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Study of a fatigue damage model for flange considering the friction coefficient of the faying face

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Abstract. In the flange connections, part of the load is transmitted by the friction moment between the faying faces, thus the distribution of the input load in the connection is changing with the friction coefficient. Current models usually calculate the fatigue damage directly according to the load spectrum without considering the variation of the load due to friction coefficient. In the paper, a fatigue damage model for flange plate considering friction coefficient is proposed, and the influence of friction coefficient on the load distribution of the connection is deduced. Based on Palmgren-Miner linear damage theory, the algorithmic expressions of fatigue damage at varying friction coefficient are derived, which not only assesses the fatigue strength of flange under variable friction coefficient, but obtains the friction coefficient range which meets the fatigue strength requirement of flange connections.

1. Introduction

Flange connections are often applied to connect pipes or to transmit loads. In a transmission system, the working condition is usually dynamic [1], which can easily lead to fatigue failure. The flange connection and transmission mechanism of a wind power system is shown in Fig. 1. As a key component of transmission system, the fatigue life of flange is directly related to the performance of the whole system. After Fatigue failure, maintenance also causes a lot of manual labor and financial losses [2]. Thus, it is of great significance to assess the fatigue damage and check the fatigue strength of flanges during mechanical design [3, 4].

At present work, numerous fatigue damage calculation approaches were proposed based on different elements, among which time domain analysis approach is extensively used [5]. The fatigue damage is modeled mainly based on nominal stress method, local stress-strain method, energy method. Although there are various fatigue damage models for various systems, the fatigue calculation based on friction coefficient and considering the load distribution is still unprecedented. Thus, a fatigue damage model of flange considering friction coefficient is presented in this work. Based on a 3.0MW wind turbine, the fatigue damage and fatigue strength of the flange based on variable friction coefficient is calculated, and the influence of friction coefficient on the stress distribution and fatigue strength of the flange is obtained.

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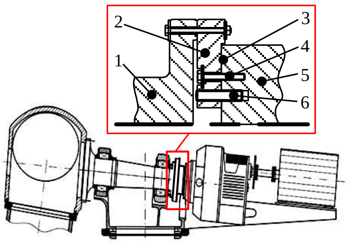

Figure 1. Inputting flange connection

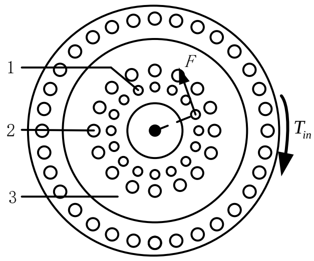

Figure 2. Force acting on flange

2. Theory model

2.1. Fatigue load considering friction coefficient

During operating, the faying face of the flange is subjected to frictional force and the pin holes of the flange are subjected to circumferential positive pressure, as shown in Fig. 2. And the wind turbine is subjected to random torsional loads during operation, which is shown in Fig. 3, so the torque transmitted by the flange is not equal to the total load. The magnitude is determined by the area of the faying surface and the friction factor. According to the Palmgren-Miner linear damage theory, the fatigue damage accumulates with the increase of stress cycle, and fatigue failure is led when the total fatigue damage D is greater than the allowable value $D_{\max}$, which is given according to particular criteria [6, 7]. Assume the faying face is a complete ring, then the maximum static friction moment acted on the flange is:

$$T_{f\max} = 2\pi pf \int_r^R x^2 dx \tag{1}$$

In Eq. (1), p is the unit positive pressure of the faying face, f is the friction coefficient, r and R are the inner circle radius and outer circle radius of the faying face, respectively.

According to the nature of static friction, when the input torque T is less than $T_{f\max}$, the friction moment is equal to the input torque, means that the load is transmitted by pins alone; otherwise the load is transmitted paratactic by frictional force and pin hole force. Based on Eq. (1), the pinhole force is calculated as:

$$\begin{cases} T_{pin} = 0 & T_{in} \le T_{f\max} \\ T_{pin} = T_{in} - T_{f\max} & T_{in} > T_{f\max} \end{cases} \tag{2}$$

In Eq. (2), T_{pin} is the torque transmitted by pins, T_{in} is the input torque. Eq. (2) implies that when the input torque increases from 0, there is a non-linear relationship between pin torque and the input torque. It is further deduced that the circumferential force acting on each pin hole of the flange is:

$$\begin{cases} F_p = 0 & T \le T_{f\max} \\ F_p = \dfrac{T_{pin}}{r_p \times n} & T > T_{f\max} \end{cases} \tag{3}$$

In Eq. (3), r_p is the radius of the pinholes' position on the flange, n is the quantity of pin hole.

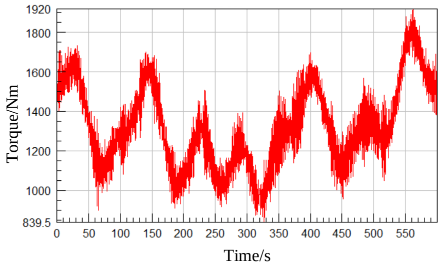

Figure 3. Random torsional load of wind turbine

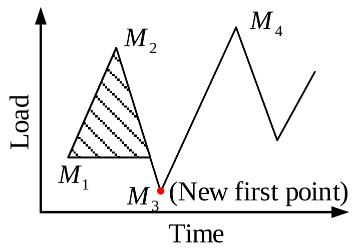

Figure 4. The RFC method

2.2. *Rainflow cycle counting*

As for the counting of load time histories, there are three main approaches to count the number of load cycles: level crossing approach [8], peak cycle approach [9] and rainflow cycle method (RFC) approach [10].Over the years, ASTM [11] has proposed many cycle counting methods to solve the fatigue damage of wind turbines and supporting structures, which usually based on the time history of stress or strain. Among these counting methods, the most widely applied is RFC. In 1968, Matsuishi et al. [12] first proposed the RFC based on the principle of raindrops dropping from eaves. Then this method was given the mathematical definition by Dowling [13]. After improvement by Rychlik [14], a simplified RFC for the periodic amplitude of rainflow in random loading process was obtained. In 1993, Frendahl [15] discussed RFC under fixed load and the linear fatigue damage accumulation theory, and then proved the accuracy of it. RFC has been improved and developed in different applications [16, 17] among which the three-point counting method, benefits from its high accuracy, has been widely used.

The three-point RFC method use three successive points to form a cycle. As shown in Fig. 4, The method is based on three successive stress points (T_1, T_2, T_3) that defined the successive ranges $\Delta T_1 = T_1 - T_2$ and $\Delta T_2 = T_2 - T_3$. If $\Delta T_1 \le \Delta T_2$, $dT_1 \le dT_2$, one cycle from T_1 to T_2 is extracted and the point T_3 will be regarded as the new first point, otherwise no cycle is counted. After counted by RFC method, the load will be translated to a symmetric cyclic one, which is shown in Fig. 5.

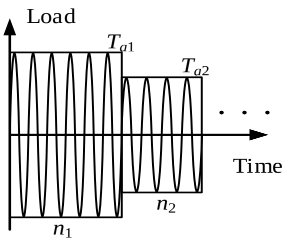

Figure 5. Symmetric cyclic loads

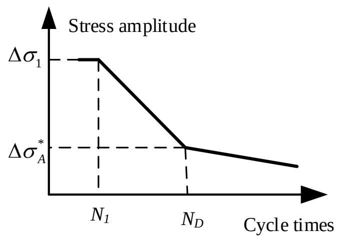

Figure 6. The S-N curve

2.3. *Linear fatigue damage theory*

At present, it is a common practice to apply the Palmgren-Miner linear damage theory to the calculation of the total fatigue damage. Hashin et al. [11] proposed the method of calculating fatigue damage of specimens with S-N curve and Palmgren-Miner linear damage theory and proved its reliability through a series of experiments. Subsequently, Leipholz [12] proposed the concept of modified S-N curve which takes more load factors into account, which is widely used in the present work. The modified S-N curve is shown as Fig. 6 and is expressed as follow:

$$\begin{cases} S_a = \Delta\sigma_A^* \left(\dfrac{N}{N_D}\right)^{b_2} & 0 < S_a \le \Delta\sigma_A^* \\ S_a = \Delta\sigma_A^* \left(\dfrac{N}{N_D}\right)^{b_1} & \Delta\sigma_A^* < S_a \le \Delta\sigma_1 \end{cases} \tag{4}$$

In Eq. (4), S_a is the stress amplitude in the loading history, $\Delta\sigma_A^*$ is the ordinate corresponding to S-N curve low cycle fatigue inflection point, N_D is the abscissae corresponding to S-N curve low cycle fatigue inflection point, b_1 and b_2 are the slopes of high-cycle fatigue curve and low-cycle fatigue curve on S-N curve, respectively.

$$D = \sum_{i=1}^{n} \frac{n_i}{N_i} \tag{5}$$

In Eq. (5), n_i is the number of load cycles in the loading history, N_i is the fatigue life. As for asymmetrical stress, the damage caused by mean stress should also be taken into account [13]. According to Goodman's model [14], it is assumed that the mean stress is linearly related to the stress amplitude as:

$$S_a = (1 - \frac{S_m}{S_u})\mathrm{S}_e \tag{6}$$

In Eq. (6), S_a is the stress amplitude, S_m is the mean stress, S_u is the ultimate tensile strength of the material, S_e is the fatigue limit of the material. With a combination of Eq. (4), Eq. (5) and Eq. (6), the total fatigue damage is:

$$D = \sum_{i=1}^{n} n_i / [\mathrm{N}_D (\frac{S_{ai} S_u}{(\mathrm{S}_u - \mathrm{S}_m)\Delta\sigma_A^*})^{1/\mathrm{b}_j}] \tag{7}$$

In Eq. (7) $j = 1$ or 2, depending on the S-N curve. In order to obtain the fatigue damage of flange based on loading history, the input loading history must be transformed into the stress history of the pin hole. In the present practical application, the procedure is: calculate the equivalent stress under unit torque first, then apply a linear relationship to obtain the stress history. The finite element method (FEM) is used to obtain the equivalent stress of flange under unit torque, which is deduced as:

$$\sigma_0 = \frac{1}{\sqrt{2}} [(\varepsilon_1 - \varepsilon_2)^2 + (\varepsilon_2 - \varepsilon_3)^2 + (\varepsilon_3 - \varepsilon_1)^2]^{\frac{1}{2}} \tag{8}$$

The stress amplitude and mean stress in the pin hole is deduced as follow:

$$\begin{cases} S_{ai} = T_{ai}\sigma_0 \\ S_{mi} = T_{mi}\sigma_0 \end{cases} T_i > T_{f\max} \tag{9}$$

With a combination of Eq. (5) to Eq. (9), the total fatigue damage of the flange is derived:

$$D^{'} = \sum_{i=1}^{n} n_i / [\mathrm{N}_D (\frac{T_{ai}\sigma_0 S_u}{(\mathrm{S}_u - \mathrm{S}_m)\Delta\sigma_A^*})^{1/\mathrm{b}_j}] \tag{10}$$

Thus the factor of stress (FOS) of the flange, through which the fatigue strength of flange can be evaluated, can be deduced as:

$$FOS = \frac{1}{D^{'}} \tag{11}$$

3. Simulation

3.1. Static Analysis of the flange

The detailed parameters of the input flange are shown in Table 1. As shown in Fig. 7, the model has a total of 774960 elements and 1122603 nodes. According to the actual force of the flange, the full constraint is applied to the outer ring, and the unit torque is applied in the pinholes.

Table 1. Analysis parameters

Parameters	E(MPa)	μ	$\Delta\sigma_A^*$ (MPa)	N_D (MPa)	P(N/mm)	r(mm)	R(mm)
value	20600	0.3	257.89	1002000	40	175	425

According to Eq. (1), the maximum static friction moment is 2674.6 kNm. The equivalent stress under the unit load (1kNm), calculated using by ANSYS, is shown in Fig. 8.

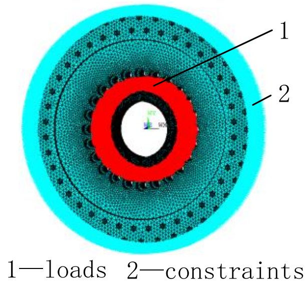

Figure 7. Finite element model

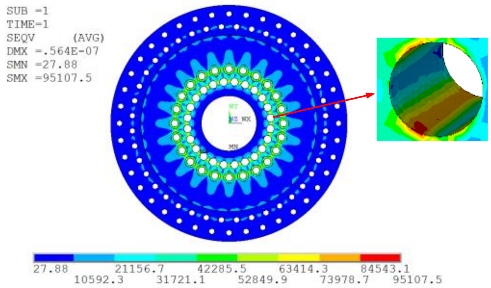

Figure 8. Equivalent stress of flange

It is shown in Fig. 8 that the maximum equivalent stress of the flange is 95107.5 MPa.

3.2. Transformation of the fatigue load

In the 3MW wind turbine, the unit positive pressure of the faying face is 40N/mm, radius of outer circle of contact surface is 425mm, radius of the inner circle of contact surface is 175mm. When the friction coefficient is 0, 0.1, 0.2, 0.3, 0.4 and 0.5 respectively, relationship between the pin torque and the input torque is shown in Fig. 9.

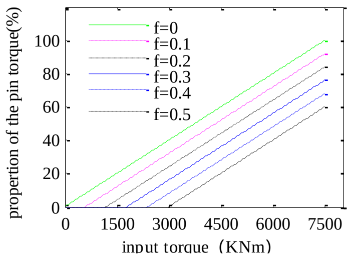

Figure 9. Loads based on different friction coefficient

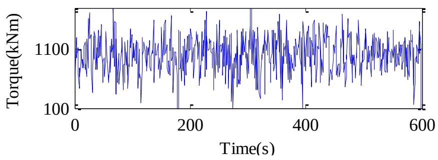

Figure 10. Pinhole load of DLC1.2-18-0($f=0.4$)

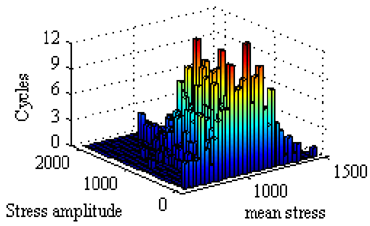

Figure 11. Rainflow matrix of load DLC1.2-18-0 ($f=0.4$)

It is shown in Fig. 9 that when the friction coefficient is constant, the pin torque increases with the increase of the input torque; and when the input torque is constant, the pin load decreases with the increase of the friction coefficient. Particularly, the pin torque equals to the input torque when the friction coefficient is 0, which means that the input torque is entirely transmitted by the pin joint.

It can be seen is Fig. 10 that the fatigue load of the working condition DLC1.2-18-0 is translated to the pinhole load using a three-point, which is then counted by RFC method as the Fig. 11 shows..

3.3. Fatigue life Assessment of the flange

Combined Eq. (12), Eq. (13) and Eq. (14), and FOS of flange under different friction coefficient are calculated. The results are shown in Table 2 and the trend of and FOS is shown in Figure 12.

Table 2. Result of D and FOS

f	*D*	*FOS*
0	2.963	0.338
0.1	1.827	0.547
0.2	1.218	0.821
0.3	0.304	3.292
0.4	0.110	9.113
0.5	0.074	13.48

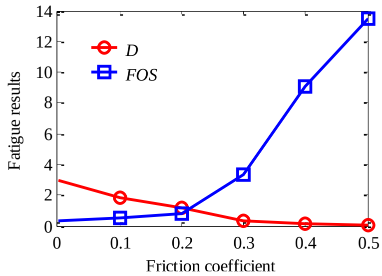

Fig. 12 The trend of D and FOS

As shown in Table (2), fatigue damage is greater than 1 without considering friction moment (f = 0), while it decreases with the increase of friction coefficient. It can be obtained by interpolation approach that the friction coefficient is 0.224 when fatigue damage reaches the value of 1. Therefore, to avoid fatigue failure, the friction coefficient must be greater than 0.224.

4. Conclusion

The total fatigue damage of flange is negatively correlated with the friction coefficient. That is, the larger the friction coefficient, the smaller the fatigue damage and the higher the fatigue strength of flange. Therefore, the fatigue strength of flange can be improved by increasing the friction coefficient.

Through the fatigue analysis, the very friction coefficient satisfying fatigue strength can be obtained, thus providing a theoretical basis for the mechanical design of flange connections.

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References

[1] Marino E , Giusti A , Manuel L . Offshore wind turbine fatigue loads: The influence of alternative wave modeling for different turbulent and mean winds[J]. Renewable Energy, 2017, 102:157-169.

[2] Fuglsang P, Thomsen K. Cost optimization of wind turbine for large scale offshore wind farms, technical report. Risø National Laboratory; 1998.

[3] Yeter B, Garbatov Y, Soares C G.. Fatigue damage assessment of fixed offshore wind turbine tripod support structures[J]. Engineering Structures, 2015, 101:518-528.

[4] Christoph W. Kensche. Fatigue of composites for wind turbines[J]. International Journal of Fatigue,2006,28(10).

[5] Yeter B, Garbatov Y, Soares C G. Evaluation of fatigue damage model predictions for fixed offshore wind turbine support structures [J]. International Journal of Fatigue, 2016, 87: S0142112316000086.

[6] Socie D. Multiaxial Fatigue Damage Models [J]. Key Engineering Materials, 2006, 324-325(4):747-750.

[7] Dong W , Moan T , Gao Z . Long-term fatigue analysis of multi-planar tubular joints for jacket-type offshore wind turbine in time domain [J]. Engineering Structures, 2011, 33(6): 2002-2014.
[8] Rice R , editor, Leis B , et al. Fatigue design handbook [M].
[9] Veers, P.S, Winterstein, et al. Variable amplitude load models for fatigue damage and crack growth [J]. Astm Special Technical Publication, 1987(1006):26.
[10] Dowling N E . Fatigue Failure Predictions for Complicated Stress-Strain Histories [J]. Journal of Materials, 1971, 7(1): 71.
[11] Standard practices for cycle counting in fatigue analysis [J]. Astm, 1985.
[12] M. Matsuishi, T. EndoFatigue of metals subjected to varying stress. Jpn Soc Mech Eng (1968), pp. 37-40
[13] Downing S D, Socie D F. Simple rainflow counting algorithms [J]. International Journal of Fatigue, 1982, 4(1): 31-40.
[14] Rychlik I. A new definition of the rainflow cycle counting method [J]. International Journal of Fatigue, 1987, 9(2): 119-121.
[15] Frendahl M , Rychlik I . Rainflow analysis: Markov method [J]. International Journal of Fatigue, 1993, 15(4): 265-272.
[16] Amzallag C , Gerey J P , Robert J L , et al. Standardization of the rainflow counting method for fatigue analysis [J]. International Journal of Fatigue, 1994, 16(4): 287-293.
[17] Glinka G , Kam J C P . Rainflow counting algorithm for very long stress histories [J]. International Journal of Fatigue, 1987, 9(4): 223-228.