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Experimental Research on the Effects of Suction Ports on Twin Screw Expander Performance

Yongning He [1], Huihong Wu [1], Yeqiang Zhang [1], Tingxiang Jin [1] and Ziwen Xing [2,*]

[1] School of Energy and Power Engineering, Zhengzhou University of Light Industry, Zhengzhou 450002, China; heyn@zzuli.edu.cn (Y.H.); wuhuihong1996@126.com (H.W.); zhangyeqiang@zzuli.edu.cn (Y.Z.); txjin@126.com (T.J.)
[2] School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an 710049, China
* Correspondence: zwxing@mail.xjtu.edu.cn

Abstract: An organic Rankine cycle (ORC) system is an efficient technology for generating power from low-grade thermal sources. Twin screw expanders are widely used in ORC systems. Pressure loss caused by a suction port influences the suction process and the system performance. In this paper, the effects of suction ports on twin screw expander performance are experimentally investigated. The pressure loss value of R245fa is measured under different working conditions, and its effects on the suction coefficient are discussed. Results show that a lower rotation speed results in a lower pressure loss value and a higher suction coefficient under certain working conditions. Increasing the suction pressure leads to a higher pressure loss value but a lower pressure loss ratio. Discharge pressure has little effect on the suction process.

Keywords: organic Rankine cycle (ORC); twin screw expander; suction port; performance

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Highlights

1. The effects of a suction port on twin screw expander performance are discussed.
2. An experimental investigation is carried out with different rotation speeds, suction pressures, and discharge pressures.
3. Lower rotation speeds result in lower pressure loss values under certain working conditions.
4. With certain suction ports, a higher suction pressure causes a higher pressure loss value.

1. Introduction

Coal, oil and natural gas consumption have increased all over the world with economic development and improvements in the average standard of living. High levels of fossil fuel consumption have caused significant environmental pollution and negative effects on human health. The sustainable development of society requires cleaner and more efficient energy. New forms of energy, such as solar energy, ocean energy and nuclear energy, have gained much attention over the last decade. The usage of low-grade thermal sources has become a popular topic for researchers all over the world. Many studies have been carried out on the recovery and utilization of energy from low-grade thermal sources [1]. Advanced new thermoelectrics may be a key technology that can be used to transform heat directly into power. Recent advances in enhancing the performance of novel fiber/fabric-based flexible thermoelectrics demonstrate an exciting direction for the development of wearable electrocardiographic systems [2,3]. However, this technology cannot meet the demand for heat recovery in high heat capacity situations. The organic Rankine cycle (ORC) is another technology used [4] for heat–power transformation.

The ORC system uses organic working fluid, such as refrigerant, instead of water in a traditional steam Rankine cycle. It is suitable for various thermal sources and has a larger capacity. It is appropriate for recovering excess heat from many kinds of heat sources, such as geothermal energy, solar energy, and industrial waste heat. The thermodynamic cycle, working fluids, expander, and heat exchanger all affect the performance of the ORC system. The expander used in the ORC system is the most important piece of equipment, and there are many kinds of expanders. At present, single-screw, twin screw, scroll and turbine expanders are mainly used [5–8]. Screw expanders are suitable for medium-scale ORC systems. The best expander performance is obtained with a rotation speed of 3000 rpm. In terms of fluids, R123 and R245fa are frequently used for ideal operation pressure and system performance. The maximum power output is about 300 kW [9]. Twin screw expanders have advantages similar to twin screw compressors, such as high reliability, multi-phase transportation and good dynamic balance. The calculation of geometric characteristics can be performed according to the twin screw compressor method.

An ORC's performance is affected by the temperature of the heat source and thermal power fluctuations. A fluctuating heat source results in a reduction in the isentropic efficiency of the expander and, consequently, a reduction in system efficiency [10]. Bianchi et al. presented numerical investigations of a twin screw expander for low-grade (≤100 °C) heat-to-power conversion applications based on the bottoming Trilateral Flash Cycle [11]. An analysis of the effects of different inlet qualities of the R245fa working fluid and of the revolution speed on the expander's performance was carried out. Different aspects of the performance of the twin screw expander, such as rotational speed, torque and output power, are all influenced by the inlet pressure of the expander [12]. Wang et al. designed a detailed contrast experiment on three expanders and investigated the effects of the suction port area and the rotor length on the expander's performance [13]. By analyzing the measured p-θ diagrams of twin screw expander, the features of the working process were identified [14].

Based on previous research, geometric construction has a significant effect on the performance of twin screw expanders. Among the various factors, suction ports play a significant role in the suction process. In this paper, an experimental investigation into the pressure loss caused by suction ports is carried out. The effects of suction ports on the pressure loss value, the suction process and performance are discussed.

2. The Suction Process of a Twin Screw Expander

The working process of a twin screw expander is shown in Figure 1. The working volume, formed by the rotors and housing, increases with the rotation of the rotor. The expansion of the high-pressure working fluid causes the rotation and outputs the power. The suction port connects the working volume to the high-pressure pipe, which plays an important role at the beginning of the expansion process and then in the performance of the expander. As shown in Figure 2, a twin screw expander has a small suction port and flow area.

There is pressure loss when the high-pressure working fluid flows through the small flow area suction port. Pressure loss at the suction port causes the actual expansion process to fall short of the ideal expansion process.

To evaluate the influence of pressure loss at the suction port, we use the pressure loss value and the ratio of pressure loss to suction pressure:

$$\Delta p = p_{\mathrm{s}} - p_{\mathrm{b}} \tag{1}$$

where Δp is the pressure difference of p_{s} and p_{b}, p_{s} is the suction pressure, $\mathrm{p_b}$ is the pressure at the beginning of the expansion process and p_{d} is the discharge pressure.

α is the ratio of pressure loss to the suction pressure and is used to evaluate the relative pressure loss.

$$\alpha = \frac{\Delta p}{p_{\mathrm{s}}} \tag{2}$$

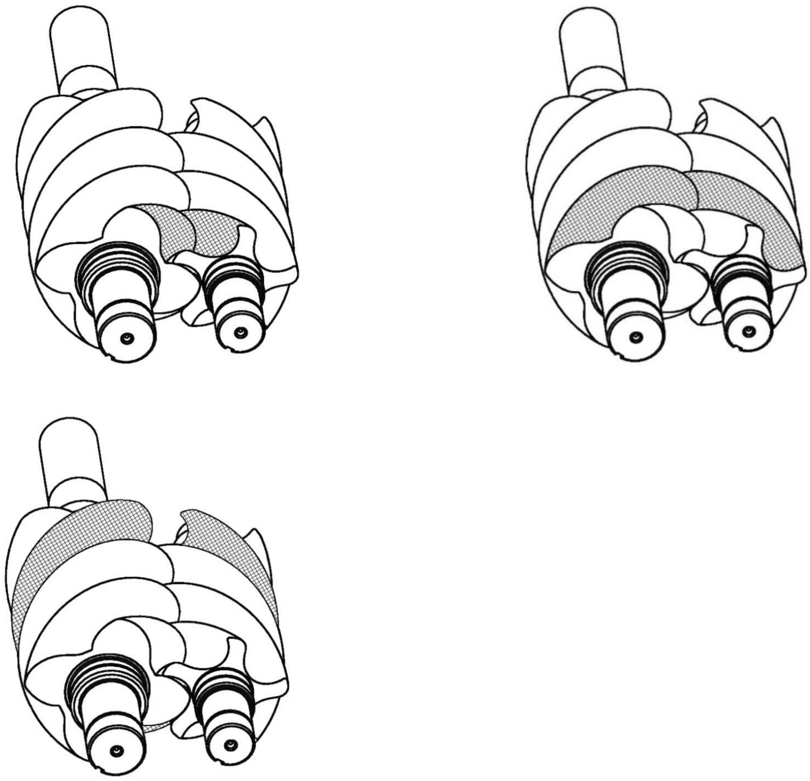

Figure 1. Working process of a twin screw expander.

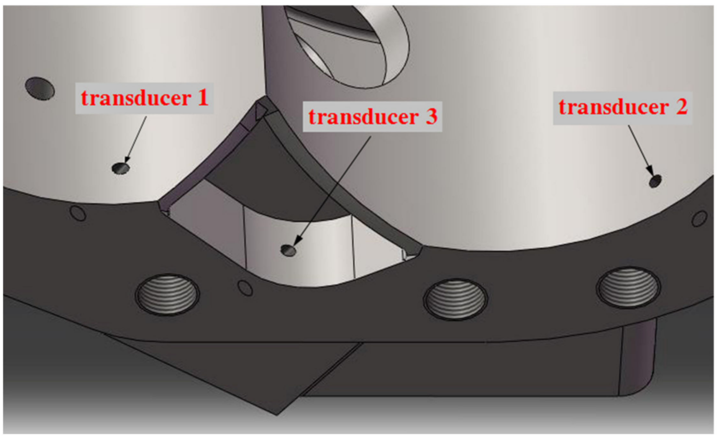

Figure 2. Suction port of a twin screw expander.

As shown in Figure 3, pressure loss during the suction process has significant effects on the expansion pressure.

In addition to the volumetric efficiency used in the evaluation of compressor performance, we consider the suction coefficient that is used to evaluate the influence of the pressure loss. The suction coefficient is defined as follows:

$$\eta_s = \frac{m_{suc}}{m_{ideal}} \quad (3)$$

where m_{suc} is the real mass flow rate considering the pressure loss and m_{ideal} is the mass flow rate with the ideal process.

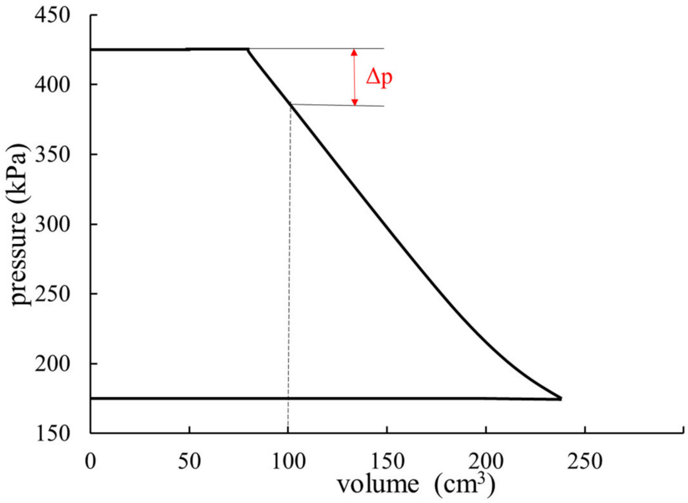

Figure 3. Pressure loss at suction port displayed in the *P*-*v* diagram.

3. Test Rig and Experimental Parameters

3.1. Test Rig

Based on the theoretical analysis, experimental research was carried out. The ORC system and test rig are shown in Figure 4. R245fa was used as a working fluid. Steam was used as the heat source, and a plate heat exchanger was used as the evaporator. The evaporation temperature reached as high as 85 °C. The condenser was cooled by a cooling tower, and the condensation temperature was 30 °C. R245fa evaporated in the evaporator and then flowed into the expander with suction pressure. High-pressure R245fa expanded in the expander and then outputted the power. The oil separator was on the discharge side of the expander. Separated oil returned into the expander, and R245fa with discharge pressure flowed into the condenser, where it was cooled by low-temperature water and then diverted into a liquid reservoir. A working fluid pump was used to increase the R245fa liquid pressure. All the connection pipes and heat exchangers were well insulated to avoid heat exchange with the environment.

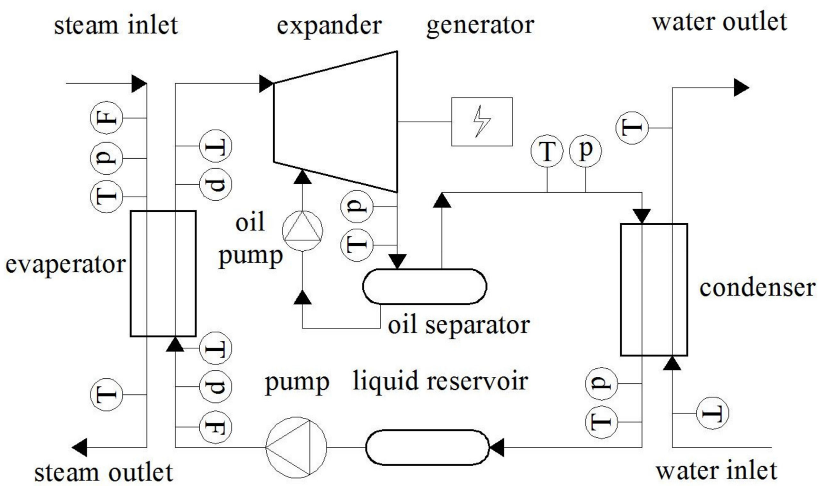

Figure 4. ORC system and test rig.

3.2. Experimental Parameters

With different rotation speeds, suction pressures and discharge pressures, an experimental analysis of the performance of the twin screw expander was carried out. The suction pressure was controlled by the evaporation pressure, and the discharge pressure was controlled by the condensation pressure. The water flow rate and the temperature were adjusted to change the evaporation pressure and the condensation pressure. The flow rates of the working fluid and water were monitored, as were the temperature and pressure. Instrumentation and propagated uncertainties are listed in Table 1. The pressure was measured by the pressure transducers. Three pressure transducers were set at the suction port, and their positions are shown in Figure 2.

Table 1. Instrumentation and propagated uncertainties.

Parameter	Instrument	Full Scale	Accuracy
Heat source temperature	Pt-100	0~150 (°C)	±0.1
Working fluid temperature	K-type thermal couple	−200~400 (°C)	±0.5
Cooling water temperature	K-type thermal couple	−200~400 (°C)	±0.5
Suction and discharge pressure	Piezoresistance type transducer	0~1.0 (MPa)	±0.25%
Other pressure	Piezoresistance type transducer	0~2.5 (MPa)	±0.25%
Mass flow rate	Coriolis flowmeter	0.1~0.76 (kg/s)	±0.5%

3.3. Error Analysis

The accuracy of the sensors is presented in the above section. The systematic errors can be calculated from the error propagation [15] as expressed by:

$$w_R = \left[\sum_{i=1}^{j} \left(\frac{\partial R}{\partial x_i} w_{x_i} \right)^2 \right]^{\frac{1}{2}}$$

where w_R is the resultant uncertainty and $w_1, w_2, \ldots, w_\mathrm{n}$ are the uncertainties of the independent variables. R is a given function of the independent variables $x_1, x_2, \ldots, x_\mathrm{n}$. Uncertainties for Δp, α and η_s are ±0.25%, ±0.35% and ±0.5%, respectively.

4. Results and Discussion

4.1. Effects of Rotation Speed

With a suction pressure of 442 kPa and a discharge pressure of 200 kPa, Δp and α variations according to rotation speed are shown in Figure 5. With the rotation speed increased from 1400 rpm to 1600 rpm, Δp through the suction port increased from 47.1 kPa to 62.8 kPa. A higher rotation speed meant a shorter suction time and, consequently, a higher pressure loss. α increased from 0.106 to 0.142. η_s variations with rotation speed are shown in Figure 6. With the rotation speed increased from 1400 rpm to 1600 rpm, η_s decreased from 0.965 to 0.912. The performance of the twin screw expander with different rotation speeds indicated that, under certain working conditions, a lower rotation speed decreases the Δp and increases η_s.

4.2. Effects of Suction Pressure

With a discharge pressure of 220 kPa and a rotation speed of 1600 rpm, Δp and α variations according to suction pressure are shown in Figure 7. With the suction pressure decreased from 500 kPa to 440 kPa, Δp through the suction port decreased from 55.8 kPa to 52.8 kPa. With a constant flow area, higher suction pressure led to a higher pressure loss. Future twin screw expander designs should consider a larger flow area to enable higher suction pressures.

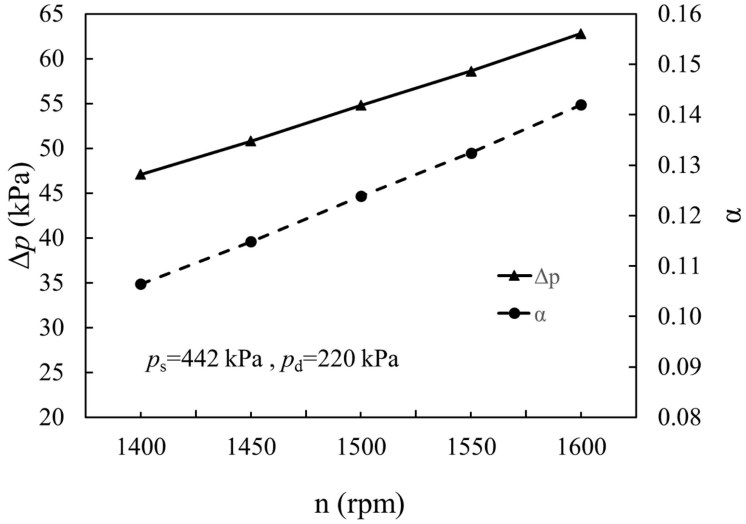

Figure 5. Δp and α variations with rotation speed.

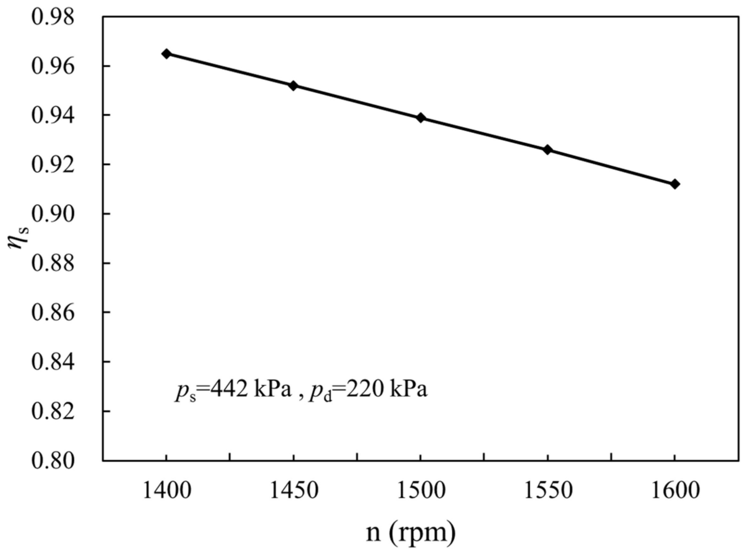

Figure 6. η_s variations with rotation speed.

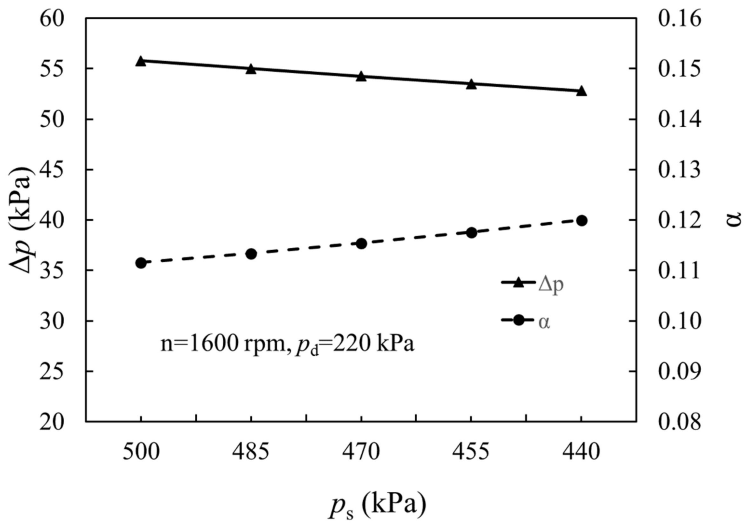

Figure 7. Δp and α variations with suction pressure.

α increased from 0.112 to 0.120. Higher suction pressure resulted in a larger Δp but smaller α. The suction coefficient variations with suction pressure are shown in Figure 8. When the suction pressure decreased from 500 kPa to 440 kPa, η_s increased from 0.905 to 0.915.

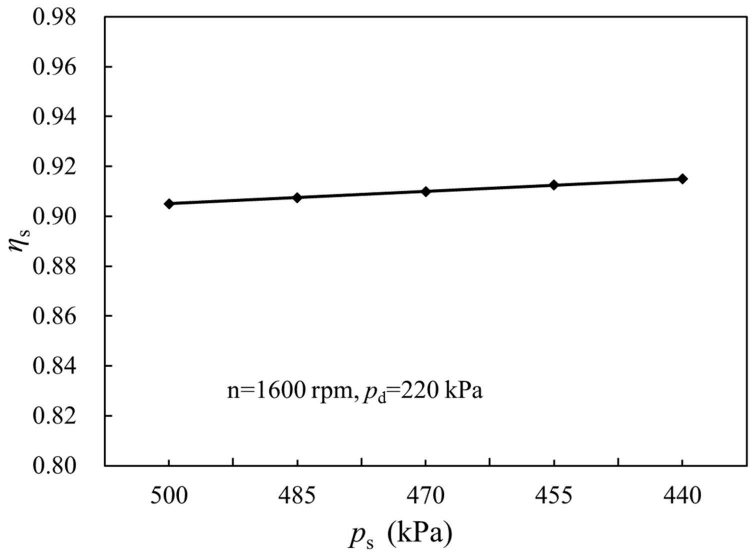

Figure 8. η_s variations with suction pressure.

4.3. Effects of Discharge Pressure

With a suction pressure of 460 kPa and a rotation speed of 1400 rpm, Δp and α variations according to discharge pressure are shown in Figure 9. With the discharge pressure increased from 180 kPa to 260 kPa, Δp and α remained at 50 kPa and 0.109, respectively. This result shows that suction pressure and rotation speed determine the pressure loss of the expander. Variations in discharge pressure did not change the R245fa properties on the suction side. η_s variations with discharge pressure are shown in Figure 10. The values were between 0.955 and 0.958. The peak value occurred at the discharge pressure of 220 kPa.

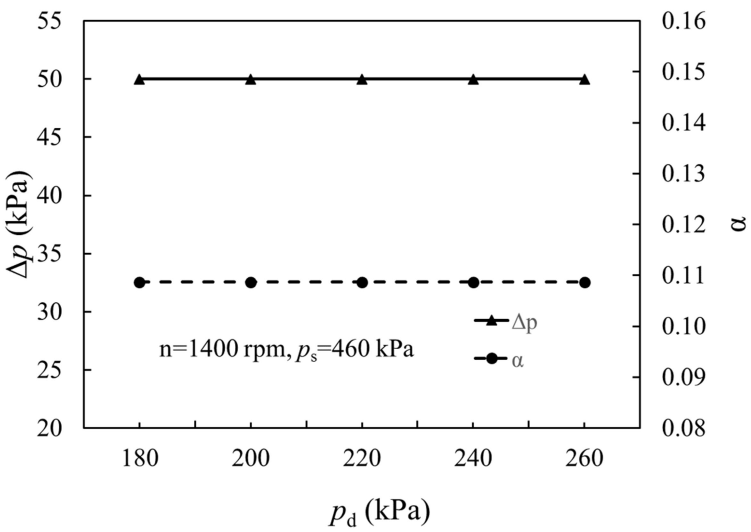

Figure 9. Δp and α variations with discharge pressure.

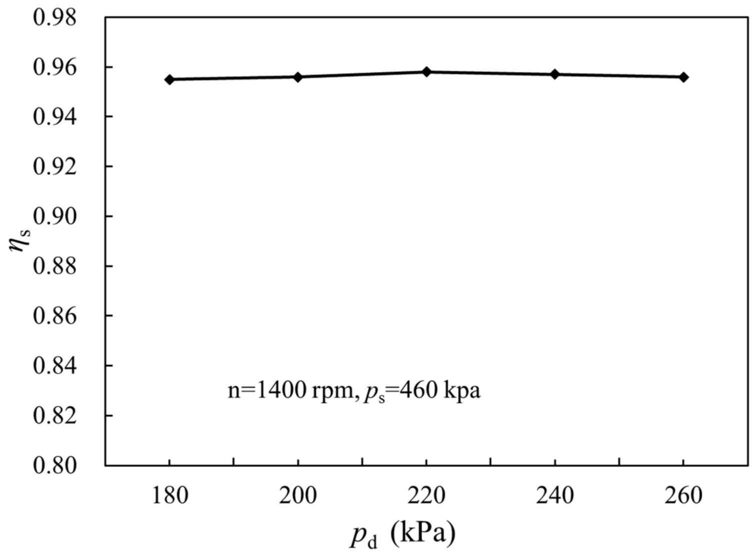

Figure 10. η_s variations with discharge pressure.

5. Conclusions

With different rotation speeds, suction pressures and discharge pressures, an experimental investigation into the performance of a twin screw expander was carried out. The pressure loss value, the ratio of pressure loss to suction pressure, and the suction coefficient were used to evaluate the influence of the suction port. Conclusions are as follows.

Lower rotation speeds decreased the pressure loss and increased the suction coefficient. With constant suction and discharge pressure, increasing the rotation speed increases the pressure loss value and pressure loss ratio. With a rotation speed of 1600 rpm, the pressure loss value and pressure loss ratio were 62.8 kPa and 0.142, respectively. The suction coefficient decreased from 0.965 to 0.912 when the rotation speed increased from 1400 rpm to 1600 rpm.

Higher suction pressure increases the pressure loss value with a constant suction port. With a constant rotation speed and discharge pressure, increasing the suction pressure increases the pressure loss value but decreases the pressure loss ratio and the suction coefficient.

With a constant rotation speed and suction pressure, discharge pressure variations did not change the suction process. The pressure loss value and the ratio of pressure loss to suction pressure had constant values. The suction coefficient showed small variations with the discharge pressure.

The results and conclusions in this paper are given based on a specific twin screw expander with a constant suction area. The pressure loss value cannot be used directly in other twin screw expanders. Further research should be carried out to guide future expander designs.

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