the radius of the screw structure in meters; R_I is the radius of the screw axis in meters; r is the radius of the screw blade in meters; P is the pitch length in meters; θ is the polar coordinate. To simplify the theoretical model, the following assumptions are made [6].

(1) The forces acting on individual surfaces are uniformly distributed.

(2) The trough is full of marine sand, and the distribution of marine sand is uniform in the trough. In addition, the mutual compression of marine sand is not considered. Actually, for different installation angles, the material filling factors of the screw sand washing machine are distinctly different [4]. For safety reasons, the influences of the extreme load on the fatigue life of screw blades are investigated in this study.

(3) The centrifugal force and cohesive force of marine sand are ignored.

(4) The flow of marine sand is relatively stable, and the axial direction is the positive direction.

(5) The friction coefficient between screw blades and marine sand is equal to that between the trough and marine sand.

2.2. Force Analysis of Screw Blades

When the screw sand washing machine desalts marine sand, the force condition of a screw blade is illustrated in Figure 3 [5,7]. In Figure 3, F is the resultant force acting on a screw blade in newtons; F_f is the force on the trailing side of a screw blade in newtons; F_d is the force on the driving side of a screw blade in newtons; F_1 is the component force of F in the normal direction of a screw blade in newtons; F_2 is the component force of F in the tangential direction of a screw blade in newtons; α_r is the screw blade helix angle at radius r in degrees; ϕ_f is the wall friction angle of marine sand on the screw blade surface in degrees; n is the rotational speed of the screw axis in radians per minute.

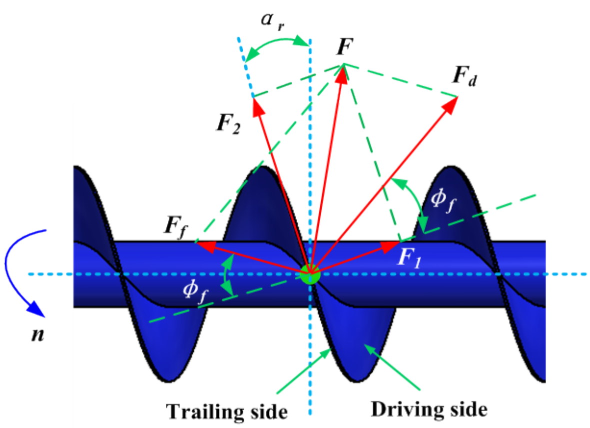

Figure 3. Forces on a screw blade

According to Figure 3, the extreme load acting on screw blades is given by

$$F_t = n_t F \tag{1}$$

where n_t is the number of pitches.

The resultant force acting on a screw blade is

$$F = \sqrt{F_1^2 + F_2^2} \tag{2}$$

but

$$F_1 = \frac{F_{da}\cos\phi_f}{\cos(\alpha_r + \phi_f)} - \frac{F_{fa}\cos\phi_f}{\cos(\phi_f - \alpha_r)} \tag{3}$$

$$F_2 = \frac{F_{da}\sin\phi_f}{\cos(\alpha_r + \phi_f)} + \frac{F_{fa}\sin\phi_f}{\cos(\phi_f - \alpha_r)} \tag{4}$$

$$\tan\alpha_r = \frac{P}{2\pi r} \tag{5}$$

$$\tan\phi_f = \mu_f \tag{6}$$

where F_{da} is the axial force acting on the driving side of a screw blade in newtons, F_{fa} is the axial resisting force acting on the trailing side of a screw blade in newtons, and μ_f is the wall friction coefficient between marine sand and screw blades.

For the convenience of calculations, non-dimensional parameters are defined as follows:

$$c_d = \frac{R_I}{R_o} \tag{7}$$

$$c_p = \frac{P}{D_o} \tag{8}$$

$$c_t = \frac{(D_o + 2c)}{D_o} \tag{9}$$

where D_o is the diameter of the screw structure in meters, and c is the clearance between the trough and the tip of screw blades in meters.

The axial force acting on the driving side of a screw blade can be expressed by [5,7]

$$F_{da} = \frac{\left\{ \begin{array}{l} 4\pi(R_o^2 - R_I^2)\sigma_0 \\ \left(\begin{array}{l} \frac{\pi}{2}\mu_e c_p \cos(\alpha_o + \phi_f) \\ + \frac{\pi c_d\,(2c_t - c_d)}{4}\sin\alpha_I \left[\exp\left(\frac{4\mu_w \lambda_s c_p}{2c_t - c_d} \right) - 1 \right] \\ + \lambda_s \left[\frac{\pi}{4}(1 - c_d^2) + \frac{\mu_f c_p}{2}(1 - c_d) \right] \\ + \frac{\pi}{4} c_t (2c_t - c_d)\cos(\alpha_o + \phi_f) \\ \left[\exp\left(\frac{4\mu_w \lambda_s c_p}{2c_t - c_d} \right) - 1 \right] \end{array} \right) \end{array} \right\}}{\pi(1 - c_d^2)} \tag{10}$$

but

$$\tan\alpha_o = \frac{P}{2\pi R_o} \tag{11}$$

$$\tan\alpha_I = \frac{P}{2\pi R_I} \tag{12}$$

where μ_e is the equivalent friction coefficient of marine sand, α_o is the screw blade helix angle at radius R_o in degrees, α_I is the screw blade helix angle at radius R_I in degrees, μ_w is the wall coefficient between marine sand and confining surface, λ_s is the stress ratio of marine sand sliding on the surface, and σ_0 is the stress exerted by marine sand in the hopper in pascals.

The axial resisting force acting on the trailing side of a screw blade is given by [5,7]

$$F_{fa} = \lambda_s \sigma_0 D_o^2 \left[\frac{\pi}{4}(1 - c_d^2) + \frac{\mu_f c_p}{2}(1 - c_d) \right]. \tag{13}$$

The stress exerted by marine sand in the hopper can be obtained by [5,7]

$$\sigma_0 = q_{f\sigma_1}\gamma B \tag{14}$$

where $q_{f\sigma_1}$ is the surcharge factor for the flow condition, γ is the specific weight of marine sand in newtons per cubic meter, and B is the opening width of the hopper outlet in meters.

A general expression of the stress ratio of marine sand sliding on the surface is [5]

$$\lambda_s = \frac{1}{1 + 2\mu_d^2 + 2\left[(1 + \mu_d^2)(\mu_d^2 - \mu_w^2)\right]^{1/2}} \tag{15}$$

but

$$\mu_d = \tan\delta \tag{16}$$

where μ_d is the tangent of the effective angle of the internal friction, and δ is the effective angle of the internal friction of marine sand in degrees.

The technical parameters of a typical screw sand washing machine provided by a manufacturing enterprise are listed as follows:

$R_o = 0.4065$ m, $R_I = 0.1235$ m, $P = 0.71$ m,
$n = 8.5$ r/min, $n_t = 10.5$, $c = 0.01$ m, $q_{f\sigma_1} = 1.4$,
$\alpha_r = 33^\circ$, $\delta = 30^\circ$, $\phi_f = 27.5^\circ$, $\mu_w = 0.3$, $\mu_e = 0.5$,
$\gamma = 1.6\times10^4$ N/m^3, $B = 0.65$ m.

3. Finite Element Analysis of Screw Blades

3.1. Establishment of Finite Element Model

The simulation model of screw blades is essential for the finite element analysis. To establish a high-quality simulation model, NURBS (Non-Uniform Rational B-Splines) surface modeling theory and 3-dimensional surface reverse technology are utilized. The full sized model of screw blades is shown in Figure 4.

In general, the fine mesh can ensure the result precision of the numerical simulation, but a great variety of computing time and the memory space will be required [8,9]. As a result, the selection of the mesh size should be based on the proper balance between the computational cost and the result precision. Considering the computational cost and the result precision synthetically, let the mesh size be 8 mm. The finite element model of screw blades is shown in Figure 5. The check function of the element quality in ANSYS Workbench is adopted to evaluate the finite element model shown in Figure 5, and assessment results show that the mesh quality of screw blades is excellent.

The material parameters of screw blades are listed in Table 1. Normally, the thicknesses of screw blades are in the range from 5 to 20 mm. To investigate the impacts of blade thicknesses on the fatigue life of screw blades under the extreme load, the finite element analysis of the screw blades with different thicknesses (7 levels of the blade thicknesses of 5, 7.5, 10, 12.5, 15, 17.5, and 20 mm) is carried out. Practically, the selection of these 7 levels is on the basis of the actual manufacturing level of screw blades.

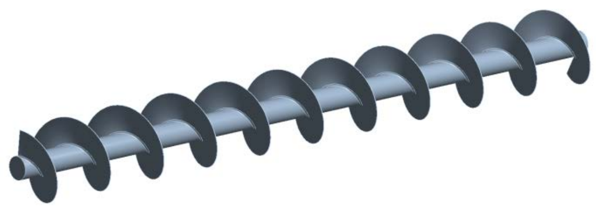

Figure 4. Geometric model of screw blades

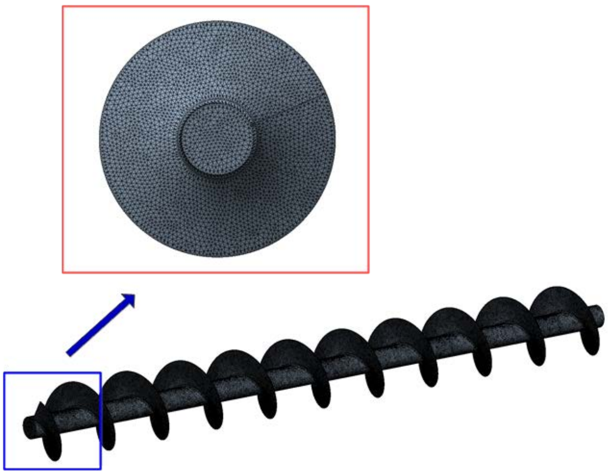

Figure 5. Finite element model of screw blades

Table 1. Material parameters of screw blades

Parameter	Value
Yield Stress, MPa	240
Tensile strength, MPa	440
Young's modulus, MPa	2×10^5
Density, kg/m^3	7850
Poisson's ratio	0.3

3.2. Results of Statics Analysis

Through substituting Eqs. (2) to (16) into Eq. (1), it is obtained that the value of the extreme load F_t is 214090 N, and the angle between the normal direction of screw blades and the extreme load F_t is approximately 34.4°. In ANSYS Workbench, the extreme load of 214090 N is applied on screw blades as the uniform loading, and fixed supports are applied at the two ends of the screw axis. Through the finite element simulation, the equivalent stress distributions of screw blades under different blade thicknesses are obtained, as shown in Figure 6. According to Figure 6, the maximum stresses of screw blades with different blade thicknesses are at the root of screw blades. Therefore, the root of screw blades is the position easy to generate fatigue failure. Particularly, it can be seen from Figure 6 (a) that when the blade thickness is 5 mm, the maximum equivalent stress of screw blades is 512.01 MPa, which is definitely more than the yield stress of 240 MPa. That is, the screw blades with the thickness of 5 mm cannot meet the demand of the static strength.

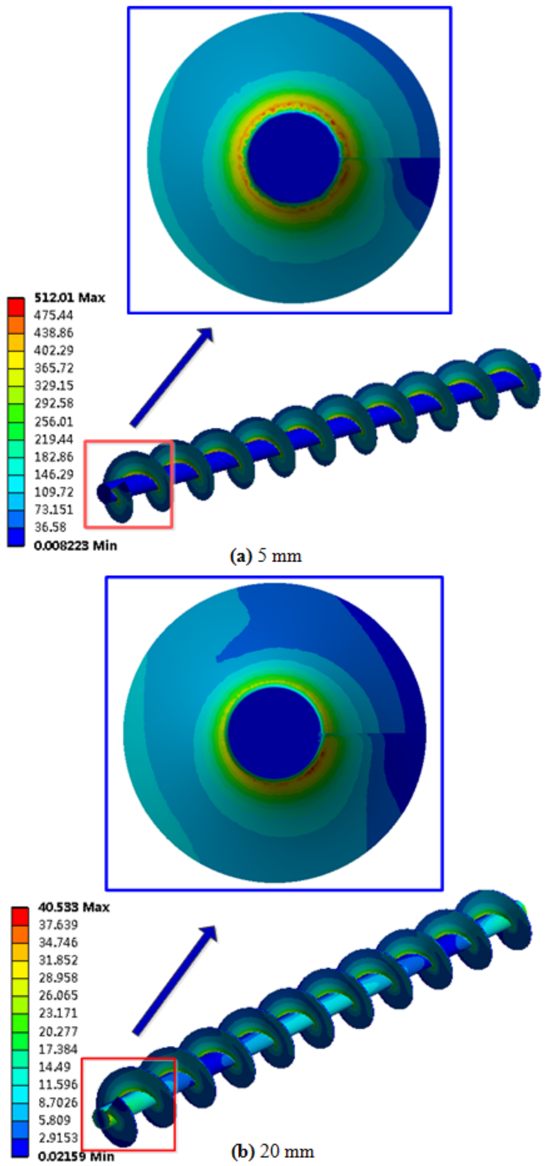

Figure 6. Equivalent stress distributions of screw blades under different blade thicknesses (MPa)

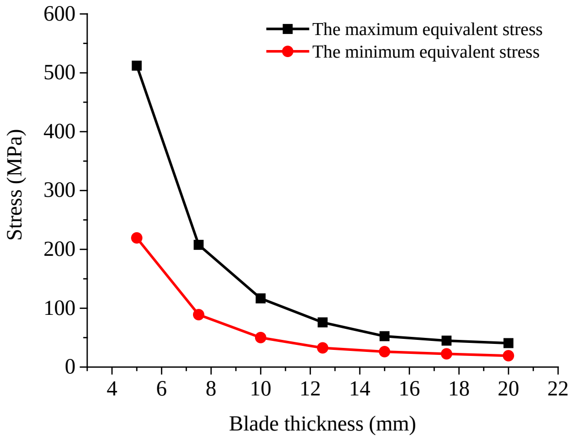

Figure 7. The maximum and minimum stresses of the root of screw blades under different thicknesses

As screw blades perform the periodically rotational motion, the stresses obtained by the static analysis are able to describe the dynamic change process of the stresses of a specific point on screw blades within a rotation period. Accordingly, the maximum and minimum stresses of the root of screw blades under different thicknesses can be employed to generate fatigue loadings. The maximum and minimum stresses of the root of screw blades are shown in Figure 7. As indicated in Figure 7, the maximum and minimum stresses of the root of screw blades decline swiftly with growing blade thicknesses. On top of this, as the screw blades with the thickness of 5 mm have already produced static failure, this sort of screw blades will not be taken into consideration in the fatigue life assessment.

4. Fatigue Life Assessment of Screw Blades

4.1. *P-S-N* Curve of Screw Blades

A large amount of *S-N* data has been historically generated on the basis of fully reversed rotating bending testing on standard specimens [10]. The *S-N* curve derived on the standard specimens under fully reversed bending loads can be constructed as a piecewise-continuous curve consisting of three distinct linear regions when plotted on log-log coordinates, as shown in Figure 8 [10]. In Figure 8, UTS is the ultimate tensile strength in megapascals, b is the slope of the *S-N* curve in the high-cycle fatigue region, Nc_1 is the transition life, Nfc is the numerical fatigue cutoff life, S_{1000} is the value of the stress at 1000 cycles in megapascals, and S_{be} is the value of the stress at the transition life Nc_1 in megapascals.

The slope of the *S-N* curve in the high-cycle fatigue region is expressed by [10]

$$b = \frac{\log S_{1000} - \log S_{be}}{\log 10^3 - \log Nc_1}. \tag{17}$$

According to [8], the parameters related to *S-N* curve are obtained as follows:

$$UTS = 440 \text{ MPa},\ Nc_1 = 1\times10^6,\ Nfc = 1\times10^{30},$$

$$S_{1000} = 396 \text{ MPa},\ S_{be} = 157.08 \text{ MPa}.$$

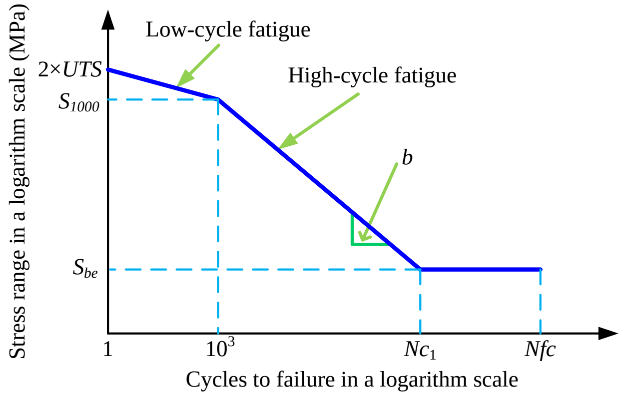

Figure 8. *S-N* curve

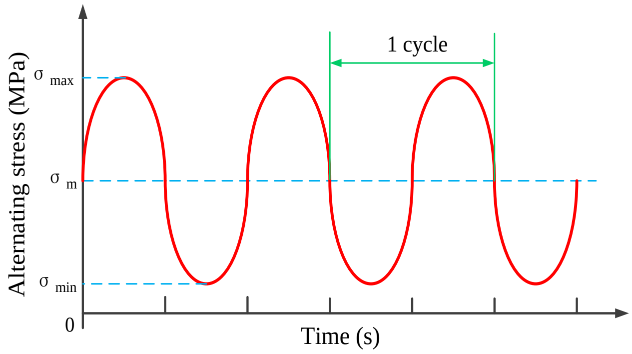

Figure 9. Fatigue loading

P-S-N curves are the expressions of fatigue life curves with given survivability [11]. Virtually, *P-S-N* curves are utilized to describe the randomness of fatigue property under different stress levels. In this study, the given survivability of 97.7% is selected according to ASTM standards.

In fatigue analysis, Soderberg, Goodman, Smith, and Gerber mean stress correction methods are the most common correction methods. Given that Soderberg mean stress correction method is more conservative than the other three mean stress correction methods, Soderberg mean stress correction method is used in this study [8].

4.2. Constant Amplitude Fatigue Loading

Actually, screw blades may fail at stress level below the static strength under alternating stresses. Provided that the loads do not trigger macroscopic cyclic plastic deformation, the failure mechanism is called stress-life or high cycle fatigue [12]. Undoubtedly, the fatigue loading is the precondition of the fatigue life assessment and fatigue test. The fatigue loading used to calculate the fatigue life of screw blades is shown in Figure 9. In Figure 9, $\sigma_{\max}$ is the maximum stress of the root of screw blades in Figure 7, $\sigma_{\min}$ is the minimum stress in Figure 7, and σ_m is the mean stress.

Evidently, the fatigue loading used in this study is the constant amplitude fatigue load. Although the constant amplitude fatigue load is uncommon in actual engineering, the fatigue behavior achieved under the constant amplitude fatigue load is the foundation of investigating the fatigue behaviors under various categories of fatigue loads [13,14,15]. That is, using the constant amplitude fatigue load has certain universality. In terms of most studies, the constant amplitude fatigue load is frequently utilized to predict the fatigue life of a wide range of products [13,14,15]. More importantly, the sand washing process is extremely sophisticated, so it is difficult to obtain the fatigue load acting on screw blades.

In particular, according to the rotational speed of the screw axis (8.5 r min^{-1}), the rotation period of screw blades, the reciprocal of the rotational speed of the screw axis, can be readily acquired.

4.3. Fatigue Life Calculation

Virtually, there are three main factors affecting the fatigue life of screw blades: the surface finish factor, fatigue notch factor, and residual stress [8,12]. In fact, the screw axis and screw blades are welded together. In terms of the current manufacturing level of this sort of screw blades, the fatigue notch factor is generally 1.5, considering the influence of welding; the surface finish factor is usually 0.8; the residual stress is in the range between 64 and 140 MPa. Particularly, tensile residual stress can have damaging effects on the fatigue resistance, whereas compressive residual stress can dramatically improve the fatigue behavior [10]. For safety reasons, let the residual stress of screw blades be 140 MPa.

The service life of the screw blades with different blade thicknesses is predicted on the basis of the *P-S-N* curve, fatigue loadings, and Soderberg mean stress correction method. Particularly, the effects of the surface finish factor, fatigue notch factor, and residual stress on the fatigue life of screw blades are taken into consideration. When the working time of the screw sand washing machine is 18 hours per day, the fatigue life of screw blades under different thicknesses is shown in Figure 10. Turning to Fig. 10, the fatigue life of screw blades is overwhelmingly sensitive to blade thicknesses. Contrary to the maximum and minimum stresses of the root of screw blades, the fatigue life of screw blades rockets with the rise in the blade thickness, ranging from 0.0153 to 1.9878×10^9 years between 7.5 and 20 mm. When the blade thickness is 10 mm, the fatigue life of screw blades under the extreme load is approximately 35 years, which meets the demand of the service life of at least 25 years in actual engineering.

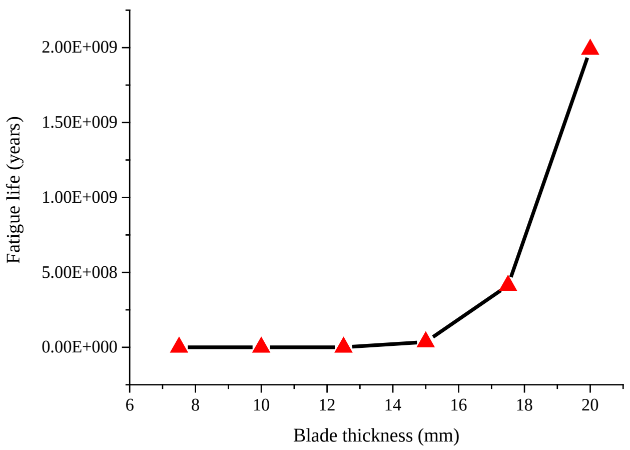

Figure 10. Fatigue life of screw blades under different thicknesses

As reported in [4], the thickness of optimized screw blades is 11 mm. If the blade thickness is 11 mm, the fatigue life of screw blades under the extreme load is around 1.1306×10^3 years. There is no doubt that the screw blades with the thickness of 11 mm can definitely meet the demand of the fatigue strength, showing the reasonableness of the optimized results in [4]. However, the screw blades with the thickness of 11 mm have excessive residual strength. Virtually, it is tough to define the fatigue strength of screw blades as the constraint condition in the optimization design of the screw sand washing machine, primarily because most of the design parameters are uncertain [4]. Unlike the diameters of the screw structure and screw axis, the blade thickness is not the sensitive factor in the optimization design, so the change of the blade thickness has far more slight effects on the optimization results obtained in [4]. In reality, it is better to determine the optimal blade thickness according to the fatigue life prediction rather than solely relying on optimization results.

Depending on the analysis above, the blade thickness of 10 mm is the most reasonable option for the screw sand washing machine reported in this study. In addition to that, it must be admitted that the fatigue life of screw blades achieved in this paper is relatively conservative.

Furthermore, when the blade thickness is 10 mm, the corresponding maximum stress in Fig. 7 is about 117 MPa, 1/2 of the yield stress 240 MPa. In other words, considering the influence of the fatigue load, a safety factor of 2 of common sense is justified in this study, which will absolutely provide valuable guidance for the engineers and designers of the screw sand washing machine.

5. Conclusions

Depending on the fatigue failure studies, the following conclusions are summarized:

(1) The stress concentration arises at the root of screw blades, and the blade thickness has significant effects on the fatigue life of screw blades.

(2) To ensure the fatigue reliability and to avoid the excessive design of screw blades, the blade thickness should be 10 mm in terms of the screw sand washing machine reported in this study.

(3) The safety factor of screw blades should be 2, considering the influence of the fatigue load.

(4) The proposed simulation approach could be employed by other researchers specializing in the fatigue life estimation of the screw conveyor, auger coal miner, screw extruder, etc.

(5) As a result of the complicated operation conditions of screw blades, a number of factors could not be fully taken into account, so a range of experiments will be carried out, and the simulation model will be revised based on experimental data in an attempt to achieve more precise simulation results in the future.

Acknowledgments

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