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ABSTRACT

BOG (Boil off Gas) compressor deals with the gas evaporated from the LNG storage tank. Its working temperature is lower than -120℃. There are two types of starting process for BOG compressor: Direct Starting and No-load Starting. During the starting process, the temperature on cylinder will change rapidly and result in additional thermal stress to some parts on the cylinder, especially to the bolted joints on the cylinder head. In this paper, transient Finite Element thermal analysis is proposed on the cylinder with some improvement of the boundary condition settings, such as the consideration of the ice increase on the cylinder wall. Then, the theoretical and transient FE analysis are proposed subject to the bolted joints of the cylinder head in two starting process. Result shows that the maximum temperature difference on the cylinder is 81.5℃ during direct starting; while it could decrease to 60.9℃in No-Load Starting. Along the bolted joints in the cylinder head, the maximum temperature difference will be up to 52℃ in direct starting and 45℃ in No-Load Starting. The preload increases rapidly over 60% and its mean tensile stress on the bolt is near to the yield strength. Besides, the preload are distributed unevenly during the starting process. The maximum uneven rate is 20% in Direct Starting and 15% in No-Load Starting. It shows that No-Load Starting could offer a more stable starting process. Finally, based on the theoretical analysis, a preload adjustment method is proposed to ensure the safety and validity of bolted joints on BOG compressor.

1. INTRODUCTION

The liquefied natural gas in LNG storage tank will produce evaporated gas when absorbing the incoming heat. Using the compression and reflux after condensation treatment can effectively save these resources. BOG compressor is a kind of equipment that deals with the gas evaporated from the LNG storage tank. Its cylinder components usually work lower than -120℃. Working at such a low temperature, the mechanical behavior of BOG compressor's various parts, especially the bolted joints, will result in great changes. Therefore, it is necessary to analyze its stress and deformation distribution at low temperature to ensure the safely and effectively operation of the compressor at low temperature.

For BOG compressor and many machines, its bolts are tightened in ambient temperature during assembling process, but its operating temperature is far lower. Various components of compressor would face thermal contraction at low temperature. And the thermal contraction of different components varies because of their various materials. For example, BOG compressor cylinder head is made of cast iron (QTiNi35), whose coefficient of thermal expansion is lower than 304 steel, used by cylinder head bolts. The properties of two materials are shown as Table 1.

Table 1: Related properties of two kinds of materials

Materials	**Elasticity modulus** Pa	**Poisson's ratio**	**Thermal expansion** $^{o}C^{-1}$
304 steel	1.95×10^{11}	0.3	16.5×10^{-6}
QTiNi35	2.10×10^{11}	0.3	0.9×10^{-6}

Therefore, the preload of the bolt will change due to the different thermal deformation. In this case, the preload will become larger in low temperature. In other words, even if the bolts were tightened properly at ambient temperature, the preload would be too larger in the low-temperature environment. Accordingly, it is very important to research on the preload characters of the bolts on BOG compressor cylinder head at low temperature and further to guide to select the preload at ambient temperature.

There are mainly two kinds of BOG compressor starting mode. One starts directly at ambient temperature, which is called Direct Starting. The other can be called No-load Starting: during this starting process, cryogenic gas flow into the cylinder and compressor works by keeping the suction valve and discharge valve open until the cylinder temperature is stable. No-load Starting is more conservative than Direct Starting to avoid sharp changes of cylinder temperature when the compressor is in the normal operation and improve the safety. But it needs more time and energy than Direct Starting. At present, both of the starting methods are applied in commercial BOG compressor. One of the tasks of this study is to compare these two starting mode and confirm the advantage of No-load Starting.

The steady and transient thermal characteristics of BOG compressor's starting process has been focused on the steady mechanical behavior of the bolted joints on the cylinder head (Shen *et al*., 2012) and (Ding *et al*., 2013). Nevertheless, the transient mechanical behavior of bolted joints during BOG compressor's starting process has not been concerned. Therefore, this paper consists two parts. Based on predecessors' analysis, the boundary conditions of cylinder thermal simulations are improved and the thermal characteristics of Direct Starting in ambient temperature and No-Load Starting are compared in Part I. In Part II, based on the transient thermal condition, theoretical and transient mechanical analysis are proposed in order to research the stress behavior of bolted joints.

2. FE Transient Thermal Analysis

2.1 Model and Analysis Condition

BOG compressor cylinder structure is shown in Figure 1. It is composed by cylinder, cylinder head, piston and bolt on cylinder head. Intake passage is set up on the cylinder top while discharge passage is on the bottom side. The axial dorsal of cylinder is connected with piston rod. In the front, 12 of M36 bolts are arranged uniformly along the cylinder axis direction. Due to the complex structure of the cylinder, all-hexahedral meshing for FE is very difficult. Thus, meshes are components with most of hexahedron and small number of tetrahedral elements. Cylinder is symmetrical along axis. So we only take half of the calculation volume to reduce the computation time. Based on the experimental data (the time when the temperature of cylinder becomes stable), analysis time is set to 10 hour or 36,000 seconds.

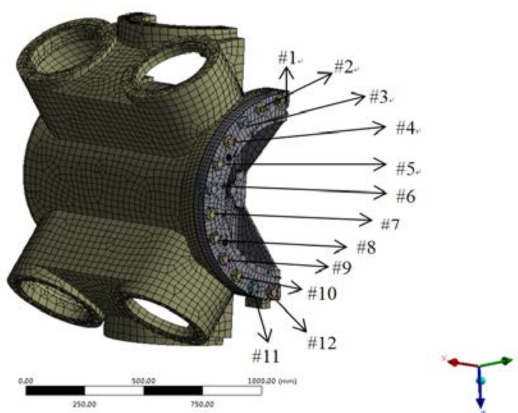


Figure 1: Model and meshing of bog compressor cylinder temperature field simulation

2.2 Boundary Condition of Direct Starting:

For BOG compressor cylinder's thermal analysis, the boundary conditions can be divided into three parts: the outer wall, intake and discharge passage and cylinder compression chamber. For the outer wall, heat can be transferred through the cylinder wall between cylinder and environment. It is considered as a stable natural convection boundary condition (Shen *et al*., 2012). However, BOG compressor's working temperature is extremely low. Discharge temperature of first stage, which is the maximum in the cylinder, is usually as low as -80℃. So the cylinder will be covered with a thick and increasing ice layer during the starting process. The ice layer will make the heat transfer conditions change, as shown in Figure 2. The ice thickness *D* increased with time. As a result, the equivalent heat transfer coefficient decreases gradually. Therefore, equivalent heat transfer coefficient of the outer wall of the cylinder h_{eq} is calculated by superposing of ice thermal resistance and natural convection thermal resistance. Its expression is shown as follows:

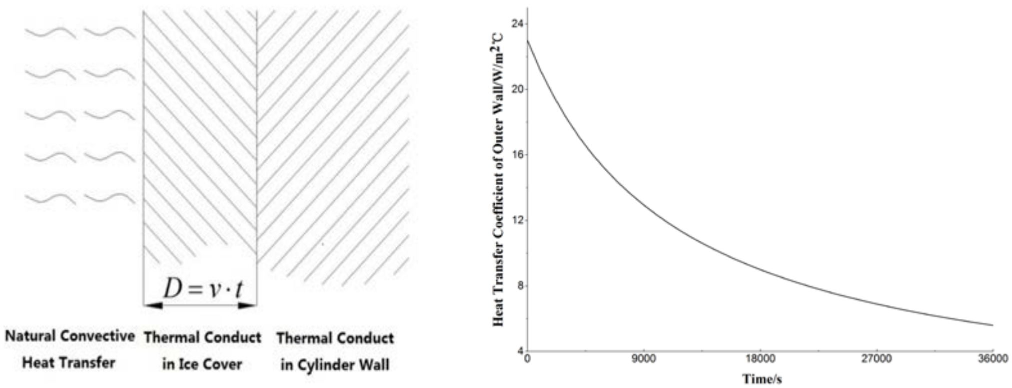


Figure 2: Variation of equivalent heat transfer coefficient

The ice thickness *D* increased with time. As a result, the equivalent heat transfer coefficient decreases gradually. Therefore, equivalent heat transfer coefficient of the outer wall of the cylinder h_{eq} is calculated by superposing of ice thermal resistance and natural convection thermal resistance. Its expression is shown as follows:

$$h_{eq} = \frac{1}{\dfrac{1}{h_n} + \dfrac{vt}{\lambda_i}} \tag{1}$$

Where v represents ice growth rate, which can be obtained by compressor starting test; t is the compressor start time; λ_i is thermal conductivity of ice; h_n is natural convection heat transfer coefficient. In this paper, the growth rate of ice is 8.35×10^{-6} m/s, which is estimated by experimental data. Convective heat transfer coefficient decreases from 23 to 5 with time. It shows that ice growth has significant effect on heat transfer boundary conditions of the outer wall of cylinder.

Heat transfer coefficient of the intake and discharge passage can be calculated by a general correlation for forced convection in tubes. Gnielinski formula (Holman, 2001) is used to apply for the boundary conditions of simulation.

$$\begin{gathered} Nu = \frac{(f/8)(Re-1000)Pr}{1+12.7\sqrt{f/8}(Pr^{2/3}-1)} \cdot \left[1+(\frac{d}{l})^{2/3}\right] \\ f = (1.82\lg Re - 1.64)^{-2} \end{gathered} \tag{2}$$

Where Re represents Reynolds number; Pr is Prandtl number; Nu is Nusselt number; d is equivalent diameter; l is passage length.

When compressor is working, the temperature and pressure of the gas in cylinder compression chamber, as well as the convective heat transfer coefficient, is changing with time. However, the convective heat transfer coefficient correlations of the cylinder compression chamber have not been found. In the similar study (Shen *et al*., 2012), convective heat transfer coefficient correlations of the inner wall of internal combustion engine cylinder are adopted for substitution. This paper adopts the Hohenberg formula (Neshat & Saray, 2014):

$$\alpha_g = 130 \cdot V_g^{-0.06} \cdot P_g^{0.8} \cdot T_g^{-0.4} \cdot (C_m + 1.4)^{0.8} \tag{3}$$

Where P_g and T_g represent the transient pressure and temperature of the gas in cylinder, C_m is the average velocity of piston, and V_g is the volume of compression chamber.

According to the regularities of the compression chamber volume change with the crank motion and the P-V Graph for steady state of compressor, the variation of parameters of formula (3) with time can be obtained. The result is shown in Figure 3. A-B segment represents the expansion process. B-C is the suction phase. C-D is the compression phase, and D-A of the discharge phase. To simplify the calculation, the average convective heat transfer coefficient, $107.67\,\mathrm{W/m^2 \cdot K}$, which is calculated from the results of Figure 3, is used in the following analysis.

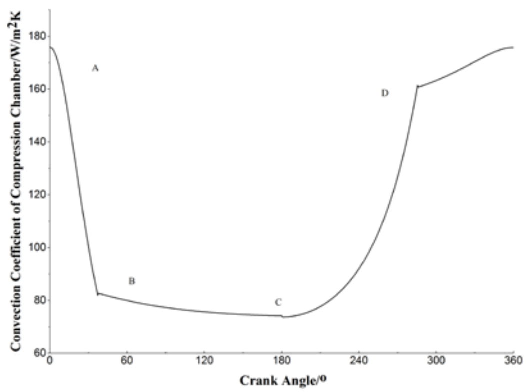


Figure 3: Variation of convection coefficient with crank angle

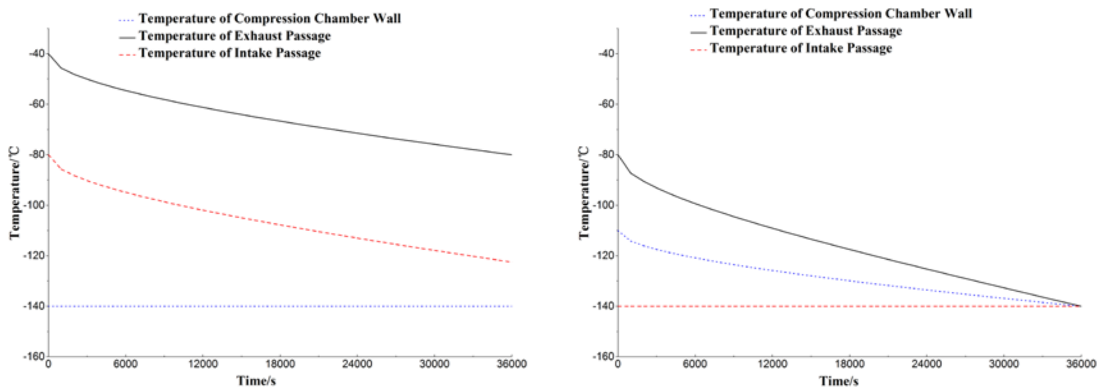
Besides, during compressor starting process, the gas temperature could be increased because of the heat obtained from cylinder inner wall. That is to say, gas temperature will be higher than its design value at the beginning of the starting process. Therefore, it is unreasonable to set the gas temperature constant. In this study, the temperatures of the gas in intake passage, the compression chamber and the discharge passage decrease with time. Based on the design parameters, an index changes correlation is used to show the compressor chamber average temperature variation ignoring variations over each crankshaft rotation. The expression is shown below.

$$T(t) = \frac{1}{\dfrac{1}{-T_\infty} + \dfrac{(\frac{1}{T_\infty} - \frac{1}{T_0})}{(t_{\max})^a} \cdot t^a} + (T_\infty + T_0) \tag{4}$$

Where T_∞ represents steady-state temperature, the design value; T_0 is the gas temperature of initial starting process, which can be obtained by the compressor test data; $t_{\max}$ is the total test time, which is 36000 seconds in this case; a is the exponent refers to the temperature decrease rate, generally taking a < 1, the smaller value represents the faster drop of temperature.

2.3 Boundary Conditions of No-load Starting

No-load Starting process is that the valves are forcing-opened and cryogenic gas flow through the cylinder without compression. So the boundary condition setting is similar to Direct Starting with a little difference. Its outer wall boundary condition is almost the same as Direct Starting. Intake and discharge passage boundary conditions use the same model as the Direct Starting: Gnielinski model. The difference is that the heat transfer coefficients of intake passage and discharge passage are equal because there is no compression process. The biggest difference between No-load Starting and Direct Starting is in compression chamber. Absent of the compression process, the boundary conditions of compression chamber can be calculated by Gnielinski model at No-load Starting process. Temperature variations of two starting methods in each boundary are shown in figure 4. Boundary conditions of two starting methods are summarized in Table 3.



a) Direct Starting b) No-load Starting

Figure 4: Temperature variation in each boundary

Table 3: Summary of temperature field boundary conditions of BOG compressor simulation

Conditions	items	Intake passage	Discharge passage	Compression chamber	Outer wall
Direct Starting	h (W/m^2·K)	123	103	Fig 6	Fig 4
	T_0 (℃)	-140	-40	-80	33
	T_∞ (℃)	-140	-80	-122.5	33
No-load Starting	h (W/m^{2741}·K)	123	123	151	Fig 4
	T_0 (℃)	-140	-80	-110	333
	T_∞ (℃)	-140	-140	-140	33

2.4 Results and analysis

The simulation of cylinder temperature field requires many initial boundary conditions, most of which are hypothetical conditions based on the design parameter. So it is necessary to compare the result with experimental data to verify its reliability. Experimental data used in this section is real-time temperature data in BOG compressor prototype without load test made by Zhejiang Qiangsheng Compressor Manufacturing Co. The compressor prototype's design parameters and structure are consistent with the simulation. Comparative results are shown in the Figure 5.

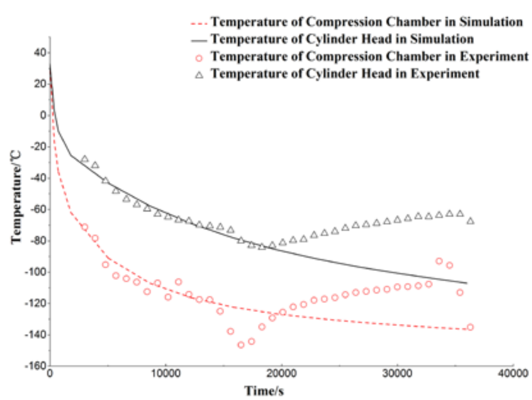


Figure 5: Comparison between simulation and experimental

Firstly, the simulation and experiment value have a good consistency on the trend, especially for the starting half of the data. In the following time, the consistency is lower. It is probably because during the test, the temperature of intake gas is controlled by the proportion of liquid and gas. The flow fluctuation will affect the temperature of the mixed gas. Therefore, the stability of the intake gas temperature is hard to keep stable. In conclusion, the results can preliminarily prove the reasonableness of boundary conditions of simulation.

Figure 6 shows the variation of the extreme value of cylinder temperature during starting process. It can be seen that the minimum temperature dropped faster than the maximum temperature inside the cylinder. Thus, the maximum temperature difference inside the cylinder, around 150℃, did not appear at the end of the start process, but at the earlier stage. Two start conditions show similar trends, and the No-load Starting has smaller temperature difference inside the cylinder. Figure 7 shows the temperatures changes of each bolts at the cylinder head in two start conditions.

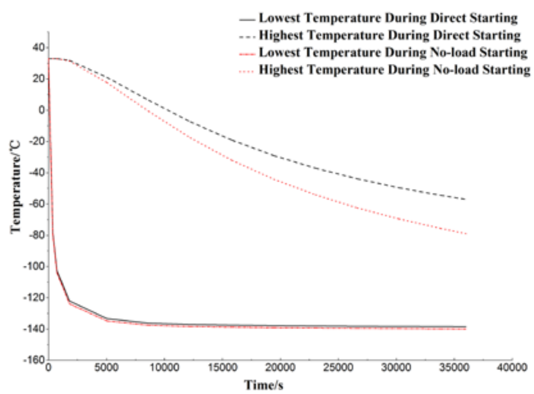
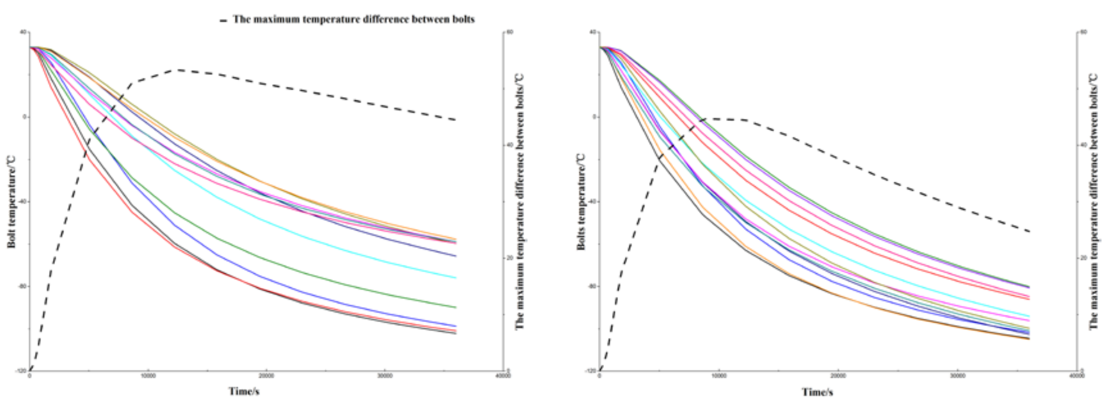


Figure 6: Variation of compressor cylinder extreme temperature during starting process



a) Direct Starting b) No-load Starting

Figure 7: Temperature distribution and temperature difference

Due to the obvious temperature gradient in the cylinder, the temperature distributions of each bolt are not uniform. When compressor starts, temperature differences of bolts rises rapidly, and decreases slowly after reaches peak temperature, whose patterns are consistent with the cylinder. There are still some differences of the temperature distribution in two starting conditions. During the Direct Starting condition, the temperatures of the bolts, which are close to the intake side, decrease quickly, but slowly close to the discharge side. Eventually, the steady temperatures difference is about 45℃ between two sides. The maximum temperature difference of bolt is 52℃, but it would decrease to 45℃ during the No-load Starting. The temperatures of bolts are low on the intake side and discharge side, but higher in the middle. Temperature distributions of bolts are more uniform. Different steady temperature indicates that the preload will change in different ranges. When compressor is working at low temperatures, cylinder head will be under an uneven preload state.

3. Structural Analysis of Bolted Joints

During the previous section, BOG compressor starting transient temperature field of cylinder were analyzed. Transient temperature field would be loaded into the transient mechanical analysis in this section. Bolt preload theoretical prediction formula would be applied to determine the cylinder head bolt preload at ambient temperature.

3.1 Introduction of Theoretical Analysis

Preload represents the degree of tightening bolts. Preload value shall comply with the relevant standards. According to the standard (ANSI/ASME B18.2.6, 1996), preload values, which will serve as an initial value in theoretical calculation and finite element calculation, can be represented by the following equation.

$$F_0 \le (0.5 \sim 0.6)\sigma_s \cdot A_s \tag{5}$$

Where σ_s represents the yield strength of the bolt material; A_s is the minimum cross area of the bolt. The formula shows that the preload should make the average tensile stress within the bolt reaches 0.5 to 0.6 times of the yield stress. Excessive preload value may make the bolt yield even failed, and small preload value cannot guarantee good preloading effect. It is necessary to analysis the bolt preload changing from ambient temperature to working temperature and adjust the assembling preload value to meet the requirements. The preload at ambient temperature is 250kN. The bolt structural sizes are in the Table 4.

Table 4: Original parameters

Structure parameters			
Bolt diameter/mm	32	Thickness of nut/mm	15
Nut outer diameter/mm	52	Thickness of the bolt head /mm	30
Bolt head outer diameter/mm	100	Nut inner diameter/mm	32
Bolt head inner diameter/mm	50	yield strength	640MPa

The theoretical model is taken from the earlier study (Yan *et al.*, 2012). The calculations can be broadly divided into the following steps: The first step is to get the original parameters. Mainly include the structure size parameters, material properties, as well as the working condition.

The second step is to calculate the axial stiffness of each component. For bolt, nut and bolt head, the axial stiffness can be calculated by the following formula:

$$k = \frac{EA}{L} \tag{6}$$

Where E represents the elastic modulus of the material; A is the force area; L is the length along direction of the force.

However, for the clamped objects, such as cylinder and cylinder head, the formula (6) cannot be used due to these complex shapes. Therefore the finite element method is applied to calculate the axial stiffness of connecting pieces in this section, whose result is $2.5\times10^9\,\mathrm{N/m}$。

Next, calculating the modified stiffness of clamped objects k_c' and the deformation distribution coefficient m :

$$k_c' = \frac{1}{\frac{1}{k_c} + \frac{1}{k_n} + \frac{1}{k_h}} \tag{7}$$

$$m = \frac{k_c'}{k_c' + k_b} \tag{8}$$

Where k_c presents the stiffness of clamped objects, k_h and k_n are the stiffness of bolt head and bolt. Axial stiffness and deformation distribution coefficient of various components are shown in the Table 5:

Table 5: The axial stiffness of several components and the distribution coefficient

Stiffness of each component			
Bolt stiffness k_b / N/m	2.614×10^9	Clamped object stiffness k_c / N/m	2.500×10^9
Nut stiffness k_n / N/m	1.507×10^{10}	Modified clamped objects stiffness k_c '/ N/m	3.822×10^9
Bolt head stiffness k_h / N/m	3.725×10^{10}		
Distribution coefficient m			
0.44			

Finally, the range of bolt preload change could be calculated by following formula:

$$F' = F - m\cdot(\alpha_b - \alpha_c)\cdot \Delta T \cdot L \cdot k_b \tag{9}$$

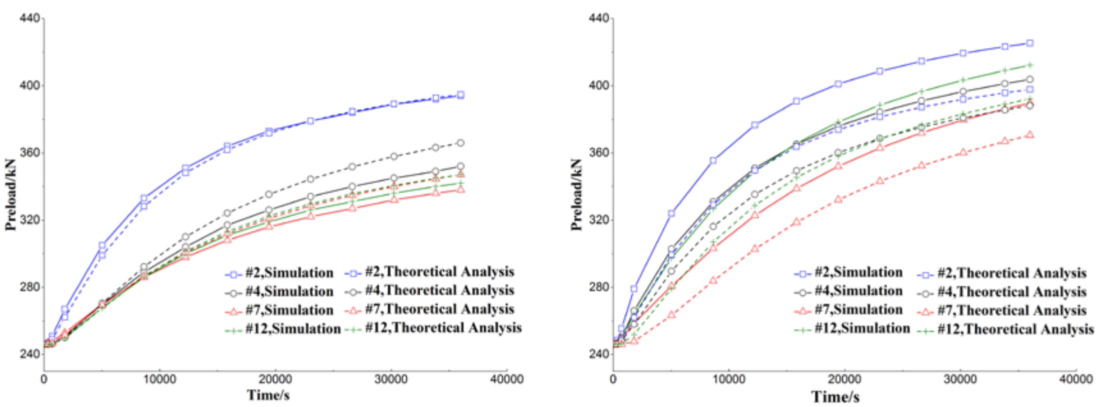
Where F' represents the preload at low temperature; F is the preload at ambient temperature; α_b is coefficient of thermal expansion of bolt; α_c is coefficient of thermal expansion of connecting piece; ΔT is temperature difference.

3.2 Introduction of the pre-processing of finite element analysis

The analysis in this section is a continuation of the cylinder transient thermal analysis in previous section. So the model and mesh are consistent with Figure 1. In accordance with the spatial position, bolts are numbered along the half circumference of the cylinder head. The preload is simulated by applying an inward compressive force, 250kN, in the middle of cross-section of the bolt. Two steps are taken in analysis. The preload of bolt is loaded in the first step and the temperature field is imported in the second step. The whole analysis time is still 36000s.

3.3 The results and analysis

4 representatives of the bolts along the cylinder head are selected to shown the preload changes in Figure 8.



a) Direct Starting b) No-load Starting

Figure 8: Bolt preload change and calculation error distribution

An analysis of the results can be viewed in several aspects: 1) The materials of cylinder and cylinder head (lower expansion coefficient) are different from the bolts (higher expansion coefficient). So the bolt preload will rise at low temperatures. Average tensile stress of bolt section increased from 320MPa in ambient temperature to 468MPa (Direct Starting condition) and 525MPa (No-load Starting condition), equivalent to 73% and 82% of the material yield strength respectively. 2) Errors of bolt preload between theoretical calculation and finite element values are within 10%, which shows good consistency between theoretical calculation and finite element analysis. The maximum error of Direct Starting conditions is 4%, and 8% of the No-load Starting conditions. 3) The bolt preload distribution at low temperatures is no longer uniform because of the significant temperature difference in cylinder head during starting. For Direct Starting condition, the area of cylinder head near the intake passage is at low temperature, but other area near the discharge passage is at higher temperature. Thus, after the start, the maximum preload is 395kN, located at #2 bolts, and minimum preload is 346kN, located at #8 bolts. However, for No-load Starting condition, the area of cylinder near the intake passage and discharge passage head is at low temperature, but other area in the middle is at higher temperature. Thus, after the start, the maximum preload is 398kN, located at #2 bolts, and minimum preload is 371kN, located at #7 bolts. The preload unevenness rate is defined as the following formula:

$$\delta = \frac{\sigma_{\max} - \sigma_{\min}}{\overline{\sigma}} \times 100\% \tag{10}$$

Where $\sigma_{\max}$ and $\sigma_{\min}$ are the maximum and minimum preload; $\overline{\sigma}$ is average preload. Bolt preload unevenness of two start conditions are shown in Figure 9. Due to the low maximum uniformity of No-load Starting condition, using this method could reduce the stress fluctuation.

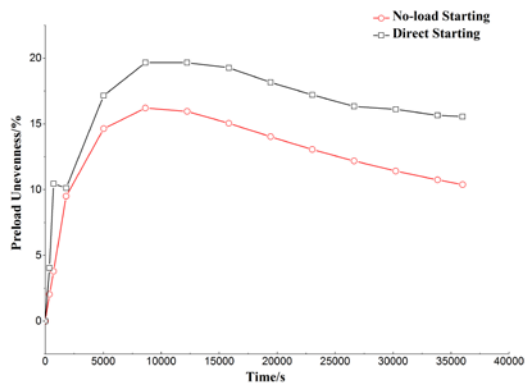


Figure 9: Preload unevenness distributions of two conditions

3.4 Adjustment method for cylinder head bolt assembling preload values

According to the previous section, preloads of BOG compressor cylinder head bolts at the operating temperature are significantly higher than the ambient temperature. Meanwhile, preload distribution is becoming uneven during the starting process. Therefore it is necessary to adjust the value of preload at ambient temperature. Finite element analysis proved the validity of the theoretical formula in previous section. In this section, the theoretical formula is used to deduce the recommended range of bolt preload value in different conditions.

As for the bolts that have great difference between operating temperature and ambient temperature, safety and effectiveness under two conditions should be considered for determine preload values,

$$\begin{cases} b\sigma_s A \le F \le \sigma_s A / a \\ b\sigma_s A \le F - m(\alpha_b - \alpha_c)\Delta T \cdot L \cdot k_b \le \sigma_s A / a \end{cases} \tag{11}$$

Where a represents coefficient of safety, whose value must be bigger than 1; b is coefficient of effect, whose value could be 0.3-0.5 depending on the circumstances, such as requirements for the sealing performance.

The formula (11) can be simplified as follow: If bolt preload rises at the operating temperature, it mustn't be excessive at operating temperature but not too small when assembled at ambient temperature. The expression is as follows:

$$b\sigma_s A \le F \le \sigma_s A / a + m(\alpha_b - \alpha_c)\Delta T \cdot L \cdot k_b \tag{12}$$

Conversely, in the situation that preload decreases at the operating temperature, it must be not too small at operating temperature and not excessive when assembled at ambient temperature. The expression is as follows:

$$b\sigma_s A / a + m(\alpha_b - \alpha_c)\Delta T \cdot L \cdot k_b \leq F \leq \sigma_s A / a \tag{13}$$

In this article, *a*=1, *b*=1.5 according to the requirement of the bolt's load condition. The ranges of each bolt preload are shown Table 7. The bolds marks the boundary of the bolt's stress condition.

Table 7: Range of bolt preload

	Preload (kN)		Tensile Stress(MPa)				Yield Ratio (%)			
			Room Temperature		Low Temperature		Room Temperature		Low Temperature	
	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
#1	148	183	**184.02**	227	289	**324**	**28.75**	35.55	45.16	**50.63**
#4	148	193	**184.02**	240	279	**324**	**28.75**	37.50	43.59	**50.63**
#7	148	226	**184.02**	281	247	**325**	**28.75**	43.91	38.59	**50.78**

3.5 Conclusions

According two starting conditions of BOG compressor, cylinder transient temperature field has been analyzed in this paper. Then theoretical formula and finite element method were used to analyze the preload changes of cylinder head bolts. Finally based on the theoretical calculation formula, adjustment method of bolts preloads at ambient temperature was given. Conclusions are following.

- Temperature distribution of cylinder is more uniform during No-load Starting. compared with the Direct Starting， The maximum temperature difference reduced over 20 ℃.
- Theoretical formula could reflect the variation of bolt preload at low temperature, and agree with the results of finite element simulation well.
- The cylinder head bolt preload will increase substantially at low temperature. The tensile stress reaches 517MPa, close to the yield strength of the material.
- Bolt preload distribution is no longer evenly at low temperatures. The maximum unevenness is 18.8% and ended unevenness is 13.5% during Direct Starting. The maximum unevenness is 16.0% and ended unevenness is 7.1% during No-load Starting.
- Based on the theoretical model, a preload adjustment method is proposed in this study, after the test verification, it can be used to ensure the preload values are even and moderate at operating temperature.

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