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Research on Safety Evaluation Calculation of Flange Bolt Connection Based on VDI2230 Standard

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Abstract. Flange bolt connection is one of the most important bolt connection methods at present. In this paper, the bolt connection between the shell and the back flange of a certain type of equipment was evaluated and calculated, the weak area of the structure is mainly the bolt joint, so bolt safety design and check is a very key part of the structure check, the finite element method and "VDL2230-2003 Bolt strength checking standard" were used to calculate, the boundary conditions of bolt load were obtained by finite element method, and the application of VDI 2230 in flange bolt connection analysis and bolt safety evaluation were systematically described. The results show that M20 bolt can meet the requirements of safe use, while M16 and M18 bolt cannot.

Keywords*:* flange bolt connection; finite element calculation; VDI2230; pre-tightening force; safety factor

1. Introduction

Bolted connection is the most common fastening connection in current industrial development [1]. High-strength bolts have the advantages of simple construction, high connection reliability, good mechanical performance, fatigue resistance, self-locking, disassembly and non-loosening under dynamic load, etc., and are increasingly widely used as a promising connection method [2]. Flange bolt connection is an important connection form of bolt connection, widely used in chemical industry, oil refining, nuclear power and other fields, with the advantages of convenient disassembly and maintenance [3]. However, in the process of bolt installation, improper handling will lead to sliding wire, twisting, yield, and even pulling, etc. Bolt breakage in the process of equipment operation will cause equipment damage, or even harm to personal safety [4,5]. Factors affecting the safety of flange bolt connection include flange strength, strength of optional bolts, bolt installation tools and methods, etc., among which insufficient bolt preload and improper installation method are the key factors [6,7]. Properly pretightening flange bolt connection can improve its sealing ability, anti-loosening ability, connection security, increase the tightness and rigidity of the connection, and prolong the service life of fasteners. Insufficient pretightening force will result in leakage of sealing surface and failure of flange seal under operating conditions, but too high pretightening force will also cause flange deformation, gasket collapse, bolt material yield and other problems, which will also lead to failure of flange seal [8].

At present, most scholars mainly use finite element software analysis and scientific calculation to design and check the reliability of bolts. Theoretical calculation is generally based on the strength theory of "Mechanics of Materials" and "VDL2230-2003 Bolt strength checking standard" (referred to as "VDI standard") for checking. In the aspect of finite element research, some scholars use isolated extraction of a pair of bolts for analysis; Some scholars simplified the bolt as BEAM unit or directly applied the load to the bolt hole [9]. In the past 30 years, both numerical and experimental models have proved the validity of VDI 2230 analysis method [10.11], and this standard provides design engineers with the range of safe preload [12,13].

In this paper, the force of bolts in pipe flange connection and the calculation method of pretightening force were analyzed, which can provide reference for determining the pretightening force and torque of bolts in the process of industrial pipe flange installation. The calculation object of this paper was the safety evaluation of bolt connection between shell and back head flange of a

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certain type of warship equipment. As shown in Fig. 1, the bolt assembly was composed of 24 titanium alloy inner hexagon bolts. Finite element method was adopted to calculate the boundary conditions of bolts under load. M16, M18 and M20 bolts were planned to be used. Based on the first strength theory, the third strength theory and the "VDI Standard", the bolt connection parts in Fig. 1 were analyzed and compared and verified by experiments.

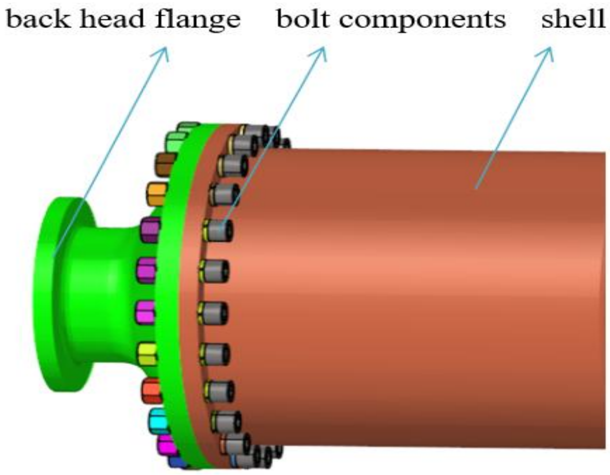

Fig. 1 Three-dimensional structure diagram

2. Finite element analysis

2.1 Finite element model

Finite element analysis was used to obtain the working load of bolts. The equipment shell and the back head were the rotating body, the rotating center was defined as the X axis, and the axial direction of the bolt was consistent with the X axis. After the shell and head with flange connection, a total of 24 hex bolt, after equipment shell and head of material for titanium alloy TB4, bolt material for titanium alloy TB2 (solid solution state), as shown in TABLE I, after the shell and head were used to simulate the three-dimensional entity unit, bolt connection was simulated with the BEAM unit and RBE2 unit combination. The flanges were set in contact with each other. The main structure was divided by solid elements, and the bolt connection was simulated by BEAM element and RBE2 element combination. The whole model includes 437,638 nodes and 327,808 units, and its finite element model was shown in Fig. 2.

Table 1 Material properties used in finite element analysis

Project	TB4	TB2
density [t/mm3]	4.5E-9	4.5E-9
Poisson's ratio	0.34	0.34
Elastic model [MPa]	108000	108000
yield strength [MPa]	825	838
Tensile strength [MPa]	895	858
Shear strength [MPa]	/	600
Surface crushing strength [MPa]	1340	/

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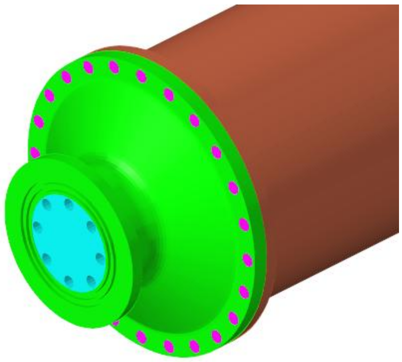

Fig. 2 Meshing diagram of equipment shell and back head

2.2 Analysis of working condition

The end face of the device shell was taken as a fixed constraint area to limit its six directions of freedom. In non-working state, the equipment shell has no internal and external pressure difference; In the working state, the internal pressure produces instantaneous pressure, and the pressure difference between the internal and external pressure was 12MPa. The pressure direction was from the inside to the outside and perpendicular to the surface. For the sake of conservatism, the transient load was simplified as quasi-static load, and the transient effect was not considered.

2.3 Calculation results

The established finite element analysis model was solved to obtain the displacement field, stress field and bolt load of the structure. The maximum displacement of the equipment structure was 2.25mm, and the maximum equivalent stress was 416MPa. The displacement distribution cloud diagram is shown in Fig. 3. The maximum axial load of bolts was 65592N, the maximum shear force was 1181N, and the maximum bending load was 248280N. mm. The stress distribution cloud diagram is shown in Fig. 4.

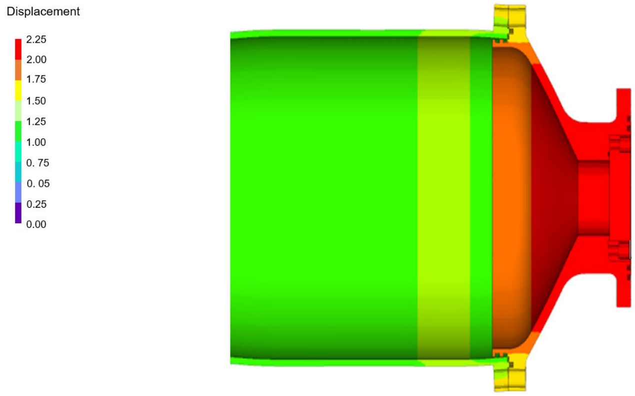

Fig. 3 Displacement distribution cloud diagram (deformation magnification 5 times)

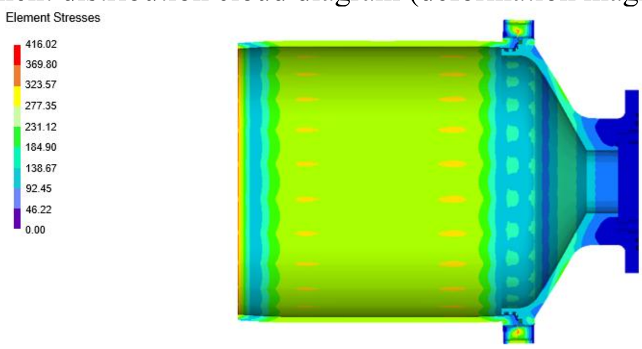

Fig. 4 Equivalent stress distribution cloud diagram

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3. Bolt strength safety check

3.1 The first strength theory and the third strength theory bolt strength check

Table 2Boundary condition parameters calculated by strength theory

Parametric	Numeral
Average diameter of sealing ring (Dcp)	392.3mm
The housing bears maximum working pressure (Pmax)	12MPa
width of sealing ring (b)	5.3mm
Sealing ring coefficient (m)	2
The number of stud (n)	24

Firstly, the first strength theory is used for calculation. This theory means that material fracture is caused by the maximum tensile stress, that is, when the maximum tensile stress reaches a certain limit value, the material fracture occurs, when the safety factor fs>1, the bolt is considered safe under tension, where ds is equivalent diameter of stud stress section, and the calculation process is as follows.

The maximum tension on the bolt：

$$Q_{max} = \frac{\pi}{4} D_{cp}{}^{2} P_{max} + \pi D_{cp} b m P_{max} \tag{1}$$

Maximum normal stress of bolt：

$$\sigma_{max} = \frac{Q_{max}}{\frac{\pi}{4} d_s{}^{2} n} \tag{2}$$

The first strength theoretical safety factor：

$$f_s = \frac{\sigma_{allowable}}{\sigma_{max}} \tag{3}$$

According to the third strength theory, the thread on the curved scissors equivalent stress and safety coefficient is calculated, and the theoretical hypothesis, the maximum shear stress is the cause of material yield, namely no matter under what kind of stress state, as long as the material somewhere within the maximum shear stress of τmax reaches the unidirectional tensile yield shear stress limit value, the material is there significant plastic deformation or yield, when the safety factor fs>1, the bolt is considered to be safe when subjected to shear force, where Z is the number of threads subjected to stress, l is the length of threads subjected to stress, t is the pitch and d0 is the equivalent diameter of bolt stress section. The calculation process is as follows.

Maximum bending stress of thread：

$$\sigma_{smax} = 0.88 \frac{Q_{max}}{n d_0 Z t} \tag{4}$$

Maximum shear stress of thread：

$$\tau_{smax} = 0.37 \frac{Q_{max}}{n d_0 Z t} \tag{5}$$

Equivalent stress in bending and shear：

$$\sigma_{max} = \sqrt{\sigma_{smax}{}^{2} + 4\tau_{smax}{}^{2}} \tag{6}$$

The third strength theoretical safety factor：

$$f_s = \frac{\sigma_{allowable}}{\sigma_{max}} \tag{7}$$

The calculation results of the first strength theory and the third strength theory are shown in TABLE III. The safety coefficients of bolts of the three specifications were all greater than 1, indicating that the bolt connections of the three specifications were safe.

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Table 3 Calculation results of bolts of three specifications

Bolt Specification (mm)	Equivalent Diameter of Stress Section（mm）	Maximum Stress of Bolt Sectionσmax（MPa）	Yield Limit Index σallowable（MPa）	Safety Factor
M16×70	14.268	The first strength theory (σmax)：435.3	825.0	1.90
		The third strength theoretical (σmax)：361.5	825.0	2.28
M18×70	15.835	The first strength theory (σmax)：353.4	825.0	2.33
		The third strength theoretical (σmax)：299	825.0	2.76
M20×75	17.835	The first strength theory (σmax)：278.6	825.0	2.96
		The third strength theoretical (σmax)：231.3	825.0	3.53

3.2 "VDI2230 standard" bolt strength check

During bolt safety check, the bolt bearing the maximum axial load, maximum shear and maximum bending load among the 24 bolts was selected for strength safety check. The bolt connection type in this case adopts SV3 in Fig. 22 of VDI2230 Standard. Since bolt specifications have been determined, steps R1-R13 are used to carry out bolt strength safety assessment calculation. Main steps of standard bolt strength checking calculation [14] :

R0：Determine the nominal diameter (d)

Firstly, the maximum axial load (FAmax) and the maximum transverse load (FQmax) were calculated. Select bolts according to Table A7 in VDI2230 standard.

R1：Determine tightening coefficient (αA)

Tightening coefficient αA considering the dispersion of assembly preloading between FM min and FM max, it should be determined according to the friction coefficient level in Table A8 in VDI2230 Standard. In this case, use torque wrench to tighten the bolt and select αA=1.8.

$$\alpha_A = \frac{F_{M\ max}}{F_{M\ min}} \tag{8}$$

R2：Determine the minimum clamping load required (FKerf)

Table 4 Calculates FKQ boundary condition parameters

Parametric	Numeral
screw pitch (P)	2.5mm
Maximum shear (FQmax)	1180N
Refers to the amount of interfacial transfer force (FQ) in the inner part of the bolt that may slip/shear (qF)	1
Friction coefficient of joint surface (uT)	0.2
Torque around the axis of the bolt (MY)	0N.mm
Refers to the amount of interface transfer torque (MY) of the inner part that may slide (qM)	1
When MY is applied, the friction radius of the clamping part (ra)	0mm

1）Calculate the minimum clamping load that transmits lateral load and/or torque through friction clamping (FKQ)，

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$$F_{KQ} = \frac{F_{Qmax}}{q_F \times \mu_{T\,min}} + \frac{M_{Ymax}}{q_M \times r_a \times \mu_{T\,min}} = 5.9\ \text{kN} \quad (9)$$

2）Calculate the minimum clamping load (FKP) to ensure the sealing function. In this case, there is no dielectric seal：

$$F_{KP} = 0.0\ \text{N} \quad (10)$$

3）Calculate the minimum clamping load (FKA) in the loosening limit to prevent the ultimate load when loosening：

$$F_{KA} = 10998.0\ \text{N} \quad (11)$$

so：

$$F_{Kerf} \geqq \max（F_{KQ}；F_{KP} + F_{KA}） = 10998.0\ \text{N} \quad (12)$$

R3：Determined the elastic springback (δS，δP) and determined load coefficient (Φ)

Amount of bolt elastic rebound：

$$\delta_S = \delta_{SK} + \delta_1 + \dots + \delta_{Gew} + \delta_{GM} = \frac{l_{SK}}{E_M \times A_N} + \frac{l_1}{E_S \times A_1} + \frac{l_{Gew}}{E_M \times A_{d3}} + \frac{l_M}{E_M \times A_N} + \frac{l_G}{E_S \times A_{d3}} = 2.7112E^{-006}\ \text{mm/N} \quad (13)$$

Table 5 Calculates the boundary conditions of springback

Parametric	Numeral
Mean supporting surface diameter (dWm)	28.29mm
Clamp part of aperture (dh)	21.5mm
Bolted connection type connection coefficient (w)	1
The clamping length (lK)	49mm
Screw Angle of bolt thread φ，(tanφ)	0.38
Interface matrix replaces outer diameter (DA)	33mm
Limit diameter (DA,Gr)	46.95mm
Elastic modulus of connector (EP)	108×103 MPa
Load introduction factor (n)	0.44

Due to the DA,Gr >DA ，

Plate springback amount：

$$\delta_P = \frac{\frac{2}{w \times d_h \times \tan\varphi} \ln\left[\frac{(d_W+d_h)(D_A-d_h)}{(d_W-d_h)(D_A+d_h)}\right] + \frac{4}{{D_A}^2-{d_h}^2}\left[l_K - \frac{(D_A-d_W)}{w \times \tan\varphi}\right]}{\pi \times E_P} = 9.3392E^{-007}\ \text{mm/N} \quad (14)$$

Load factor for eccentric clamping and loading：

$$\Phi^*_{en} = n \times \frac{\delta^{**}_P}{\delta_S + \delta^{**}_P} = 0.113 \quad (15)$$

R4：Calculate the change of preloading (FZ、ΔF´Vth)

Refer to Table 5 in VDI2230 Standard. The embedding quantity fz is 13.5 μm.

Loss of bolt preloading caused by embedding：

$$F_Z = \frac{f_Z}{\delta_S + \delta_P} = 3703.6\ \text{N} \quad (16)$$

Preloading changes occur due to temperature changes：

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$$\Delta F'_{Vth} = 0.0\ N \tag{17}$$

R5：Determine minimum assembly preloading (FM min)

Minimum assembly preloading：

$$F_{M\,min} = F_{kerf} + (1 - \Phi^{*}_{en}) \times F_{A\,max} + F_Z + \Delta F'_{Vth} = 72697.5\ N \tag{18}$$

R6：Determine the maximum assembly preload (FM max)

Maximum assembly preload：

$$F_{M\,max} = \alpha_A \times F_{M\,min} = 130855.4\ N \tag{19}$$

R7： Determine the assembly stress (σred,M) and allowable bolt preload (FM zul)

assembly stress：

$$\sigma_{red,M} = \nu \times R_{p0.2min} = 754.2\ Mpa \tag{20}$$

Allowable assembly preloading：

$$F_{Mzul} = A_0 \times \frac{\nu \times R_{p0.2min}}{\sqrt{1+3\times[\frac{3}{2}\times\frac{d_2}{d_0}\times(\frac{P}{\pi\times d_2}+1.155\mu_{Gmin})]^2}} = 148280.9\ N \tag{21}$$

Safety factor of bolt pretightening force：

$$S_M = \frac{F_{Mzul}}{F_{Mmax}} = 1.13 \tag{22}$$

The safety factor of bolt preload is used to evaluate whether the preload applied by bolt connection meets the assembly requirements, need to meet $S_M = \frac{F_{M\,zul}}{F_{M\,max}} \geqq 1.0$。

R8：Determination of working stress (σred,B)

Working state bolt load：

$$F_{Smax} = F_{Mzul} + \Phi^{*}_{en} \times F_{A\,max} = 155692.8\ N \tag{23}$$

Maximum tensile stress：

$$\sigma_{zmax} = \frac{F_{Smax}}{A_0} = 636.8\ N/mm^2 \tag{24}$$

Proportion of tightening torque acting on thread MG= 373718.2 N.mm，Bolt cross section polar resistance moment WP= 1080.4 mm3，

Maximum torsional stress：

$$\tau_{max} = \frac{M_G}{W_P} = 345.9\ N/mm^2 \tag{25}$$

Recommended in the standard kτ=0.5；Working state compares stress：

$$\sigma_{red,B} = \sqrt{\sigma_{z\,max}{}^2 + 3 \times (k_\tau \times \tau_{max})^2} = 703.7\ Mpa \tag{26}$$

Safety factor of bolt yield：

$$S_F = \frac{\sigma_{red,B}}{R_{p0.2min}} = 1.19 \tag{27}$$

The yield safety factor of bolt is used to evaluate whether the yield failure of bolt connection will occur in the working process，Need to meet $S_F = \frac{\sigma_{red,B}}{R_{p0.2min}} \geqq 1.0$。

R9：Determine the alternating stresses (σa、σab)

Continuous alternating stresses acting on bolts：

$$\sigma_a = \frac{F_{SAo} - F_{SAu}}{2A_S} = 15.5\ MPa \tag{28}$$

Continuous alternating stresses on bolts during eccentric clamping and loading：

$$\sigma_{ab} = \frac{\sigma_{SAo} - \sigma_{SAu}}{2} = 23.2\ MPa \tag{29}$$

R10：Determination surface pressure (pmax)

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Table 6 Calculates surface pressure boundary conditions

Parametric	Numeral
Minimum supporting area of bolt head or nut (Apmin)	272.6 mm2
Clamping material limits surface pressure (pG)	855 MPa
Material shear strength (τB)	600 MPa

Maximum surface pressure in assembly condition：

$$P_{Mmax} = \frac{F_{M\,zul}}{A_{p\,min}} = 544.0\ \text{MPa} \tag{30}$$

Operating condition maximum surface pressure：

$$P_{Bmax} = \frac{F_{V\,max}+F_{SA\,max}-\Delta F_{Vth}}{A_{p\,min}} = 558.3\ \text{MPa} \tag{31}$$

Minimum surface strength safety factor in assembly：

$$S_{PM} = \frac{p_G}{P_{Mmax}} = 1.57 \tag{32}$$

Minimum surface strength safety factor in operating condition：

$$S_{PB} = \frac{p_G}{P_{Bmax}} = 1.53 \tag{33}$$

The safety factor of surface strength is used to evaluate whether the surface collapse of clamping parts will happen in the process of bolt connection，need to meet $S_P = \frac{p_G}{P_{M/Bmax}} \geqq 1.0$。

R11：Determine the minimum spin length (meff min)

The maximum bolt pulling force must be less than the critical tripping force of internal thread or bolt thread $F_{mS} \leqq F_{mGM}$; the maximum thread length meff for M4 to M39 standard threads can be approximated from Fig. 36 in VDI2230 standard for critical internal thread standard conditions. This example uses a standardized nut corresponding to the bolt strength, so this step was omitted.

R12：Determine safety margin against skid (SG) and shear stress (τQmax)

Minimum residual clamping load：

$$F_{KR\,min} = \frac{F_{M\,zul}}{\alpha_A} - (1-\Phi_{en}^{*})\times F_{A\,max} - F_Z - \Delta F_{Vth} = 20678.9\ \text{N} \tag{34}$$

Safety factor of joint surface against unilateral opening：

$$S_K = \frac{F_{KR\,min}}{F_{KP}+F_{KA}} = 1.88 \tag{35}$$

Safety factor against unilateral opening of joint surface whether unilateral opening will occur in the process of bolted connection，need to meet $S_K = \frac{F_{KR\,min}}{F_{KP}+F_{KA}} \geqq 1.0$。

The clamping load required to deliver the lateral load：

$$F_{KQ\,erf} = \frac{F_{Qmax}}{q_F \times \mu_{T\,min}} + \frac{M_{Ymax}}{q_M \times r_a \times \mu_{T\,min}} = 5905.0\ \text{N} \tag{36}$$

Safety margin against skid：

$$S_G = \frac{F_{KR\,min}}{F_{KQ\,erf}} = 3.50 \tag{37}$$

The anti-sliding safety factor is used to evaluate whether contact surface slip will occur in the process of bolt connection，need to meet $S_G = \frac{F_{KR\,min}}{F_{KQ\,erf}} \geqq 1.2$。

Shear area of bolt during transverse loading Aτ=244.8mm2，

$$\tau_{Qmax} = \frac{F_{Qmax}}{A_\tau} = 4.8\ \text{MPa} \tag{38}$$

Shear safety factor：

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$$S_A = \frac{\tau_B}{\tau_{Qmax}} = 125 \quad (39)$$

Shear safety factor indicates whether bolt shear failure occurs during bolt connection， need to meet $S_A = \frac{\tau_B}{\tau_{Qmax}} \geqq 1.1$。

R13：Determine tightening torque (MA)

Table 7 Calculates tightening torque boundary conditions

Parametric	Numeral
Minimum friction coefficient of thread (uGmin)	0.2
Minimum friction coefficient of head support area (uKmin)	0.2
Effective diameter of friction torque in bolt head or nut support area (DKm)	25.63mm

Tightening torque：

$$M_A = F_{M\,zul} \times \left[0.16 \times P + 0.58 \times d_2 \times \mu_{Gmin} + D_{KM} \times \frac{\mu_{Kmin}}{2}\right] = 755.5\ \mathrm{N.m} \quad (40)$$

The above verification method was adopted to calculate the M16 and M18 titanium alloy bolts used in this case device. The results are shown in TABLE VIII. The results show that the maximum preload of M16 bolt was 94.9kN and the maximum preload torque was 404.2N.m. However, the results show that the preload was not enough, the bolt connection interface slips, the joint surface opens and the surface strength was not enough. At this time, the maximum load of the bolt was 100.6kN, the working stress was 708.4 MPa, and the bolt cannot meet the requirements of safe use. The maximum preload of M18 bolt assembly was 115.6kN and the maximum preload torque was 537.7N.m, but the results show that the preload was not enough, the joint surface was open and the surface strength was not enough. At this time, the maximum load of the bolt was 120.5kN and the working stress was 696MPa, and the bolt cannot meet the requirements of safe use. The maximum preload of M20 bolt assembly was 148.3kN and the maximum preload torque was 755.5N.m, among which the six safety factors all meet the safety requirements.

Table 8 Comparison of bolt connection safety assessment of three specifications

Bolt Specification	SM ≥1.0	SF ≥1.0	SA ≥1.1	SG ≥1.2	SK ≥1.0	SP ≥1.0	Maximum Preload	Maximum Preload Torque	Maximum Bolt Load	Bolt Working State Comparison Stress
M16	0.68	1.18	102.15	≦0	≦0	0.47	94.9 kN	404.2 N.m	100.6 kN	708.4 MPa
M18	0.82	1.20	129.28	≦0	≦0	1.34	115.6 kN	537.7 N.m	120.5 kN	696 MPa
M20	1.13	1.19	125	3.5	1.88	1.53	148.3 kN	755.5 MPa	155.69 kN	703.7 MPa

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3.3 M16, M18 and M20 bolt test verification

M16, M18 and M20 bolts were tested and verified on this device. When M16 bolts were used, one bolt broke during the experiment, as shown in Fig. 5. When M18 bolts were used, the bolts bend during the experiment, as shown in Fig. 6. However, M20 bolts did not bend or break during the experiment, meeting the requirements of use.

The experimental results verify the reliability of VDl2230 standard in bolt safety assessment calculation, and illustrate the limitations of the first strength theory and the third strength theory.

Fig. 5 Fracture photos of M16 bolt after test

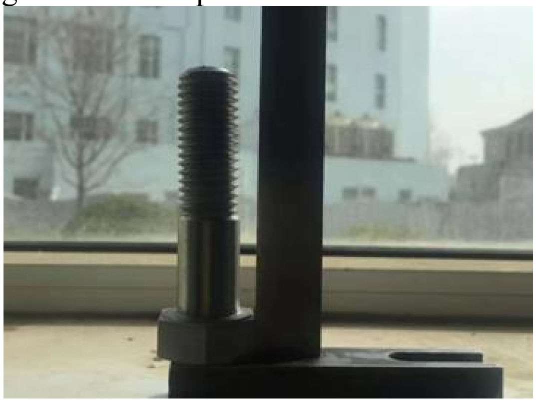

Fig. 6 Bending photos of M18 bolts after test

Conclusion

In this paper, the calculation method of "VDL2230-2003 Bolt Strength Checking Standard" was analyzed. For the bolt connection between the shell of a certain type of equipment and the flange of the back head, the safety evaluation calculation was carried out. The finite element method was used to calculate the load boundary conditions of the flange bolt connection, wherein the maximum axial load of the bolt was 65592N, the maximum shear force was 1181N. The maximum bending load was 248280N.mm. Bolt safety calculation was carried out by combining the first strength theory, the third strength theory and VDl2230 standard. The results show that, based on the first strength theory and the third strength theory, the bolts of the three specifications all meet the requirements of safe use. Based on "VDL2230-2003 Bolt Strength Checking Standard", the safety check and pretightening force prediction of connecting bolts were carried out. M16 and M18 bolts do not meet the requirements of safe use, while M20 titanium alloy bolts can meet the requirements of six safety factors, and the assembly pretightening force was 82.4kN-148.3kN. Finally, three kinds of bolts were tested and verified, M16 bolt fracture phenomenon, M18 bolt bending phenomenon, M20 bolt meet the requirements. In the actual tightening process, the accuracy of tightening tools and other discreteness should be considered to ensure that the actual pretightening force of bolts does not exceed this range.

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