Experimental study on performance of BOG compressor

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Abstract: The boil-off gas (BOG) compressor is widely used for recycling the excessive boil-off gas of liquefied natural gas (LNG), and the extra-low suction temperature brings about great challenges to design of the BOG compressor. In this paper, a test system was built to examine the effects of low suction temperature on the compressor performance, in which the lowest temperature reached -178℃ by means of a plate-fin heat exchanger with liquefied nitrogen. The test results showed that, as the suction temperature decreased from 20℃ to -150℃, the volumetric efficiency of the compressor dropped by 37.0%, and the power consumption decreased by 10.0%. The preheat of the gas by the pipe through the suction flange to suction valve was larger than 20℃ as the suction temperature was -150℃, and this value increased with the decreased suction temperature. The pressure loss through the suction valve at lower suction temperature was larger than that at ambient temperature while the volume flow rate was kept the same.

Keywords: BOG compressor, volumetric efficiency, power consumption, preheat

1. Introduction

Natural gas(NG) is one of the most promising clean energy sources and plays a more and more important role in our daily energy consumption. NG is usually stored and transported in the form of compressed natural gas(CNG) and liquefied natural gas(LNG). After liquefied, LNG is only 1/625 of NG in terms of volume at atmosphere pressure, and considering the physical nature of NG, it is obvious the most economical way for transportation of NG over long distances is in the form of LNG. While usually stored in containers at the low temperature of about -162℃,it is inevitable that heat transfers from surroundings to the cryogenic LNG, which results in continuous vaporizing BOG(0.05%/day, mass fraction)[1-3]. In the LNG terminals, BOG generation drastically increases when LNG ship unloads. The excessive BOG in the containers gradually increases the pressure of the container. Therefore, there must be effective way to deal with BOG [4]. The best two choices of recycling BOG are directly compressing to the distribution pipelines and liquefying, and the latter is more widely used to recover BOG due to its 30-60% higher energy-utilization efficiency[5]. A typical BOG recycle system in LNG terminal is shown in Fig.1[6], in which the BOG is liquefied in a BOG reliquefier after pressurized by the BOG compressor. BOG compressor is an essential part of dealing with the BOG in this system.

BOG compressors normally work at extra low suction temperature of about -162℃， and require operating at an oil-free condition. At present, the BOG compressors in LNG terminals are classified as either vertical labyrinth compressors [7] or horizontal piston ring compressors [8].The suction temperature of BOG compressor are between -162℃ and -100℃,and suction pressure between 0.003 and 0.01MPa(gauge pressure)[9]. The compressor usually has two or three stages, and the pressure ratio distribution is different with compressors working at ambient temperature, the pressure ratio of first stage is higher than the others. On this condition , the discharge temperature of the first stage of BOG compressor can arise up to -50℃,which requires less attention on the design of the rest stage.

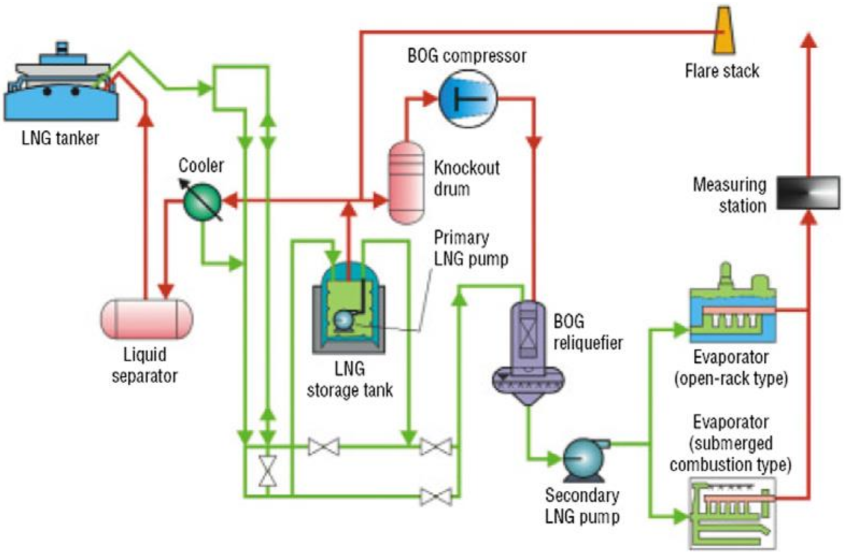

Figure 1. LNG system in a typical LNG terminal[6]

The temperature of BOG in the LNG system can be as low as -162℃ and the effects of such low temperatures must be taken into consideration when designing the BOG compressor. It was proposed that preheat was one of the most important considerations when designing BOG compressors in cryogenic service[6]. K. Murai et al.[10] measured a BOG compressor and provided a correlation to calculate preheat. Yoshida and Katsumi ShinyaKoji[11] researched the development of the suction and discharge valves using non-metallic valve plates for BOG compressors, Shen et al.[12] built a 3-D finite element model of BOG compressor to study the heat transfer in the cylinder of BOG compressor and the temperature distribution of the cylinder .The results of the simulation proved to be an effective way to acquire the temperature distribution of BOG compressor. Chen et al.[13] investigated the effect of low suction temperature on flow rate.

In the above investigations, preheat was always included. Because, at low temperatures the heat transfer from the cylinder to the gas affects the thermodynamic process in the compression chamber. This directly determines the design method and the parameters of BOG compressors. The large temperature difference between the low-temperature suction chamber and the surrounding environment leads to thermal stresses on the cylinder, influencing the structure and materials of the cylinder and the heat treatment process. As the piston is in direct contact with the freezing 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 compression chamber are critical issues in the design of BOG compressors. Rare research were done on the characteristics of BOG compressor,

As mentioned above, preheat is one of the most important design parameter of BOG compressor because it strongly affects the actual flow rate of BOG compressor. There is some research involving the experimental testing to measure preheat, however in those research either the structure of tested compressor is totally different with practical applied BOG compressor or the empirical correlation contains unknown parameters. In this paper, a BOG compressor was designed to investigate preheat and corresponded flow rate at different suction temperature and pressure, and the pressure-time diagram inside the cylinder was measured. To meet this demand, a testing system (shown in Fig.2) was set up to obtain different suction temperature and pressure. In this testing system the suction gas was firstly cooled by changing heat with liquid nitrogen through a plate-fin heat exchanger, and then it flowed through an evaporator and was accurately adjusted to designed temperature.

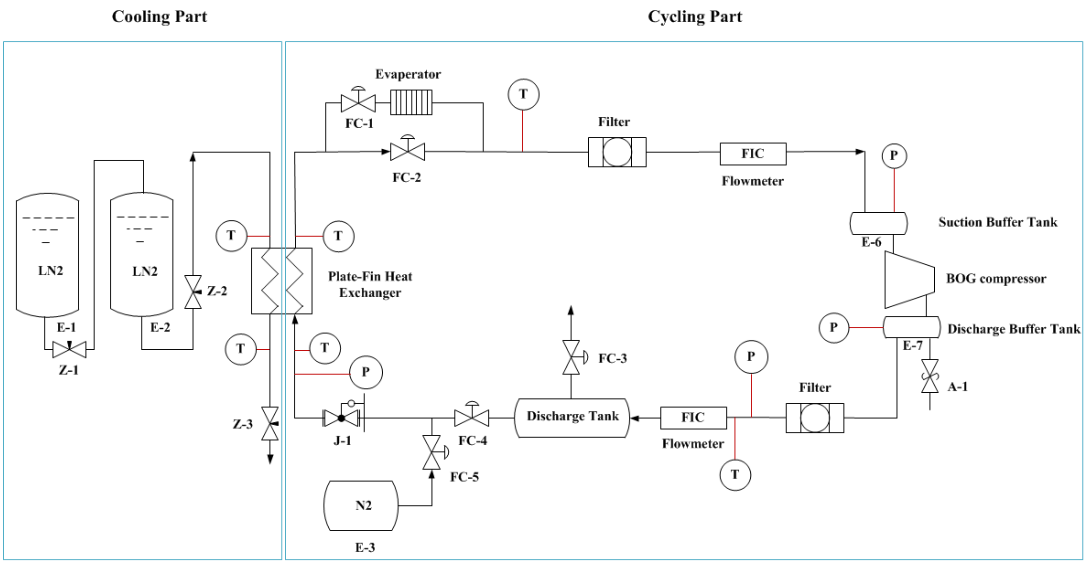

Figure 2. The Testing system of the BOG compressor

2. BOG compressor and testing system

2.1 BOG compressor

The tested BOG compressor, as shown in Fig. 3, was a L-type double-acting reciprocating unit with two cylinder running at 411rpm(6.8Hz).Only the cylinder of first stage was used in this experiment considering of the discharge temperature of the first stage was higher than -45℃. The working medium in the compressor is nitrogen. The diameter of first stage cylinder is 300mm, and the piston is 299mm.The nominal swept volume was 700.53m^3/h. The tested suction temperature is between -160 ℃ and -40 ℃. The suction pressure is between -0.03MPa to 0.11MPa and the discharge pressure is between 0.15MPa and 0.25MPa. The all connecting bolts around the cylinder were made of austenitic stainless steel.

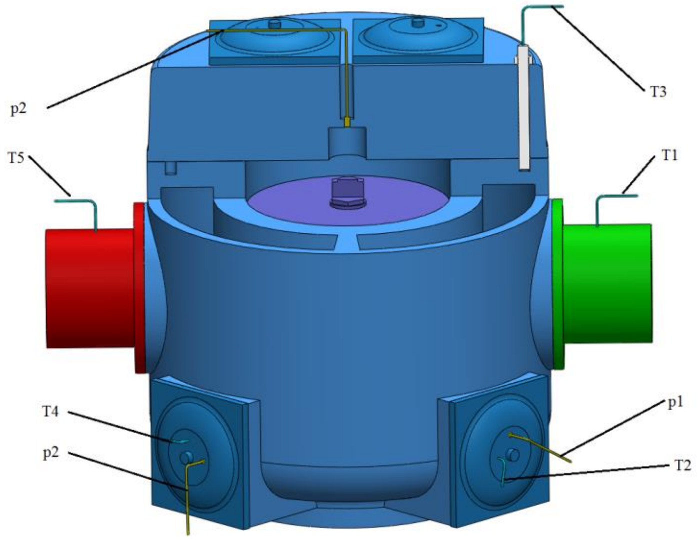

Figure 3. The BOG compressor studied

2.2 BOG compressor testing system

A testing system was set up to obtain different suction temperature and pressure, as shown in Fig2 .This system contains two parts: the cooling part and the cycling part. The cooling part mainly includes several liquid nitrogen containers which was connected to a plate-fin heat exchanger. The cycling part is made up of compressor, discharge buffer tank, discharge tank, flow meters, nitrogen tank, plate-fin heat exchanger, evaporator, filter and suction buffer tank.

In the testing system the nitrogen, after pressurized by BOG compressor, flows through the plate-fin heat exchanger and is cooled by liquid nitrogen flowing through the heat exchanger, then the cooled nitrogen flows through evaporator and is adjusted to different suction temperature. As the operating pressure of plate-fin heat exchanger is relatively low, pressure of nitrogen is decreased by a reducing valve. The nitrogen tank in the cycling part is used to replace the air inside the compressor, tanks and pipelines before start-up, and supply nitrogen when the discharge pressure increases after start-up. For the temperature of nitrogen is low toward -178℃ when flowing out of the heat exchanger, it is necessary that the pipelines, flow meter and suction buffer tank are covered with heat preservation cotton.

3. Experimental method

3.1 Experimental facilities

As shown in Fig 1, five thermocouples with an accuracy of ±0.2℃ were arranged to measure temperature around the cylinder. Four thermocouples were installed to measure the gas temperature around the cylinder, including the suction flange（T1）, suction valve（T2， shown in Fig. 4）, discharge valve(T4) and discharge flange(T5). The fifth thermocouple was installed to measure the temperature at the end of bolt that connected the cover head and the cylinder on suction side(T3). The thermocouple of T2 was used to measure the gas temperature before flowing through the suction valve. Three pressure sensors were installed in the cylinder of the compressor to measure the pressure of the suction chamber(p1), the pressure inside the cylinder(p2) and the pressure in the discharge chamber(p3). All the pressure sensors displayed the absolute pressure with an accuracy of ±0.25%. Two target flow meters were installed before suction buffer tank and after the discharge buffer tank. The accuracy of the flow meter was ±1%. The power was measured with a three-phase wattmeter with an accuracy of ±1%.

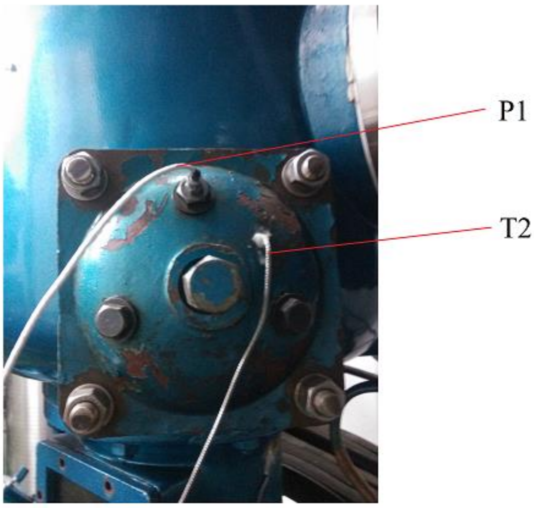

Figure 4. The pressure sensor and thermocouple installed on the suction side

3.2 p-v indicator diagram

Figure 5. The pressure sensor measuring the pressure inside the cylinder

An experimental study on recording the p-v diagram was conducted. In order to measure the transient pressure inside the cylinder changes with the rotation angle, a miniature pressure sensor was adopted. Considering the practical compressor structure, the pressure sensor was installed as shown in Fig. 5. The pressure sensor was embedded in the cover head and the signal wire was led out. The signal was collected and analyzed by the dynamic signal acquisition and analyzing system. The sensor had a uncertainty of less than ±0.1% and its response frequency was 1400 kHz. The maximum working pressure was 0.7MPa with an output voltage of 100mv, and the range of working temperature was between -196℃ and 120℃. A top dead center sensor (TDCS) was also installed and the signal was diminished so that the maximum of output signal was a little higher than the pressure sensor's which allowed a much more convenient observe during the experiment..

3.3 Dynamic data acquisition system

The dynamic data acquisition system was based on an acquisition card with a high acquisition speed. The sampling frequency of the acquisition card was 1 MHz and the analog input was up to 16 simultaneous ways. The maximum sampling frequency of a single channel was 250 kHz. The signal of pressure sensors, TDCS and thermocouples were input to the conditioning device by means of 8 synchronous channels. After undergoing amplification and conditioning, both the pressure and temperature signals were collected by the high speed acquisition card and displayed on computer screen. The structure of the dynamic data acquisition system is shown as fig. 6. Fig. 7 showed the graphic display of acquisition system which was programmed by LabVIEW.

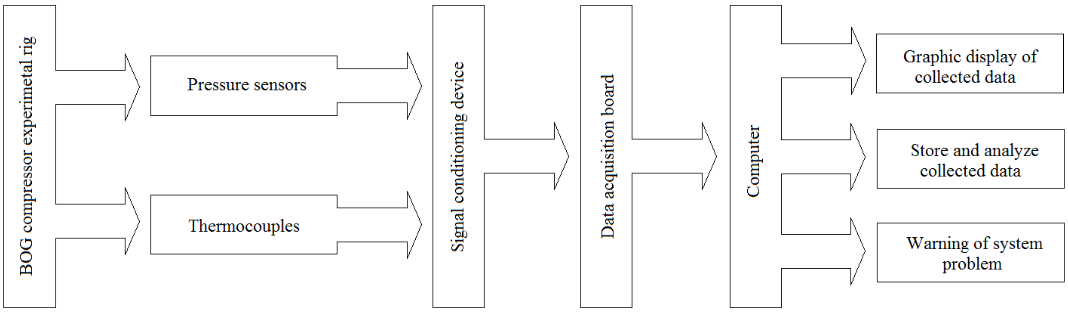

Figure 6. Dynamic data acquisition system

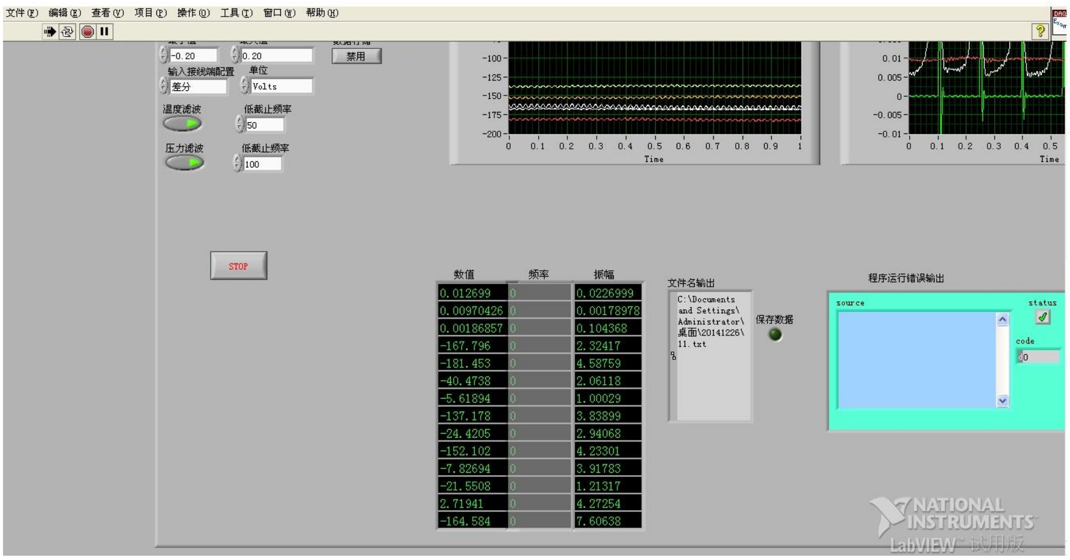

Figure 7. graphic display of acquisition system

4. Results and discussions

By adjusting the suction temperature, the temperature distributions of the cylinder were obtained, allowing for analyzing how greatly preheat influenced the flow rate of BOG compressor. Tests were performed using different suction pressures and pressure ratios. The pressure-time diagrams inside the cylinder at different suction temperatures were analyzed and were compared with that of ambient suction temperature in order to find out the difference of compression and expansion process between BOG compressor and ordinary compressor of ambient suction temperature.

According to the experimental results of the temperature distribution, the preheat increased with the decrease of suction temperature and increased with the increase of pressure ratio. The pressure-time diagrams at low suction temperature showed the suction pressure loss were higher than those of ambient suction temperature which resulted in the increase of the actual pressure ratio and power consumption and the decrease of flow rate. In addition, the flow rate and power consumption were recorded and the specific power consumption were calculated. The calculated results showed the power consumption of low suction temperature is less than that of ambient suction temperature.

4.1 Effects of different suction temperatures on BOG compressor

Suction pressure was kept constant at 0.1MPaA (Absolute pressure) and discharge pressure was kept 0.22MPaA, while the suction temperature was adjusted to -50℃, -70℃, -90℃, -110℃, -130℃ and -150℃. Fig.8 showed a picture of BOG compressor working steadily at a suction temperature of -150℃.The outer surface of the cylinder was covered with frost, and the frost of suction side was thicker than that of the discharge side.

Figure 8. The cylinder of the BOG compressor at suction temperature of -150℃

Fig.9 illustrated the volumetric efficiencies and isentropic efficiencies of the BOG compressor under different suction temperature. The volumetric efficiencies decreased from 76% to 51%and the total isentropic efficiency decreased from 60% to 46% when the suction temperature decreased from -50℃ to -150℃, while these two values were 81% and 74% at suction temperature of 20℃. Fig.10 showed the temperature distributions of all measured points around cylinder and preheats at different suction temperatures. The temperature of T2, T3, T4 and T5 decreased with the decrease of suction temperature. When the suction temperature was -50℃, the temperature of T2, T3, T4 and T5 was

-45℃, 42℃, 39℃ and -1℃, respectively. The preheat, i.e. temperature difference between T1 and T2, was 5℃, and the value was 22℃ when the suction temperature is -150℃. When the suction temperature decreased from -50℃ to -70℃, the temperature diagrams of T4 and T5 shared a cross point. That was because heat transferred from cylinder to discharged gas when discharged temperature decreased rapidly. The temperature at end of bolt on the suction side was -39℃ while the suction temperature was -150℃, then the temperature of other end of bolt embedded into the cylinder could be between the suction temperature and temperature of suction valve, i.e. between -150℃ and -128℃, thus the temperature of the whole bolt was obtained, which contributes to the pretension design of bolts.

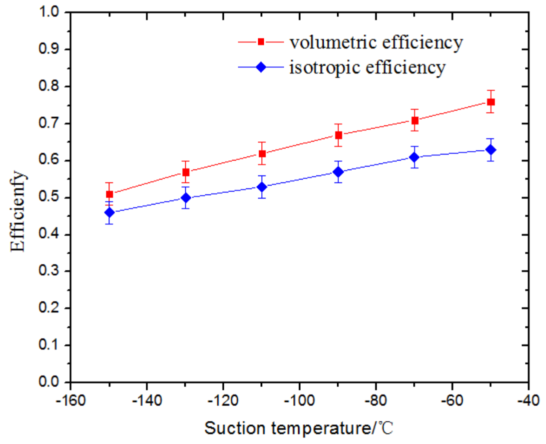

Figure 9. The volumetric efficiency and isentropic efficiency at different suction temperature

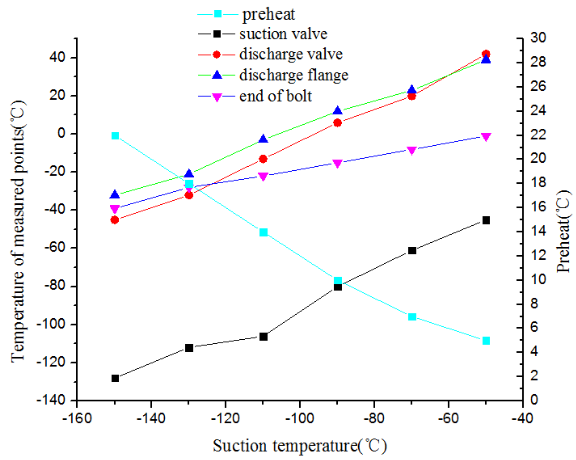

Figure 10. Temperature of measured points and preheat at different suction temperature

When the suction temperature was 20℃ and -150℃, the discharged flow rate was kept to 446.64Nm^3/h, while the power consumption was 17.06kW and 15.35kW, the power consumption decreased 10.02%. This meant it was more economical to pressurize at a lower suction temperature despite the decrease of volumetric efficiency and isentropic efficiency.

The p-v diagram was showed in fig.11 while the suction temperature was -150℃. The suction pressure loss consisted 17% of the power consumption, and the discharge pressure loss consisted 13%.

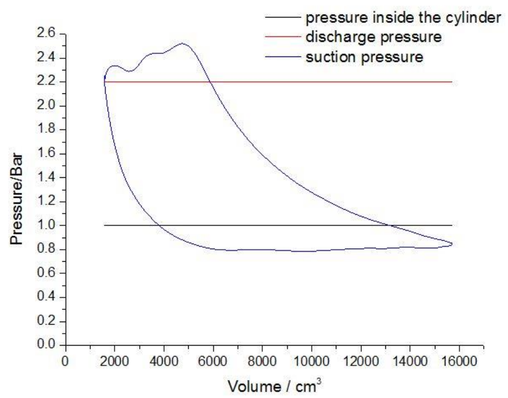

Figure 11. The p-v diagram under suction temperature of -150℃

4.2 Effects of different discharge pressure on BOG compressor

Suction pressure was kept constant at 0.1MPaA and suction temperature was kept -150℃, while the discharge pressure was adjusted to 0.15MPaA, 0.2MPaA, 0.25MPaA and 0.3MPaA. As shown in Fig. 12, volumetric efficiencies decrease from 72% to 48% and the total isentropic efficiencies decreased from 69% to 0.54% with the increase of discharge pressure from 0.15MPaA to 0.3MPaA. While the suction temperature was 20℃, the discharge pressure changed from 0.15MPaA to 0.3MPaA, the volumetric efficiency decreased from 78% to 61%, and isentropic efficiency decreased from 73% to 65%.

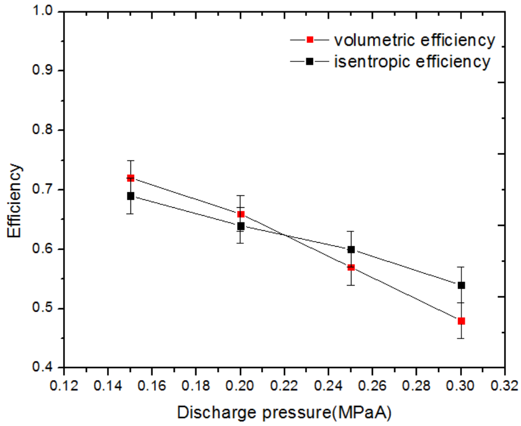

Figure 12. Volumetric efficiency and isentropic efficiency at different discharge pressure

5. Conclusions

A testing system was set up to measure the performance of BOG compressor, and the compressor worked steadily at the suction temperature low to -150℃. In this condition, the effects of low suction temperature on BOG compressor were examined.

When the compressor operated steadily with a suction temperature of -150℃,the temperature around the suction valve was about -128℃.Temperature around the discharge valve was relatively higher, with a temperature of -45℃.

The preheat was 5℃ with a suction temperature of -50℃,and it kept increasing to 22℃ while the suction temperature was -150℃.

The volumetric efficiencies decreased from 76% to 51% and the total isentropic efficiencies decreased from 6% to 46% when the suction temperature decreased from -50℃ to -150℃, while these two values were 81% and 74% at suction temperature of 20℃. The power consumption decreased 10% when the suction temperature decreased from 20℃ to -150℃.

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