Experimental study on expansion ratio of single screw expanders under variable working conditions

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Abstract: In the waste heat recovery utilization at medium and low temperature, the organic Rankine cycle(ORC) was regarded as the most potential technique. The expander played an important role as the output power equipment of the ORC, and single screw expander (SSE) had a great prospect due to its unique structure and working characteristics. Besides expander efficiency, expansion ratio was another important performance parameter, which has a great influence on the thermal efficiency of the ORC system. Based on the SSE prototype made by ourselves, the experimental study on expansion ratio of SSEs was carried out. From experimental result, at the same inlet pressure, internal expansion ratio increased with the increasement of rotational speed, and the external expansion ratio decreased gradually. At the same rotational speed (3000rpm), internal and external expansion ratio of increased with the increasement of inlet pressure. When the inlet pressure interval was 0.4~0.65 MPa, the external expansion ratio was 3.26~ 3.73, and the internal expansion ratio was 4.8~ 8.0. It can be found that internal expansion ratio was always higher than external expansion ratio, and under expansion effect was not occurred, even in the working condition of high inlet pressure. It was indicated that the internal flow of SSE was very turbulent during expansion process. So, SSE might have some characteristics of speed expanders. If improving inlet pressure or decreasing back pressure, the higher expansion ratio could be obtained without changing the internal volume ratio of SSEs.

Key words: organic Rankine cycle； single screw expander； internal expansion ratio; external expansion ratio; volumetric efficiency

Introduction

ORC system had natural advantages in low and medium temperature thermal power generation technology [1]. At present, a lot of related research work has been carried out. Kang et al. [2] used R245fa as working fluid, when condensation temperature and evaporation temperature were 30℃ and 80℃ respectively, the adiabatic efficiency of the radial turbine prototype was 78.7%, expansion ratio was 2.65 and thermal efficiency of ORC was 5.22%. Li et al. [3] used R123 as working fluid, when condensation temperature and evaporation temperature were 39.4℃ and 90.7℃ respectively, the adiabatic efficiency of the radial turbine prototype was 58.53%, expansion ratio was 4.19 and thermal efficiency of ORC was 6.15%. Pei et al. [4] used R123 as working fluid, when condensation temperature and evaporation temperature were 35℃ and 105℃ respectively, the adiabatic efficiency of the radial turbine prototype was 63%, expansion ratio was 4.3 and thermal efficiency of ORC was 6.8%. Lei et al. [5] used R123 as working fluid, expander type was single screw expander (SSE), when condensation temperature and evaporation temperature were 67.9℃ and

126.1℃ respectively, the efficiency of SSE prototype was 56%, expansion ratio was 8.5 and thermal efficiency of ORC was 7.98% (considered pump power consumption). Zhou et al [6] used R123 as working fluid, expander type was scroll expander, the maximum efficiency of the expander was 57%, expansion ratio was 7.5 and thermal efficiency of ORC was 8.5%. Obviously, expander played a vital role for ORC system. Improving the efficiency of the expander was beneficial to improving the thermal efficiency of ORC system. When the efficiency of the expander was approximately equal, increasing the expansion ratio could significantly improve the thermal efficiency. low expansion ratio of expanders was the important technical bottleneck of small capacity ORC system utilization.

There are usually two types of expander: speed and volume. The former mainly includes single-stage or multi-stage axial and radial turbines; volumetric expanders have scroll [7], reciprocating piston [8], single screw, twin screw [9], etc. Because SSE had many advantages such as simple structure, high expansion ratio, balanced force and large capacity range, it was one of the most suitable expander types for medium and small capacity ORC system. At present, our research team carried out systematic research work on SSEs, including cycle analysis, structure optimization, irreversible analysis, expander performance and ORC performance, etc [10-16]. In common sense, as a volumetric power machine, internal expansion ratio was different to external expansion ratio, the former depended on structure parameters and the latter depended on working conditions. At present, there were few studies on the internal and external expansion ratios in the literatures. In this paper, the experimental study on the performance of a single screw expander prototype made by our laboratory was carried out, and the influence of working conditions on the expansion ratio was explored. This work could provide basic data and technical support for subsequent performance optimization of SSEs.

1 Definition of expansion ratio

For screw compressors, compression ratio was usually expressed as a function of internal volume ratio and adiabatic exponent. Due to various irreversible losses in actual compression process, such as leakage, friction and heat dissipation, adiabatic exponent was replaced by polytropic exponent. The concept of internal volume ratio can be referred to in reference [17], and it was the ratio of the elementary volume at the end of inspiration to that at the end of compression. The compression ratio was expressed as follows:

$$\varepsilon_{\mathrm{c}} = \tau_{\mathrm{c}}{}^{n} = \left(\frac{V_{\mathrm{s}}}{V_{\mathrm{d}}}\right)^{n} \qquad (1)$$

In the formula, V_{s} was the maximum suction elementary volume, V_{d} was the maximum discharge elementary volume, τ_{c} was internal volume ratio, n was polytropic exponent.

Referring to the concept of compression ratio of screw compressors, expansion ratio was also determined by internal volume ratio and polytropic exponent for SSEs. The expansion ratio was expressed as follows:

$$\varepsilon_{\mathrm{e}} = \tau_{\mathrm{e}}{}^{n} = \left(\frac{V_{\mathrm{e}}}{V_{\mathrm{i}}}\right)^{n} \qquad (2)$$

In the formula, V_{i} was the maximum injection elementary volume, V_{e} was the maximum exhaustion elementary volume, τ_e was internal volume ratio, n was polytropic exponent.

In actual condition, polytropic exponent could be fitted on a lot of experimental data. So, expansion ratio was obtained by measuring the pressure of at the inlet and outlet of expanders in the experiments. For volumetric expanders, expansion ratio could be divided into internal and external. Internal expansion ratio was the ratio of the instantaneous pressure of injection elementary volume to the instantaneous pressure at the end of expansion. External expansion ratio was the ratio of the pressure at inlet pipe to the pressure at outlet pipe. Under different conditions, internal expansion ratio and external expansion ratio were not necessarily equal. The difference was called additional loss, which is referred underexpansion and overexpansion loss. The internal expansion ratio of SSEs was as follows:

$$\varepsilon_i = \frac{p_{in}}{p_{ex}} \qquad (3)$$

In the formula, p_{in} was the injection pressure, p_{ex} was the exhaustion pressure.

The external expansion ratio of SSEs was as follows:

$$\varepsilon_e = \frac{p_i}{p_o} \qquad (4)$$

In the formula, p_i was the pressure at inlet pipe, p_o was the pressure at outlet pipe.

2 Definition of expansion ratio

2.1 Experimental system

The experimental rig mainly tested the performance of the SSE prototype, and chooses compressed air as working fluid. The flow chart was showed in Figure 1.

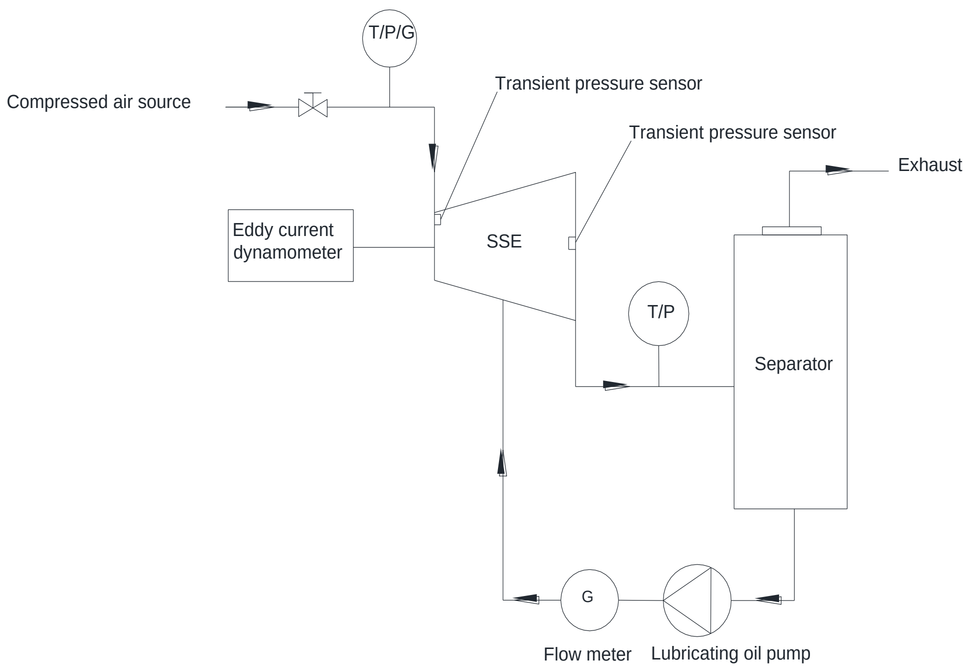

Fig.1 The flow diagram of test rig

The high-pressure air supplied by the compressor injected into the SSE to do work, and the exhausted low-pressure air was led out by the oil-gas separator. The axle power output was measured by the eddy current dynamometer. Various sensors are arranged on the inlet and outlet pipelines of the expander, which are used to monitor the pressure, temperature and flow parameters of the inlet and outlet of the expander. The photo of test rig was showed in Figure 2.

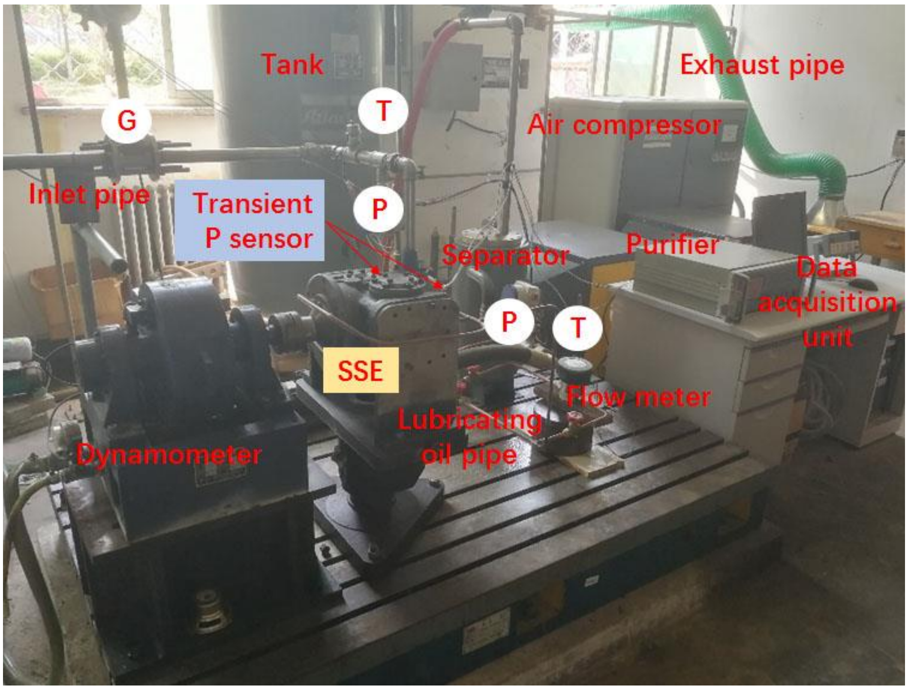

Fig.2 The photo of test rig

The main structural parameters of the prototype were showed in Table 1, and the photo of SSE was showed in Fig.3.

Fig.3 The photo of SSE prototype

Table 1 Main structural parameters of the SSE prototype

Screw diameter	117mm
Gaterotor diameter	117mm
Center distance	93.6mm
Tooth width	17.1mm
Maximum elementary volume	0.0454L
Internal volume ratio	2.95

2.2 Measurement apparatus

Temperature: standard thermocouples of grade A Pt100 were used, model was WZPT-01, the measurement ranges of inlet and outlet were 0 to 100℃ and -100 to100℃ respectively, and accuracy was 0.5℃.

Pressure: two pressure sensors installed on inlet and outlet pipes were used, model was JYB-KO-PAG, the measurement ranges of inlet and outlet were 0 to 2MPa and 0 to 1MPa respectively, and accuracy were 0.2%. Two transient pressure sensors installed on injection port and exhaust port were used, model was CYY8, the measurement ranges of inlet and outlet were 0 to 2MPa and 0 to 1MPa respectively (absolute pressure), and accuracy was 0.5%.

Flow rate: vortex flow meter was used, model was LUGB-25-D, the measurement range was 10 to 70 m^3/h, accuracy was 1.0%。.

During the experiment, the temperature, pressure and flow data are collected by Agilent 34970A data acquisition instrument. Torque, rotational speed and power data are shown on three windows by CYB-808 intelligent torque meter.

2.3 performance parameters

The main performance parameters of SSEs were as follows:

(1) Shaft power

Shaft power was the actual output power by the shaft of expanders. In the experiment, by measuring the torque N and rotational speed n, it was obtained according to the following formula:

$$Pe = \frac{N \cdot n}{9550} \tag{5}$$

In the formula, the unit of Pe was kW, the unit of N was N.m, and the unit of n was r/min.

(2) Volumetric efficiency

Volumetric efficiency was the ratio of theoretical exhaust volume flow to actual exhaust volume flow, and it reflected the perfection of geometric structure utilization of expanders. The relationship formula was showed as followed:

$$\eta_V = \frac{V_{theoretical}}{V_{actual}} \tag{6}$$

(3) Shaft efficiency

Shaft efficiency was the ratio of actual shaft power to theoretical enthalpy drop. It was equal to the product of adiabatic efficiency and mechanical efficiency, reflected the overall efficiency of the expander. The relationship formula was showed as followed:

$$\eta_s = \frac{Pe}{\dot{m}(h_1 - h_{2s})} \tag{7}$$

In the formula, $\dot{m}$ was the mass flow of working fluid and its unit was kg/s, h_1 was the inlet specific enthalpy and h_{2s} was the outlet specific enthalpy, its unit was kJ/kg.

3 Experimental result and discussion

In this paper, the performance experiment of the SSE prototype made by our laboratory was carried out under two type working conditions, the one was variable rotational speed and constant inlet pressure, the another was variable inlet pressure and constant rotational speed. The variation trend of expansion ratios and volumetric efficiency under different working conditions were obtained.

3.1 Variation rotational speed

The pressure of the first condition set as 0.6 MPa, and the variation of rotational speed was from 800 to 3000 r/min.

Figure 4 showed the variation of outlet pipe and exhaust port pressure with rotational speed. It showed that the pressure of the exhaust pipe increased with the rotational speed. Due to the volume flow increased with rotational speed, under the same pipe diameter and the same exhaust pressure (atmospheric pressure), the flow resistance of exhaust pipe increased with exhaust air velocity, so the pressure of the outlet pipe increased. Correspondingly, the transient pressure of exhaust port showed a decreasing trend with the increase of rotational speed. The reason for this phenomenon might be that the leakage decreased with the increase of rotational speed, and the pressure sharing effect of leakage on exhaust port was weakened.

Figure 5 showed the variation of volumetric efficiency with rotational speed. It showed that volumetric efficiency increased significantly with rotational speed, which to some extent verifies the analysis of the above experimental results.

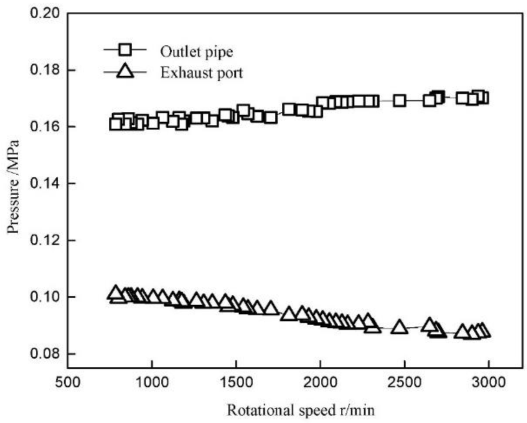

Fig.4 The variation of outlet pipe and exhaust port pressure with rotational speed

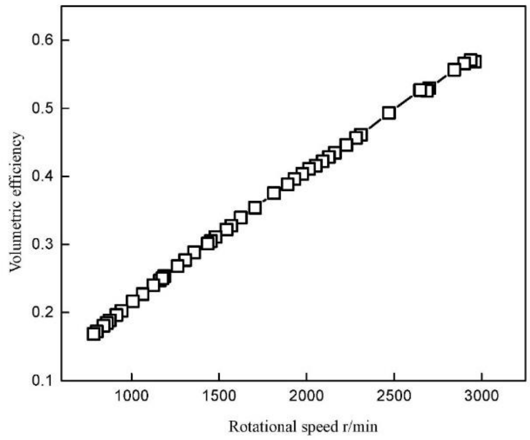

Fig.5 The variation of volumetric efficiency with rotational speed

Figure 6 showed the variation of internal and external expansion ratios with rotational speed.

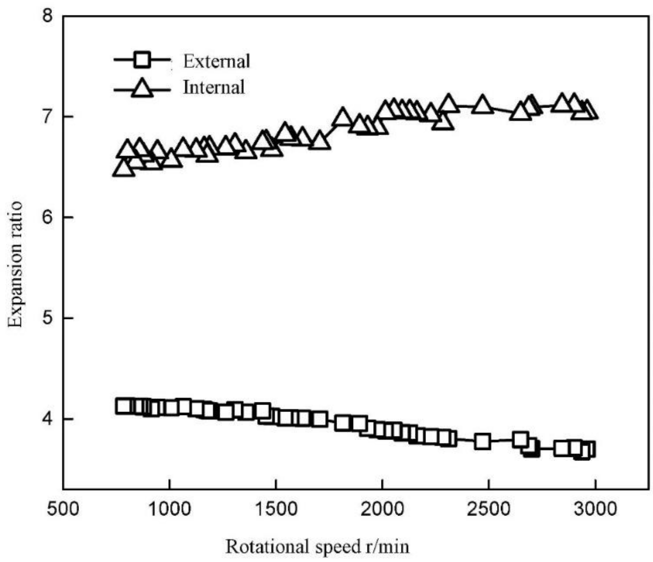

Fig.6 The variation of internal and external expansion ratios with rotational speed

It showed that the internal expansion ratio increased gradually and external expansion ratio decreased gradually with rotational speed. Because the inlet pressure was constant, the internal expansion ratio increased with the decrease of the transient pressure of exhaust port, while the external expansion ratio decreased with the increase of the outlet pipe pressure. From figure 6, the range of internal expansion ratio was 6.4 to 7.0, and the range of external expansion ratio was 4.1 to 3.7. Qualitative analysis of the influence factors on expansion ratio, it could be found that the internal expansion ratio was mainly affected by the internal working process of the expander, while the external expansion ratio was affected by both the internal working process and the back pressure, and its influencing mechanism was more complex. From the data, the expander was in serious overexpansion state. However, the internal volume ratio was 2.95. When the polytropic exponent was from 1.2 to 1.4, the expansion ratio by calculated was from 3.7 to 4.6 according to the definition of expansion ratio described in the literature. The experimental value of the external expansion ratio was lower than that calculated according to the literature formula (2). This phenomenon might be influenced by back pressure. However, the experimental value of the internal expansion ratio was obviously higher than the range calculated according to the literature formula (2). The reasons for this phenomenon cannot be analyzed clearly and need further study. From the experimental results, it can be further analyzed that if the inlet parameters of single screw expander increased, the larger internal expansion ratio could be obtained, and if the exhaust back pressure reduced through technical measures, the larger external expansion ratio could be obtained. So, SSEs could obtain higher expansion ratio only by adjusting the operating parameters. Moreover, the low single-stage expansion ratio was a key technical issue for the utilization of small and medium capacity ORC system. The above experimental result indicated the advantages of SSE in this flied.

3.2 Variation inlet pressure

The rotational speed of the second condition set as 3000r/min, and the variation of inlet pressure was from 0.4~0.6MPa.

Figure 7 showed the variation of outlet pipe and exhaust port pressure with inlet pressure. It can be found that the pressure of outlet pipe increased gradually with inlet pressure. Due to the density of inlet compressed air increased meanwhile inlet volume flow slightly changed with the increase of inlet pressure, so the mass flow increased accordingly. Moreover, due to the exhaust pressure (atmospheric pressure) was constant, the outlet volume flow increased. Under the same pipe diameter and the same exhaust pressure (atmospheric pressure), the flow resistance of exhaust pipe increased with exhaust air velocity, so the pressure of the outlet pipe increased. Correspondingly, the transient pressure of exhaust port showed a slightly decreasing trend with the increase of rotational speed. The reason for this phenomenon might be that the leakage would increase slightly with inlet pressure, due to constant rotational speed.

Figure 8 showed the variation of volumetric efficiency with inlet pressure. It showed that volumetric efficiency decreased slightly with rotational speed, which to some extent verifies the analysis of the above experimental results.

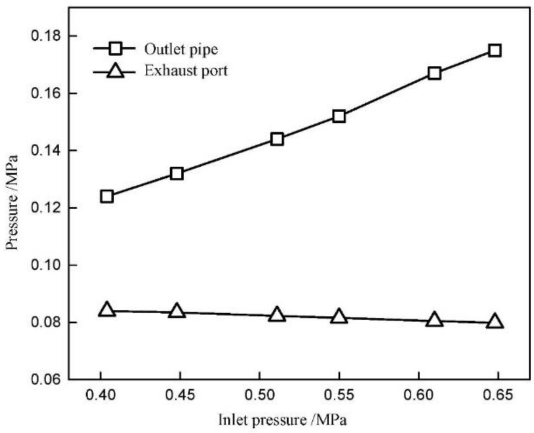

Fig.7 The variation of outlet pipe and exhaust port pressure with inlet pressure

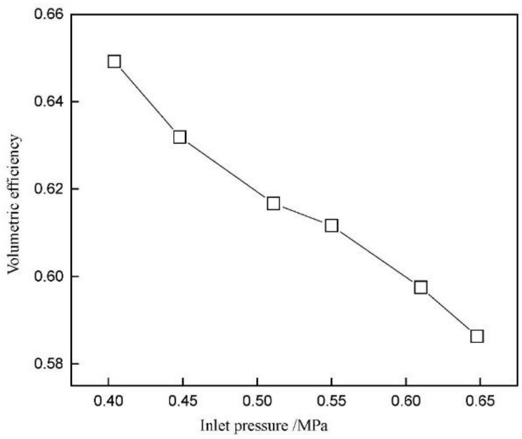

Fig.8 The variation of volumetric efficiency with inlet pressure

Figure 9 showed the variation of internal and external expansion ratios with inlet pressure. Both of expansion ratios increased with the inlet pressure, and the increase of the internal expansion ratio was obvious meanwhile the external expansion ratio only increased slightly. From the figure, the range of internal expansion ratio was 4.8 to 8.0, and the range of external expansion ratio was 3.2 to 3.7. The experimental value of the external expansion ratio was lower than that calculated according to the literature formula (2), but the gap reduced with the increase of the inlet pressure. It was indicated that the external expansion ratio could be improved by increasing the inlet parameters when the back pressure was constant. However, when the inlet pressure was low, the difference between experimental data and the calculated value of the literature formula (2) was not significant. With the increase of the inlet pressure, the difference increased significantly. It was indicated that the internal expansion ratio was greatly affected by the inlet parameters. This experimental phenomenon was difficult to explain with the concept of compression ratio of screw compressors, and its mechanism needs further theoretical analysis and experimental research. From the experimental result, the working process of the expander and the compressor was not a simple inverse thermodynamic process. The functional relationship of the expansion ratio defined by referring to the concept of the compression ratio of the screw compressor could not accurately reflect the actual working process. Therefore, the expansion ratio of SSEs should be obtained from the measured pressure data.

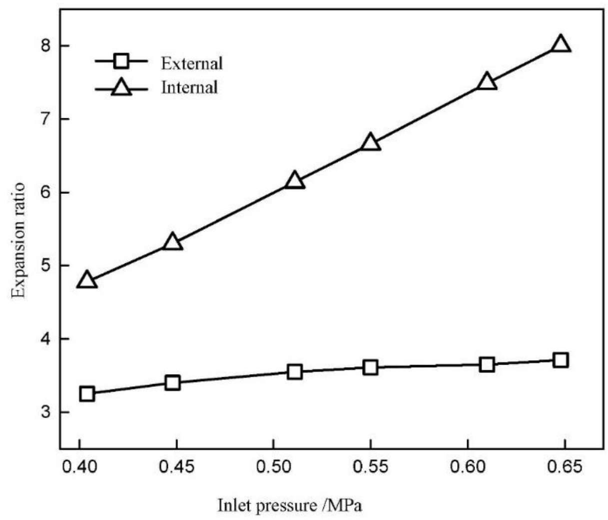

Fig.9 The variation of internal and external expansion ratios with inlet pressure

In addition, the internal expansion ratio increased rapidly with operating parameters, meanwhile the external expansion ratio was more obviously affected by back pressure. In summary, for the optimization of the working conditions of the ORC system based on SSEs, the expansion ratio improved by increasing the inlet parameters and reducing the exhaust back pressure simultaneously, and then the thermal efficiency of the whole cycle could be improved. The above conclusions were the same as those in reference [5].

4 Conclusions

In this paper, the experimental study on the expansion ratios of SSEs under different working conditions was carried out, and several conclusions were showed as followed:

1) Under the condition of constant inlet pressure and variable rotational speed, the external expansion ratio decreased, and the internal expansion ratio increased with rotational speed.

2) When the speed was 3000rpm and the inlet pressure was variable, the external expansion ratio increased slowly, and the internal expansion ratio increased rapidly with inlet pressure.

3) Under expansion effect was not occurred in the working condition of high inlet pressure.
4) Under various working conditions, internal expansion ratio was always higher than external expansion ratio, and slight over expansion was occurred. It was indicated that the internal flow of SSE was very turbulent during expansion process. So, SSE might have some characteristics of speed expanders.
5) Improving the inlet parameters of single screw expander and reducing the exhaust back pressure were effective measures to improve the thermal efficiency of ORC system based on SSEs.

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