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Zhao, Bin; Du, Bin; Peng, Xueyuan; and Feng, Jianmei, "Effects of Low Suction Temperature on the Boil-off Gas compressor" (2014). *International Compressor Engineering Conference.* Paper 2352.
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Effects of Low Suction Temperature on the Boil-off Gas compressor

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ABSTRACT

The Boil-off Gas (BOG) compressor is used as a key facility in the liquefied natural gas (LNG) terminal, to recycle the excessive boiled gas for re-liquefaction or direct application. The low suction temperature down to -162℃ brings about big challenges in design of the BOG compressor. In this paper, the three-dimensional finite element model was used to simulate both the static and periodic transient temperature distribution in the cylinder of a BOG compressor, and a computational fluid dynamics (CFD) model was established to calculate the flow and heat transfer inside the compression chamber and suction/discharge pockets. A test rig was built up to validate the simulated results. The results showed that, the average temperatures in the suction and discharge pockets were about -109℃ and -60℃, respectively, and the temperature of the compression chamber was in-between. The maximum temperature difference between outer and inner surface of the compressor cylinder reached up to 84℃ during start-up of the compressor, which yielded a thermal strain and stress in the cylinder much larger than those during steady operation of the compressor with only 31℃ of temperature difference. A variety of pre-cooling temperatures ranging from -20℃ to -60℃ were examined, and the results showed pre-cooling down to -20℃ before start-up was good enough. The amplitude of temperature fluctuation due to the periodic movement of the piston was less than 0.1℃ in the cylinder wall. The temperature coefficient tended to decrease at lower suction temperature. As the suction temperature decreased from -54.2℃ to -142.2℃, the suction coefficient dropped drastically by 24.4%.

1. INTRODUCTION

Natural gas is one of the three largest energy sources in the world. Its high-efficiency, high quality, and extensive applications give it an increasingly important role in the energy infrastructure. After liquefaction, one cubic meter of liquefied natural gas(LNG) generally contains 625 cubic meters of natural gas in the gaseous state, which makes it remarkably economical for storage and transportation. As a result, the global demand for LNG has grown rapidly in recent years[1-3].

It is well known that LNG boils and evaporates as heat transfers from the surrounding environment to the container which have an internal pressure close to atmosphere pressure, and that this process pressurizes the container. As the amount of boil-off gas(BOG) increases, the pressure in the container rises. If the pressure in the container exceeds its designed pressure, the operation of transportation and receiving systems for LNG will be severely threatened. To maintain the steady pressure in the container , there must be a mechanism to dispose of the excessive BOG[4]. Figure 1 shows one entire liquefied natural gas system including handing of the BOG in a typical LNG terminal[5]. One traditional method of dealing of extra BOG is burning it and using the energy to power steam turbines. Another solution is to recycle the excessive Bog in liquid state using a BOG reliquefaction plant[6]. So the Bog must first be pressurized, and then either directly delivered to the pipelines or recycled back into the LNG containers[7,8]. The BOG compressor is an essential part of the process of handling the boil-off gas of LNG[9,10] ,and the receiving systems of LNG.

The suction temperature of BOG compressors in the LNG system can be low to -162℃,consequently the effects of such low temperature must be taken into consideration while designing the BOG compressor. The large temperature difference between the low-temperature suction chamber and the surrounding environment leads to thermal stresses on the cylinder, especially the bolts connecting cylinder and cylinder head, influencing the structure and materials of the cylinder and the heat treatment process. As the piston is in direct contact with the low temperature gas , the piston rings must be oil-free ,and the self-lubricated material must have appropriate friction and wear characteristics. For these reasons the heat transfer in the cylinder and the thermodynamic process in the compressor chamber are critical issues in the design of BOG compressors.

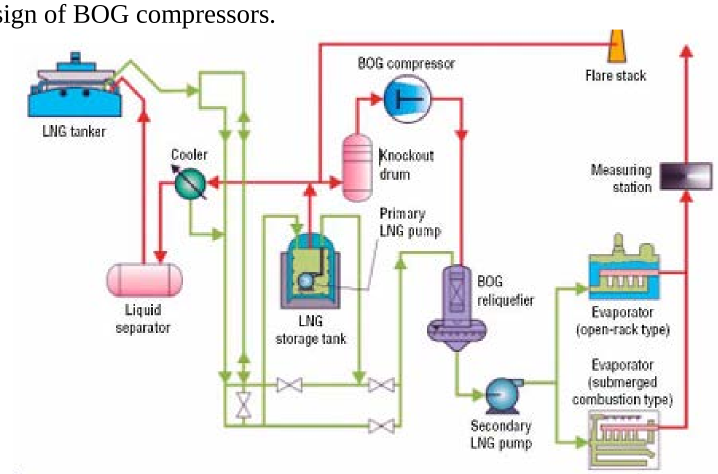

Figure 1: LNG system in a typical LNG terminal [5]

Many researchers have studied the thermodynamic process and heat transfer in the cylinders of positive displacement compressors. Adair et al. [12] examined the available correlations for inchamber convective heat transfer in reciprocating compressors and engines. They derived a new correlation that successfully addressed the effect of in-chamber convective heat transfer in reciprocating compressors. Chong and Watson [13] developed a precise procedure to predict the heat transfer and pressure variation in the cylinders of reciprocating air compressors, taking into consideration the influence of the crank shaft rotational speed. Liu and Zhou [14] determined the temperature distribution on the cylinder wall of a reciprocating refrigeration compressor and the heat transfer coefficient between the cylinder wall and the gaseous refrigerant under various pressure ratios, rotational speeds and suction temperatures. In addition, Recktenwald et al. [15] used the three-node model, together with the finite-difference model, to investigate the instantaneous heat transfer between the cylinder walls and gas in a reciprocating compressor. Keribar and Morel [16] developed a new methodology to calculate the gas-to-wall heat transfer based on in-cylinder flow velocities. This new methodology could be applied to predict heat transfer in compressors as a function of speed, pressure ratio, fluid properties and compressor valve and piston geometry. Fagotti et al. [17] compared several widely used heat transfer models in a compressor simulation program, and established the most reasonable model. Pereira et al. [18] presented a numerical analysis of the unsteady in-cylinder heat transfer using a simplified model of a small reciprocating compressor under actual operating conditions, and concluded that heat transfer was strongly affected by suction and discharge processes. A two-dimensional model was developed by Ooi and Zhu [19] to study the fluid flow and heat transfer in the working chamber of the scroll compressor, and a significantly higher convective heat transfer between the gas and scroll wrap walls was demonstrated. Ooi [20] focused on the analytical study of heat transfer and temperature distribution for a hermetic reciprocating refrigeration compressor using the lumped thermal conductance approach. When the results of his study were applied to actual compressor designs in industry, they visibly enhanced compressor performance. Tan and Ooi [21] produced a theoretical study of inchamber convective heat transfer between the working fluid and the surrounding chamber wall for a novel Revolving Vane (RV) compressor design, and, as expected, improved the accuracy of the mathematical prediction of the novel RV compressor. An unsteady state analysis of the compression cycle of a small hermetic reciprocating compressor for domestic refrigeration was carried out by Longo and Gasparella [22], and the mass and energy balances were applied to the refrigerant inside the cylinder to determine the mass, pressure and temperature, and the heat transfer during the compression process. Yuhong Shen et al.[24] adopted a three-dimensional finite element model to investigate the heat transfer process in the cylinder of a BOG compressor, and it is proven that temperature fluctuation only existed on the inner surface of the cylinder and that its amplitude was lower than 0.003 ℃.

In the above investigations, the factors including pressure ratios, rotational speeds, suction temperatures, fluid

properties, compressor valve and piston geometry, which may affect the heat transfer in the compressors were analyzed, and a 3-D model was used to simulate the temperature distribution of the cylinder of BOG compressor.
In this paper, a 3-D model of a BOG compressor cylinder including cover head and bolts connecting cylinder and cover head was established to calculate the temperature distribution and thermal stress. The temperature distribution was obtained based on the method from Yuhong Shen et al.[24] . The threads of bolts applied in this research were built with the helix angle neglected, and so were the internal threads in the cylinder.

2. Numerical analysis

2.1 Fundamental method

Heat exchanges in the BOG compressor occur in four ways. As shown in Figure 2, forced convection heat transfer occurs between the gas in the compression, suction or exhaust chambers and the inner surface of cylinder, natural convection heat transfer takes place between the ambience and the outer surface of the cylinder, frictional heat generation and conduction between the piston rings and the inner surface of cylinder, thermal conduction happens between the middle body and the cylinder. According to Yuhong Shen et al.[24] ,the factor with the strongest influence on the cylinder temperature distribution of BOG compressors is the temperature of the gas in the compression chamber. The heat exchange between the gas in the compression chamber and the wall of the cylinder is accompanied with the gas temperature variation and the moving heat transfer interface. To study such an involved process, researchers must consider not only the temperature variations in the expansion, suction, compression and discharge processes, but also take into account the reciprocating moving heat exchange boundary between the gas and the wall of cylinder.

2.2 Mathematical model

To study the temperature field with low temperature suction, the rational expression of the geometry of the BOG compressor was simplified. As the cylinder was a perfectly symmetrical structure, only a semi-cylinder was analyzed to better observe the temperature distribution and reduce computation time. Furthermore, to refine the finite element mesh, some tiny structures that have negligible influence on the temperature field of the cylinder were diminished. The FEM model of the cylinder is shown in Fig.4. There were 1,770,431 elements, and the mesh independence study had been adopted.

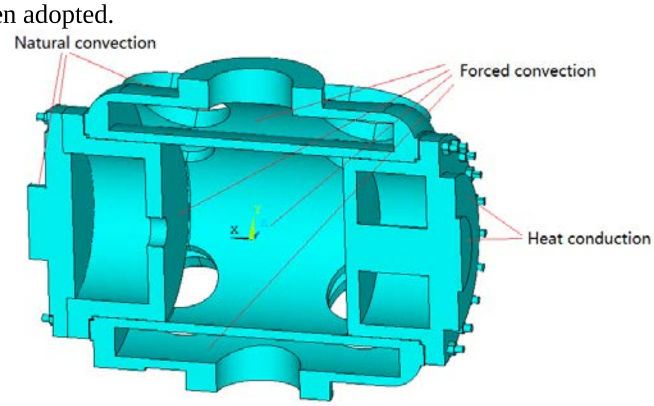

Figure 2: Heat exchange locations in the cylinder of the BOG compressor.

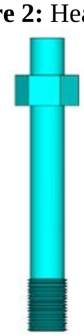

Figure 3: Model of coupling bolt ignoring helix angle

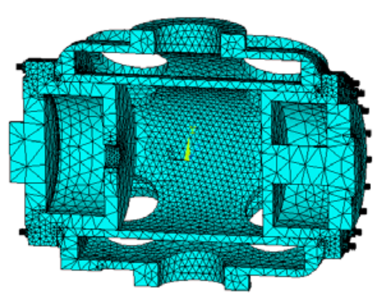

Figure 4: FEM model of the cylinder.

The cylinder was made from popular cryogenic materials, such as austenitic stainless steel and cast iron containing 35% nickel. In this simulation, the austenitic stainless steel is used as the material of bolts, and cast iron containing 35% nickel of cylinder. Their properties are shown in Table 1.
The rotational speed of the BOG compressor was 420 rpm, and the operating conditions are shown in Table 2.

Table 1: Material properties.

Material	Density $kg{\cdot}m^{-3}$	Conductivity $W \cdot m^{-2} \cdot K^{-1}$	Specific heat $kg^{-1} \cdot K^{-1}$	Elasticity module Pa	Thermal expansion coefficient
austenitic stainless steel	7980	14.6	370	2.1e11	1.65e-5
cast iron	8100	10.4	350	1.47e11	5.5e-6

Table 2: Operating conditions.

Conditions	Suction temperature ℃	Suction pressure MPa	Discharge temperature ℃	Discharge pressure MPa
Values	-162	0.115	-86	0.3

2.3 Boundary conditions

The Finite Element Analysis(FEA) of the temperature field of the cylinder needs boundary conditions of the inner and outer surfaces of the cylinder. There were three major boundary conditions taken into consideration in this paper , i.e. the pressure, temperature and convective heat transfer coefficients of the inner and outer surfaces of the cylinder. The pressure and temperature in the cylinder were calculated using a compressor thermodynamic process program based on the operating conditions, as shown in Figure 5.
The convection heat transfer coefficient in the compression chamber was calculated according to the Woschni formula [10]

$$h = 2817 \times D^{-0.214} \times (V_p \times P_i)^{0.786} \times T_i^{-0.525} \qquad (1)$$

Where, D is the cylinder diameter, V_p is the average speed of the piston, P_i is the pressure in the cylinder and T_i is the temperature in the cylinder. It can be seen that in Figure 6, the variation of the heat transfer coefficient was similar to that of the pressure. It decreased at the expansion stage, fluctuated near a constant at the suction stage. During the compression process, the heat transfer coefficient increased gradually, and at the discharge stage, the heat transfer coefficient increased firstly, and then decreased. The convection heat transfer coefficient in the suction and discharge chambers was calculated using the turbulent forced convection heat transfer in-tube formula (Dittus–Boelter correlation formula [10]).

$$Nu = 0.023 Re^{0.8} Pr^{1/3} \qquad (2)$$

Using the above equations, the heat transfer coefficients obtained for the suction and discharge chambers were 135 $W \cdot m^{-2} K^{-1}$ and 106 $W \cdot m^{-2} K^{-1}$, respectively.

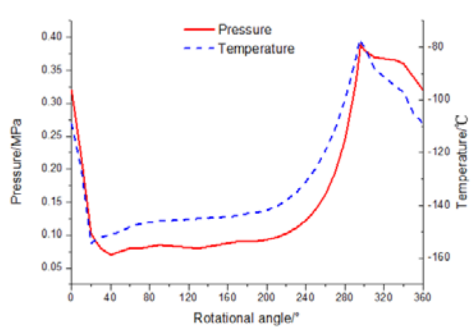

Figure 5: Pressure and temperature change versus rotational angle

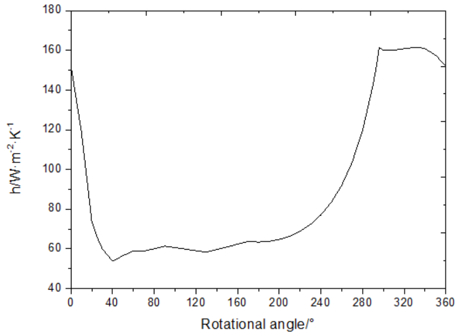

Figure 6: Convection heat transfer coefficient versus rotational angle

3. Experimental set-up

A test rig was built to measure the gas temperature distribution of inner and outer surface temperature of the cylinder at several suction temperatures, and the suction coefficient and temperature coefficient was analyzed based on the results, which significantly influence the volumetric efficiency. The main steps were as follows:

1) Precooling was adopted to reduce the thermal stress of cylinder. The BOG compressor would not run until the temperature of cylinder is below -30℃.

2) When the compressor operates steadily, the suction pressure was set at 0.12Mpa by adjusting voltage regulator and the discharge pressure was set at 0.32Mpa.

The experiment test rig was shown in Figure 7.

Figure7: Running experimental compressor and test rig

The experimental conditions were shown in Table.3.

Table 3 : Experimental conditions

	Condition.1	Condition.2	Condition.3	Condition.4	Condition.5
Suction temperature ℃	-142.2	-123.8	-88.8	-73.5	-54.2
Suction pressure MPaG	0.12				
Exhaust pressure MPaG	0.32				

The location of temperature sensors and pressure sensors was shown in Figure 8.The temperature sensor T1 is used to measure the suction temperature, and T2 to measure the outer surface temperature close to suction inlet, T3 to measure the outer surface temperature of the cylinder head, T4 to measure the exhaust temperature, T5 to measure the outer surface temperature of the cylinder close to exhaust inlet, T6 to measure the outer surface temperature of the cylinder close to exhaust inlet, T7 to measure the outer surface temperature of the cylinder away from exhaust inlet.

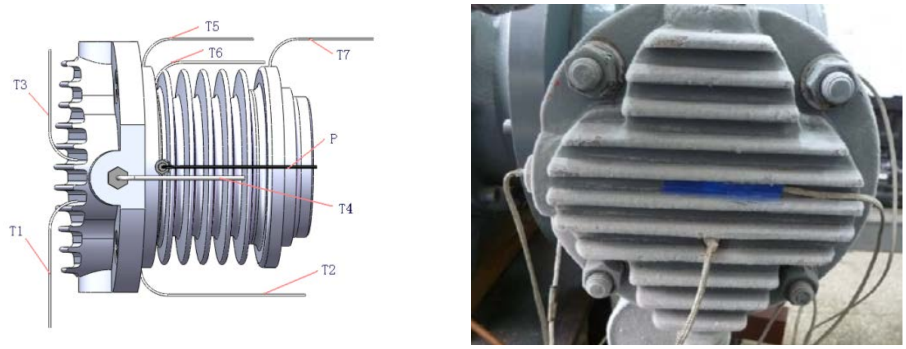

Figure 8: The location of the sensors

4. Results and discussion

4.1 The simulated temperature field

During the BOG compressor works steadily, the temperature field is still dynamic because of the reciprocating piston and the periodically varying boundary conditions in compression chamber. According to Yuhong Shen et al.[24] the temperature fluctuation only existed on the inner surface of the cylinder and its amplitude was very small, less than 0.003℃, so the temperature distributions for each rotational angle could be considered as steady-state in one cycle.

Figure 9 shows the dynamic temperature field during the steady running of the compressor. When the compressor operated steadily, the largest temperature difference in different parts of the cylinder was close to 127℃. The temperatures in the suction chamber were relatively low, with an average of -114℃. Temperatures in the discharge chamber were relatively high, averaging -62℃, while the temperatures in the compression chamber were between these two values.

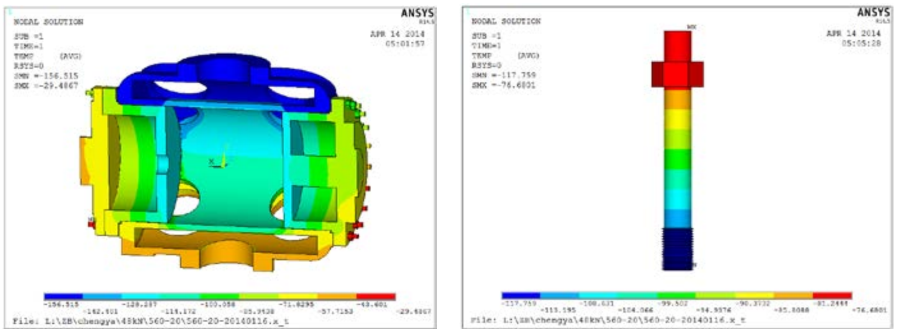

Figure 9 : Temperature distribution at low suction temperature(-162℃)

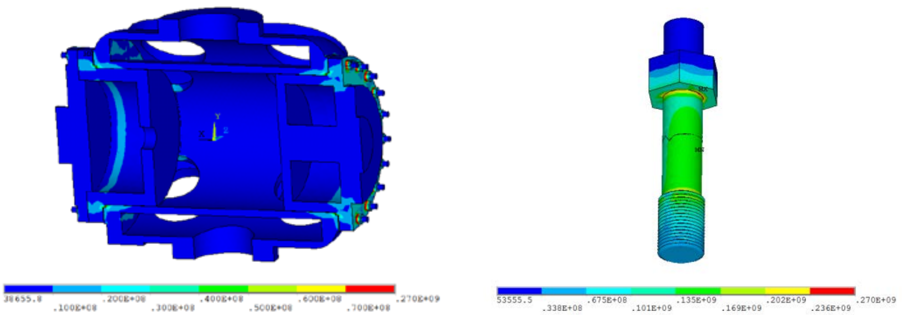

Figure 10 : Stress distribution of cylinder and typical bolt at ambient temperature

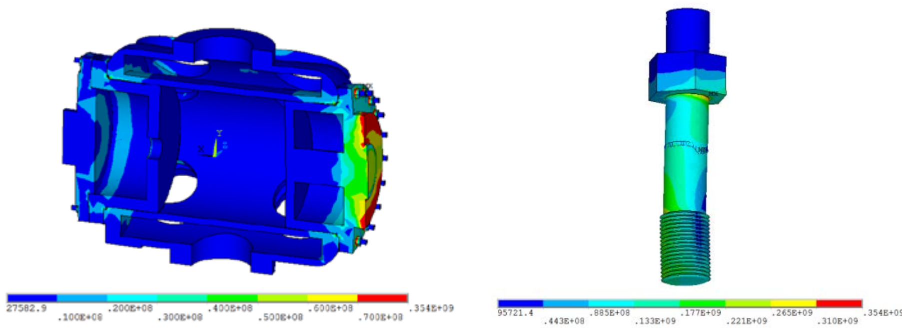

Figure 11: Stress distribution of cylinder and typical bolt at low suction temperature(-162℃)

A variety of pre-cooling temperatures from -20 ℃ to -60 ℃ were tested, the final temperature differences were same for any pre-cooling temperature, when the compressor worked steadily. But the lower pre-cooling temperature was, the lower largest temperature difference appeared at the first ten operation minutes. When the pre-cooling temperature was -20 ℃, the largest temperature difference decreased 23.8%, to about 64℃, when the pre-cooling temperature was -60 ℃, the largest temperature difference decreased 46.4%, to about 45℃, which was only 14℃ higher than when the compressor ran steadily. The relationship between the pre-cool temperature and the pre-cool time is shown in Figure 16. It needed only 195 s to pre-cool to-20 ℃, but needed 2392 s to pre-cool to -60 ℃. As pre-cooling to -60 ℃ took a very long time, and such a large drop in temperature difference is not necessary, pre-cooling to -20℃ would be satisfactory.

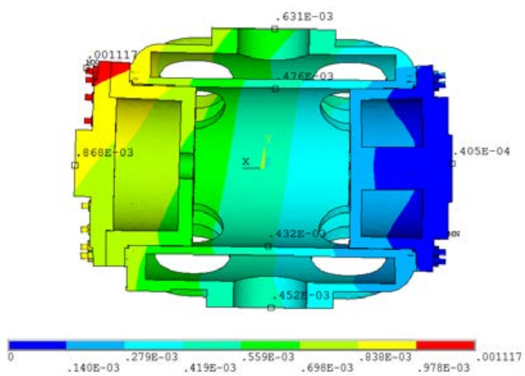

Figure 12: Displacement distribution of cylinder at low suction temperature(-162℃)

4.2 The simulated thermal stress

The obtained steady-state temperature was executed to study the effect of low suction temperature on the cylinder of BOG compressor, especially thermal stress of the cylinder and coupling bolts connecting the cylinder and cover head. Usually, the bolts were pretightened at ambient temperature to provide pretention forces, the bolts would be much tighter when the temperature of cylinder and bolts decrease, the more the temperature dropped, the tighter the bolts were pulled. The thread of bolts may be damaged, even the bolts were cracked if the pretention forces were not appropriate. This could be dangerous for a BOG compressor. A series of pretention forces from 20kN to 60kN were examined, and the stress distributions of cylinder at ambient temperature and low suction temperature were shown in Figure 10 and Figure11 while the pretension force is 48kN.The maximum stress at low suction temperature occurred was 354MPa at the thread of the nut and the bolt engaging on suction side while the value was only 270Mpa at the ambient temperature. The maximum displacement at low suction temperature was 1.117mm, shown in Figure12 .

4.3 Experiment results

4.3.1The temperature distribution. The temperature distribution of the main parts of BOG compressor obtained by FEM simulation and the experimental data was shown in Figure13. In addition, Figure14 shows the FEM simulation results at the suction temperature of -88℃. As can be seen, the simulation can forecast the temperature distribution of the cylinder well, and the temperature of the main parts decreased from crankcase to cylinder head, a large temperature difference existed on the cylinder outer surface. When the suction temperature decreased to -142.2 ℃, the temperature difference on the cylinder outer surface was up to 76℃.

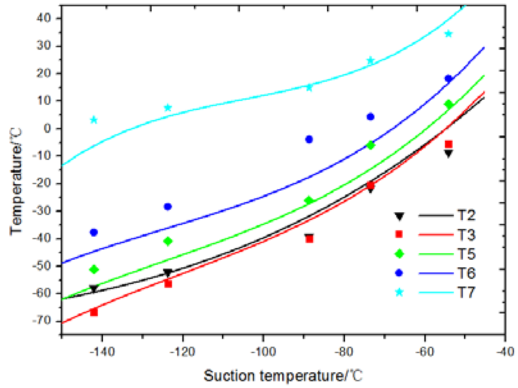

Figure 13: Simulated and measured temperature distribution

4.3.2 Effects of different suction temperatures on suction coefficients. The suction coefficient is the ratio of actual suction volume flow (which was measured by flow meter at suction end) and theoretical stroke volume. Figure15 shows that the experimental and the simulated values of suction coefficients decreased as the suction temperature decreases. It can be seen that the simulation value can agree well with the measured value. The suction coefficient decreased from 0.825 to 0.615 with the decrease of suction temperature from -54.2℃ to -142.2℃. The reason of this change is that the suction coefficient is the product of volume coefficient, pressure coefficient and temperature coefficient, The volume coefficient and pressure coefficient changed little for different conditions altered the suction temperature only. So the suction coefficient keeps consistent with the temperature coefficient.

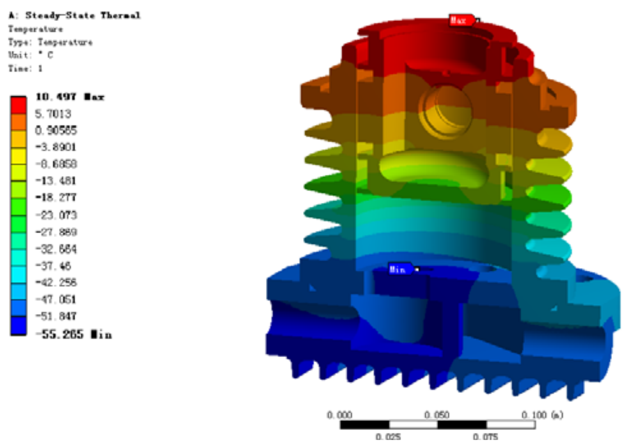

Figure 14: FEM results of temperature distribution at low suction temperature (-88℃)

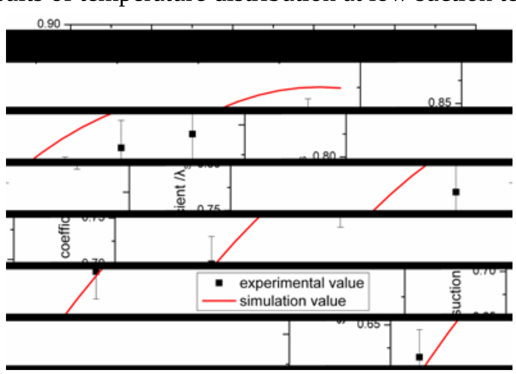

Figure 15: Simulated and measured suction coefficient

4.3.3Effects of different suction temperatures on temperature coefficients .Pressure coefficient λ_p and volumetric coefficient λ_v can be got by the P-V diagram which was measured in this experiment, so the temperature coefficient λ_T can be calculated according to the Formula (1). Figure16 shows the experimental and simulation values of temperature coefficient decreasing with the suction temperature. From the Figure8, we can see that the temperature coefficient λ_T decreased from 0.96 to 0.72 by 25% with a decrease in suction temperature from -73.5℃ to -142℃, When the suction temperature was above -73.5℃, the temperature coefficient is close to 1. So the suction heating increases with the decreasing of the suction temperature.
4.3.4 Effects of different suction temperature on volumetric coefficient. The volumetric coefficient is the ratio of practical discharge amount (which was measured by flow meter on discharge end) and theoretical stroke volume. Figure13 shows the experimental and simulation results of volumetric coefficient, and that the volumetric coefficient decreased from 0.8 to 0.61 with the decrease in suction temperature from -54.2℃ to -142.2℃. Both the experimental and simulated results provided a tremendous direct while designing stroke volume of BOG compressor.

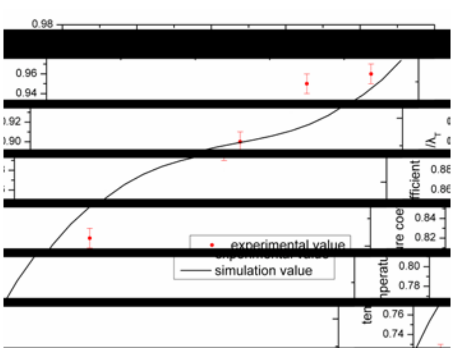

Figure 16: Simulated and measured temperature coefficient

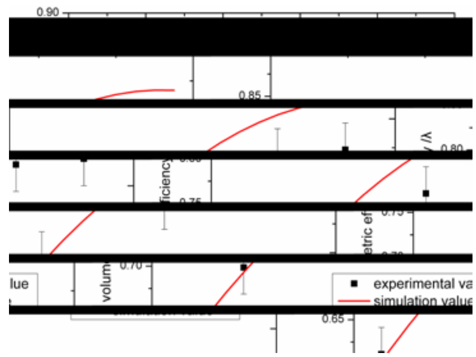

Figure17: Simulated and measured volumetric coefficient

5. Conclusions

When the BOG compressor operated steadily with a suction temperature of -162℃, the temperatures in the suction chamber were relatively low, with an average temperature of -109℃. Temperatures in the discharge chamber were relatively higher, with an average temperature of -60℃, and temperatures in the compression chamber were in the range from -60℃ to -109℃.
The largest temperature difference between the different parts of the cylinder occurred in the first ten minutes after start, instead of in the final stable state. Pre-cooling the cylinder is probably necessary to reduce the temperature difference.
The maximum stress at low suction temperature was 354MPa,which occurred at the thread of the nut and the bolt engaging on suction side while the value was only 270Mpa at the ambient temperature. The maximum displacement at low suction temperature was 1.117mm.
The suction coefficient decreased from 0.825 to 0.615 with the decrease of suction temperature from -54.2℃ to -142.2℃. The temperature coefficient decreased from 0.96 to 0.72 with a decrease of suction temperature from -73.5℃ to -142℃.The above two coefficients made a great contribution to the decrease of volumetric coefficient, and the volumetric coefficient decreased from 0.8 to 0.61 with the decrease of suction temperature from -54.2℃ to -142.2℃. Both the experimental and simulated results provided a guide to design of BOG compressors.

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ACKNOWLEDGEMENT

This work was supported by the National Natural Science Foundation of China (Research Project 51106120/E060105).