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Experimental Research on Indicator Diagrams of a Water Lubricated Screw Air Compressor

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Experimental research on indicator diagrams of a water-lubricated twin-screw air compressor

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ABSTRACT

In recent years, water-lubricated twin-screw compressors have attracted more and more attention, because this type of compressors can produce the high-quality air completely free of oil and there is no contamination. However, poor sealing and lubricating properties of water might result in different operating characteristics compared with those of oil-injected compressors. A good understanding of the operating mechanism is essential in any attempt to increase the performance of the compressor, which can be achieved by means of the indicator diagram. In this paper, a prototype of the water-lubricated twin-screw air compressor was developed and a test rig was established. A series of pressure transducers were arranged in consecutive positions of the casing to measure the pressure distribution inside working chamber of the compressor. In addition, volumetric efficiency, adiabatic efficiency and mechanical efficiency of the compressor under different operating conditions were calculated based on measured volumeric flowrate and power consumption. As a result, the influence mechanism of rotating speed, discharge pressure and water injection flow-rate on the compressor performance and working process was analyzed based on indicator diagrams.

1. INTRODUCTION

The twin-screw compressor is a positive displacement machine that contains a pair of intermeshing rotors in the casing to achieve compression. Twin-screw compressors have been widely used as the air compressor because of their easy operation, well adaptability, high efficiency and excellent running reliability. In a screw compressor, liquid is always deliberately injected into the compression chamber to enhance sealing, lubrication, noise reduction and cooling effect. Oil is the most common choice, but on some occasions the compressed gas provided by the compressor must be clear and free of oil. Therefore, some measures have to be taken to avoid oil leakage into the compressor chamber, either by fitting sealing systems or by absolutely avoiding the use of oil. In recent years, water becomes a very potential option to be used as the lubricant and the coolant. Especially, with the development and applications of water-lubricated journal bearings, conventional oil-lubricated bearings can be replaced, and thus compressed gas completely free of oil can be provided by this type of compressor, which is called water-lubricated twin-screw compressor. Water as an auxiliary fluid offers sufficient potential for investigation.

Nowadays, little research is focused on the water-lubricated twin-screw compressor, and only several researchers have studied the general water-injected twin-screw air compressor from different perspectives. For the general water-injected twin-screw compressor, bearings are still lubricated by oil or grease, and a pair of synchromesh gears are always designed in the compressor in order to avoid two rotors contacting each other. Madhav et al. designed a family of water injected compressor system in the power range of 15 to 315 kW, to compare performance and economic aspects of water injected twin-screw compressor system with traditional oil-free compressor package and oil-injected twin-screw compressor in the multi-filtration system. Based on these findings, they suggested that water injected twin-screw compressor systems are economically viable and reliable, and thus could become preferred option for delivering oil free air (Madhav and Kovacevic, 2015). Li et al. (2009) presented a mathematical model of working process in a water-injection twin-screw compressor used in the PEM fuel cell systems, and theoretically analyzed the effect of rotation speed and volumetric flow rate of water. In addition, the performance of the developed twin screw compressor was tested by the experiment. Ous et al. (2012) adopted water injection to humidify the reactant air and cool the fuel cell stacks as well as the air twin-screw compressors. Moreover, the temperature and the relative humidity of the air at suction and exhaust of the compressor were monitored under constant pressure and water injection rate and at variable compressor operating speeds, and thus the power of the compressor and fuel cell stacks were calculated and analyzed. Yang et al. (2016) experimentally investigated performance characteristics of process-gas twin-screw compressor under different working conditions by varying rotational speed, pressure ratio, and the amount of injected water. The actual process gas was replaced by air in the process-gas twin-screw compressor test rig due to the restrictions of experimental condition.

In the above literatures, only Li Jian Feng et al. investigated the indicator diagrams of the water-injected twin-screw air compressor, but they were restricted to theoretical investigation. Shen et al. (2014) measured the *p-t* diagram of a water injected twin screw compressor, but the working fluid in the compressor was water vapor, and the related research was still not sufficient. In addition, Alexander and Andreas (2016) experimentally investigated indicator diagrams of a water injected twin-screw expander and analyzed relevant influence mechanisms on the expander's operational behavior resulting from water injection. As a kind of screw-type machines, twin-screw expanders have the nearly same structure and the reverse working process compared with twin-screw compressors. Some conclusions about twin-screw expanders can also provide some reference to the investigation of twin-screw compressors.

In this paper, in order to have a good understanding of the operating mechanism of water-lubricated twin-screw compressors, a series of pressure transducers were arranged in consecutive positions of the casing to measure the *p-t* diagrams under different rotating speed, discharge pressure and water injection flow-rate, and then the measured *p-t* diagrams were transformed into indicator diagrams. Based on the indicator diagrams, relevant influence mechanisms on the compressor's operating characteristics resulting from water injection were analyzed in detail.

2. EXPERIMENTAL SET-UP

2.1 Test Rig

As shown in Figure 1, an air compressor test system was designed and constructed to investigate the performance characteristics of the developed water-lubricated twin-screw compressor. It shows the schematic diagram of the test rig which consists of the compressor, electric motor, water injection system and data measurement system.

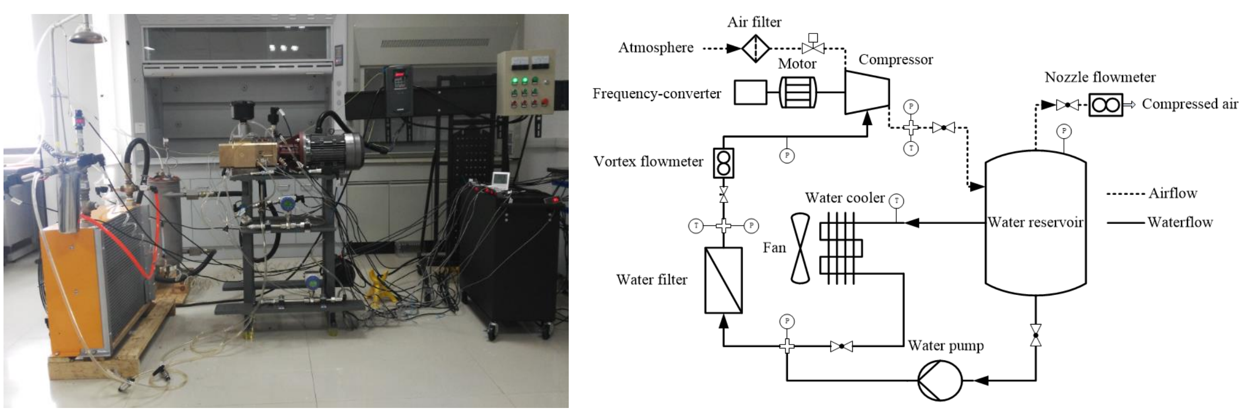

Figure 1: Photograph and schematics of the experimental test rig

The main parameters of the specially designed water-lubricated twin-screw compressor driven by the variable speed electric motor are listed in Table 1. The rotor profile with the gear ratio of 4:5 was generated by the SCCAD software which is designed by Xi'an Jiaotong University. Water was selected as the only substance injected into the compressor for the purpose of cooling, lubrication, seal and noise reducing. The presence of water is associated with scaling and corrosion of materials and causes their deterioration. Hence, the special attention was given to water treatment in order to maintain water quality. In addition, corrosion was avoided by providing a special corrosion-free coating on the rotors and using corrosion-free material of albronze to manufacture the casing. Due to the poor lubricity of water compared to oil, the water-lubricated journal bearings made of a kind of plastic was adopted to ensure their wear-resistance.

Table 1. Main parameters of the compressor.

Item	Value
Outside diameter of the male rotor (mm)	75
Outside diameter of the female rotor (mm)	60
Center distance between the rotors (mm)	53
Length of the rotors (mm)	120
Built-in volume ratio	4.5
Theoretical displacement per rotation (cm^3)	310.21

The water injection system consists of a water reservoir with a water separator, a fan assisted water cooler, a water filter, a water pump and control valves. The stored water in water reservoir is driven by the pressure difference between the water reservoir and working chamber or bearing chamber. After cooled by the water cooler and filtrated via the water filter made of stainless steel, water from water reservoir enters the compression chamber, the suction and discharge journal bearings, respectively. It deserves to be mentioned that water is pumped into the journal bearings for several seconds to prevent from dry running before starting the compressor.

2.2 Data Measurement

The system is well instrumented in order to analyze the performance of the water-lubricated twin-screw compressor. The static temperature and pressure of air and water at different locations in the system were measured by the same temperature sensors and pressure transducers, respectively. However, the atmospheric temperature and pressure were monitored by a thermometer with an uncertainty of 0.1℃ and a barometer with an uncertainty of 0.1 kPa. The volumetric flowrate was measured by a standard ASME nozzle flowmeter with a nozzle diameter of 19.05 mm. The water injection flowrate could be adjusted by the controlling valve and measured by the vortex flowmeter on the injected water pipe. The power consumption of the compressor was measured by an air compressor function analyzer created by AIRbetter in Suzhou, China. The variable rotating speed of the male rotor was controlled and obtained by a frequency-converter. Detailed information about the measurement instruments and their accuracy is listed in Table 2.

Table 2: Measurement instruments adopted in the compressor test rig

Parameter	Measurement instrument	Full scale	Accuracy
Temperature	Pt100	0~100 ℃	±0.5% of full scale
Static pressure	BP801	0~2.5 MPa	±0.5% of full scale
Water injection flowrate	Vortex flowmeter	3~20 L/min	±0.5% of full scale
Power consumption	Air compressor function analyzer	1~350 kW	2% of measured valve

The measurement of *p-V* indicator diagrams was conducted by six CYG1505ALLF pressure transducers produced by Kunshan Shuangqiao Sensor Measurement Controlling Co., Ltd. The maximum working pressure is 2 MPa and the range of working temperature of the transducer is from -40 to +120ºC. The overall measurement error of the transducers is less than 0.5% of full scale. As shown in Figure 2, five pressure transducers, No.1-5, are installed in the casing of compressor to measure the *p-V* diagrams and the pressure transducer No.6 is installed in the discharge cavity. The transducer No.1 can measure the pressure of both part of the suction process and part of the compression process, and the transducer No.5 can measure the pressure of both part of the discharge process and part of the compression process. The other three transducers are merely for the measurement of part of the compression process.

Figure 3 shows the measured data of *p*-*t* diagram at rotating speed of 1800 rpm and with discharge pressure of 0.8 MPa(A). A complete *p*-*t* curve is composed of partial data collected from different pressure transducers, No.1~No.5. By the transformation between the rotating angle of rotor and the volume of working chamber, the *p*-*t* diagram can be converted into the *p*-*V* indicator diagram. It is impossible to arrange sensors to obtain the complete discharge process due to the limit of space, so the pressure transducer No.6 is installed in the discharge cavity. It can be observed from Figure 3 that it is reasonable to assume the constant pressure loss at discharge process which can't be measured.

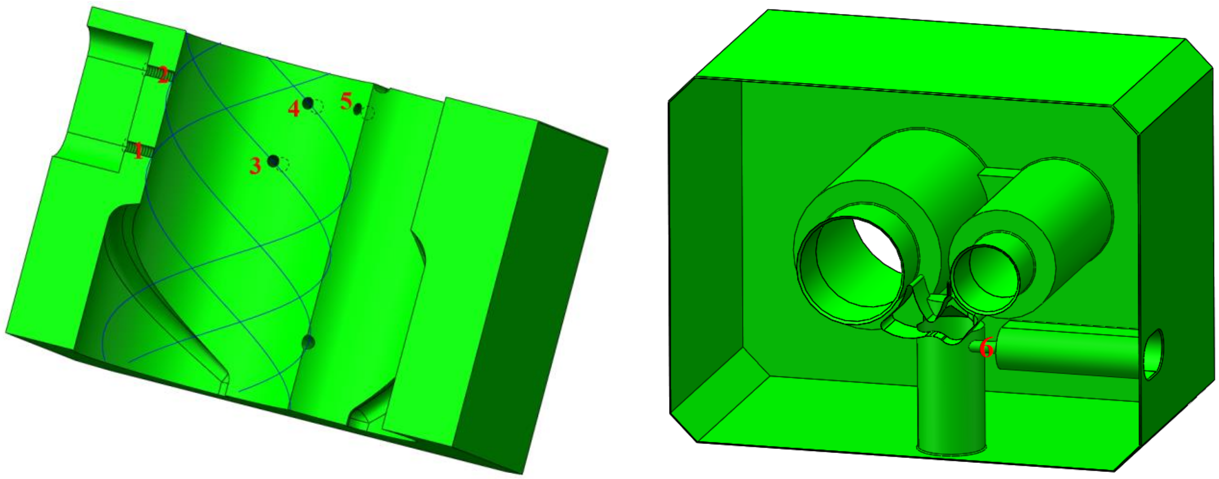

Figure 2: Positions of pressure transducers in the casing and discharge cavity

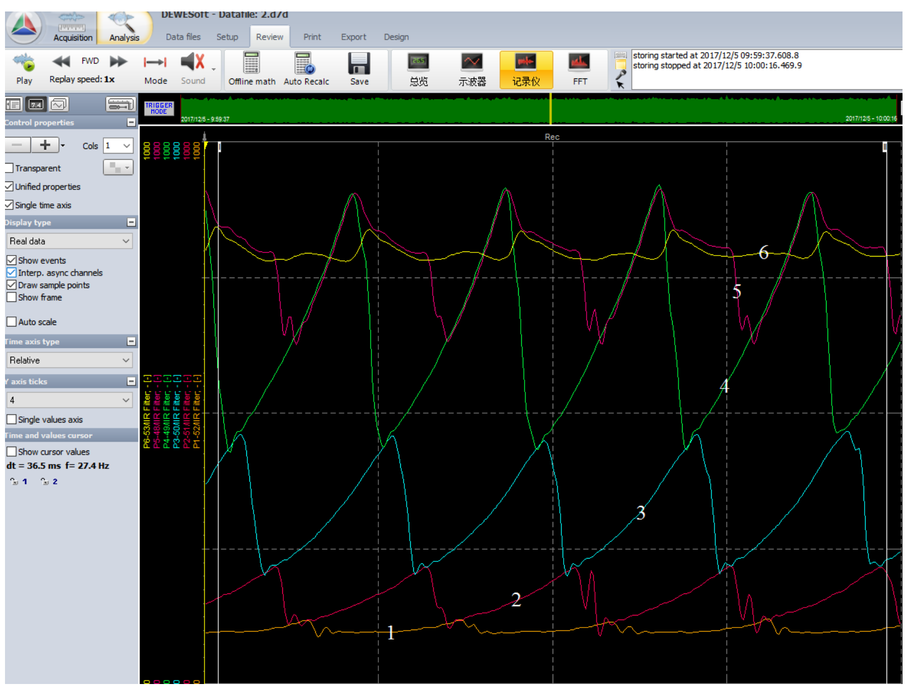

Figure 3: Measured data of *p*-*t* diagram on the condition of 1800 rpm/0.8 MPa(A)

2.3 Performance Evaluation

Some compressor performance parameters such as temperature, pressure, flowrate and power consumption can be obtained directly from the sensors, while other performance parameters including volumetric efficiency, adiabatic

efficiency and mechanical efficiency require to be calculated from the measured data.
The volumetric efficiency of the compressor is defined as the ratio of actual volumetric flowrate to theoretical volumetric flowrate, which can be represented as

$$\eta_V = q_V / q_{Vt} \tag{1}$$

The adiabatic efficiency of the compressor is defined as the ratio of adiabatic power to the actual shaft power, which can be calculated by the following equation:

$$\eta_{\mathrm{ad}} = P_{\mathrm{ad}} / P_{\mathrm{s}} \tag{2}$$

The adiabatic power, P_{ad}, can be calculated by the air enthalpy at the inlet and outlet of the twin-screw compressor on the condition of the adiabatic compression process. It can be expressed as:

$$P_{\mathrm{ad}} = q_{\mathrm{m}}\left(h'_{\mathrm{dis}} - h_{\mathrm{suc}}\right) \tag{3}$$

The mechanical efficiency of the compressor is defined as the ratio of indicated power to the shaft power, as shown below:

$$\eta_{\mathrm{m}} = P_{\mathrm{ind}} / P_{\mathrm{s}} \tag{4}$$

where P_{ind} is the indicated power related to the lobe number and rotating speed of the male rotor, which can be represented as:

$$P_{\mathrm{ind}} = -zn\oint pdV \tag{5}$$

3. RESULTS AND DISCUSSION

3.1 Influence of rotating speed and discharge pressure

To evaluate the influence of rotating speed on the performance of the water-lubricated twin-screw air compressor, the ambient temperature and pressure were nearly kept constant, 15 ℃ and 104 kPa, respectively, while the rotating speed is varied from 1800 rpm to 5400 rpm for the experiment.

Figure 4 shows the measured *p*-*V* indicated diagrams of the twin-screw compressor under different rotating speed and discharge pressure. The suction pressure is almost the same on each condition, about 96 kPa, which implies that the pressure loss at suction process is about 8 kPa and hardly change with rotating speed and discharge pressure. In addition, throttling loss at discharge process increases at the higher rotating speed, due to the higher flow velocity through the discharge port. Moreover, the *p*-*V* indicator diagrams will approach the isothermal working process, with the increase of the rotating speed. At the rotating speed of 1800 rpm, most pressure at compression process of the measured indicated diagram exceeds that of the adiabatic process. It is likely that the leakage in the compressor would increase with the decreasing rotating speed and leakage has a more important influence than heat exchange at low rotating speed. It can also be seen from Figure 4 that discharge pressure mainly affects the latter part of compression line and the effect is more obvious at the lower rotating speed. In addition, the compression line deviates the isothermal line with the increase of discharge pressure. Mainly because higher discharge pressure results in a larger amount of leakage in the latter part of compression process.

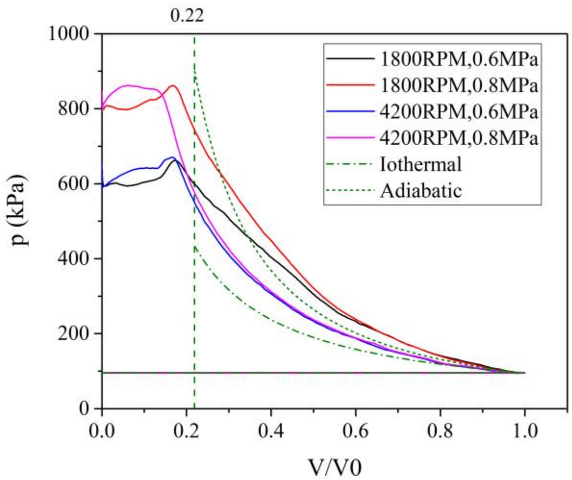

Figure 4: *p-V* diagrams under different rotating speed and discharge pressure

Figure 5 shows the influence of rotating speed on compressor efficiencies including volumetric efficiency, adiabatic efficiency and mechanical efficiency, when the discharge pressure is kept constant of 0.8 MPa(A). As shown in Figure 5, volumetric efficiency of the water-lubricated twin-screw compressor increases with the rotating speed, but the increment rate falls down. When rotating speed increases from 1800 rpm to 5400 rpm, volumetric efficiency increases from 60.64% to 81.22%. This is because the theoretical mass flowrate of the compressor is proportional to the rotating speed for a given twin-screw compressor at constant operating condition. Meanwhile leakage mass in a working cycle decreases with increasing rotating speed due to shorter leakage time. It can also be seen that adiabatic efficiency and mechanical efficiency increase first and then decrease with rotating speed. For the present compressor, the maximum adiabatic efficiency of 74.36% occurs at the rotating speed of 4800 rpm and the maximum mechanical efficiency of 90.58% occurs at the rotating speed of 3600 rpm. It deserves to be mentioned that the effect of injected water occupying the working volume is doesn't be taken in consideration, or else the mechanical efficiency would be lower. Both efficiencies represent the range of internal losses in the compressor, including solid friction between rotors, in the bearings and contact seals, as well as hydraulic losses due to water injection. With the increase of rotating speed, the friction losses increase rapidly, but the pump power of water nearly keeps constant and is the key factor consuming power.

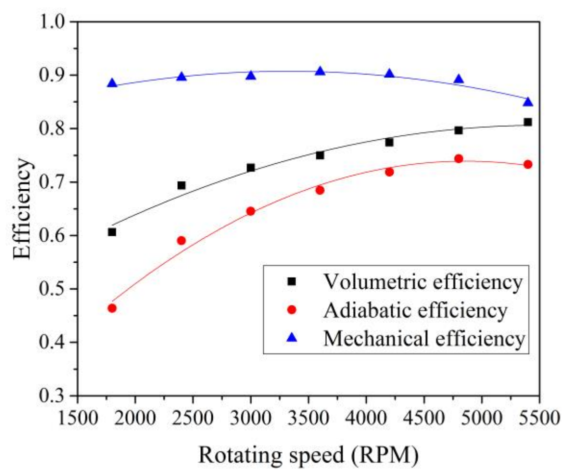

Figure 5: Efficiencies under different rotating speed

Figure 6 shows the influence of discharge pressure on compressor efficiencies at the rotating speed of 4200 rpm. The volumetric efficiency declines from 78.7% to 77.3% when discharge pressure increases from 0.6 MPa(A) to 0.9 MPa(A). On the one hand, rising discharge pressure leads to the larger pressure difference between the discharge cavity and the working volume, and thus increases the leakage; on the other hand, rising discharge pressure results in more water injected into the working volume, decreasing the leakage. At 4200 rpm, the first factor has the dominant effect. With regard to adiabatic efficiency and mechanical efficiency, a nearly linear increase with discharge pressure can be observed. As Figure 4 shows, the compressor at rotating speed of 4200 rpm is in the stage of under-compression. In addition, discharge pressure mainly affects the discharge process and has few effects on the compression process. Higher discharge pressure means larger driving power in the discharge process. Meanwhile, higher discharge pressure also results in larger pump power of injected water, but the increase in driving power of air in the discharge process is larger than that in pump power of injected water.

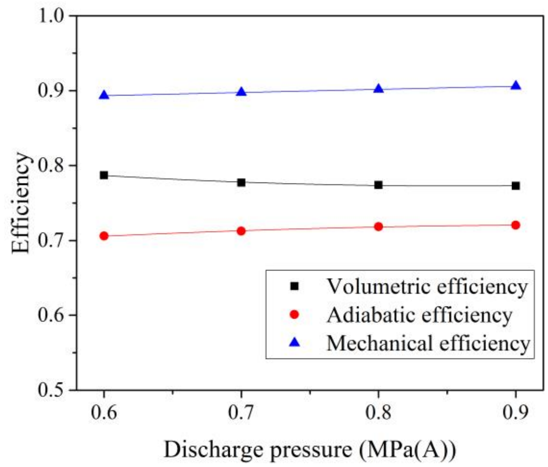

Figure 6: Efficiencies under different discharge pressure

3.2 Influence of water injection flow-rate

In order to investigate the influence of water injection flowrate on the performance of the compressor, the *p-V* diagrams and performance parameters were measured on the condition that the water injection flowrate varied from 2 L/min to 10L/min at rotating speed of 3000 rpm. The water injection flowrate only refers to the flowrate of water injected into the rotor chamber by a hole in the casing and doesn't include backwater of bearings and seals. The total backwater flowrate of bearings and seals for the experiment is kept constant of 12 L/min.

Figure 7 shows the measured *p-V* indicated diagrams of the twin-screw compressor under different water injection flowrate. It can be seen that the working process of the compressor will be close to the isothermal process with the increase of the water injection flowrate. It may be that rising water flowrate decreases the air temperature by enhancing the heat exchange between water and air. Consequently, the pressure will reduce slightly in the compression process with the increase of the water flow rate.

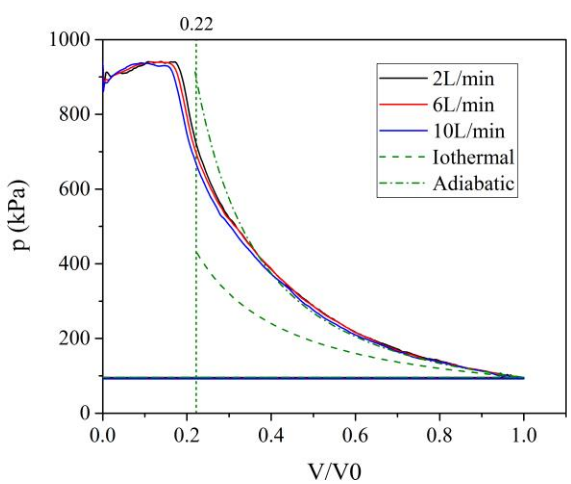

Figure 7: *p-V* diagrams under different water injection flowrate

Figure 8 shows the influence of water injection flowrate on compressor efficiencies. A nearly linear increase in volumetric efficiency up to the water injection flowrate of 6 L/min can be observed. Afterward, the growth flattens out due to enough water to seal the clearance of compressors. When water injection flowrate increases from 2 L/min to 10 L/min, the volumetric efficiency increases from 62.49% to 66.24%. With regard to adiabatic efficiency, it has the similar tendency with volumetric efficiency and small variations of adiabatic efficiency in the range between 56.82% and 58.87% can be observed. One reason for this could be the increase in volumetric efficiency leads to the increase in adiabatic power. In addition, mechanical efficiency decreases almost linearly with the increasing water injection flowrate. It is likely that indicated power determined by the indicated diagram decreases as water injection flowrate increase, while hydraulic losses would increase, especially in pump power of water.

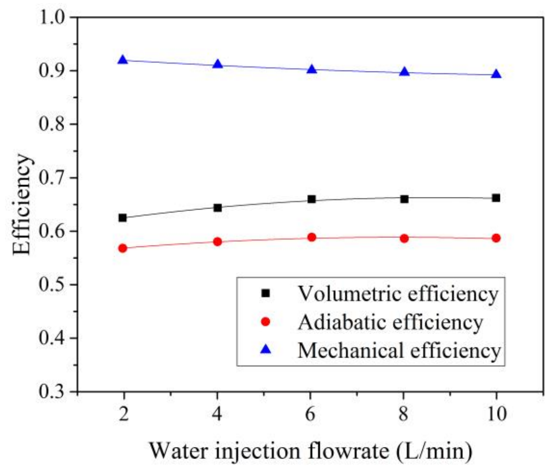

Figure 8: Efficiencies under different water injection flowrate

4. CONCLUSIONS

A prototype of the water-lubricated twin-screw air compressor was developed in this paper and a comprehensive experimental research was carried out to evaluate the operating characteristic of the compressor under various conditions. The *p*-*V* diagrams were measured by a series of pressure transducers in order to provide a good understanding of the operating mechanism for the compressor. In addition, the influences of rotating speed, discharge pressure and water injection flowrate on the compressor performance including volumetric efficiency, adiabatic efficiency and mechanical efficiency were investigated experimentally and analyzed based on indicator diagrams. The following conclusions can be drawn from the results of the investigation:

- With the increase of the rotating speed, *p*-*V* indicator diagrams will be close to the isothermal working process. Volumetric efficiency of the compressor increases at a falling increment rate with the increasing rotating speed, and the maximum value can reach 81.22% at rotating speed of 5400 rpm. Moreover, adiabatic efficiency and mechanical efficiency increase first and then decrease as rotating speed increases.
- Discharge pressure mainly affects the latter part of compression line in indicated diagrams and the effect is more obvious at the lower rotating speed. In addition, the compression line deviates the isothermal line with the increase of discharge pressure. The volumetric efficiency declines when discharge pressure increases. With regard to adiabatic efficiency and mechanical efficiency, a nearly linear increase with discharge pressure can be observed
- The working process of the compressor will approach the isothermal process with the increase of the water injection flowrate. A nearly linear increase in volumetric efficiency and adiabatic efficiency up to the water injection flowrate of 6 L/min can be observed, and then the growth flattens out. In addition, mechanical efficiency decreases almost linearly with the increasing water injection flowrate.

The results of this study may contribute to evaluating the performance of the water-lubricated twin-screw air compressor and providing reference for the design of the compressor in the future development. In addition, an important direction for future work might be to develop the simulation model of working process, in order to analyze the heat and mass transfer characteristics in the compressor.

NOMENCLATURE

H	specific enthalpy	(kJ/kg)
n	rotating speed	(rpm/s)
P	power	(kW)
q_m	mass flow rate	(kg/s)
q_v	volume flow rate	(m^3/h)
z	lobe number	(–)
η	efficiency	(–)
Subscript		
ad	adiabatic	
dis	discharge	
ind	indicated	
m	mechanical	
s	shaft	
suc	suction	

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