FACTOR ANALYSIS OF EXPANSION RATIO FOR SINGLE SCREW EXPANDERS

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ABSTRACT

For low temperature waste heat recovery, Organic Rankine Cycle (ORC) is generally considered the most promising choice in varieties of potential technologies, and become a hotspot of research and development in international academic and industrial fields. However, reviewing the research results related ORC system in recent years, the actual situation was not optimistic. There were many technical bottlenecks hindered the application of ORC, especially for small scale system. Among those problems, the performance of expander was the key issue, and how to improve it was two aspects. First one was improving the shaft efficiency of expander, and it was the common sense of the researchers in this field. Second one was controlling appropriate expansion ratio (ε), and it was special requirement of small scale ORC system. Due to relative small expansion ratio, the thermal efficiency of ORC was low even if high efficiency of expanders. In actual conditions, the high efficiency and expansion ratio of expanders were hardly obtained simultaneously. So, it is necessary to carry out the special discussion about expansion ratio. Single screw expander was a type of volumetric prime mover. Due to special configuration, it had the potential of realized relative high expansion ratio. So, it makes possible getting high thermal efficiency of ORC. In this paper, we tried to analyze the influence factors of internal expansion ratio for single screw expanders. Firstly, the thermodynamic model of ORC was described, and the analysis of expansion ratio influenced cycle thermal efficiency was carried out. From the calculation results, it was found that thermal efficiency was increased with expansion ratio, but the accelerated velocity was decreased gradually. Considering the actual efficiencies of expander and pumps, appropriate expansion ratio should be existed. Secondly, the analysis of the influence factors of internal expansion ratio for single screw expanders was carried out, from the aspects of configuration, process and condition. From the results, high expansion theoretically could be obtained by changing the configuration of screw and gaterotor, inlet and outlet structure. The maximum volumetric ratio could above 20, and it could completely cover the temperature range of low temperature waste heat recovery. However, in actual condition, different meshing and fit clearances would influence leakage and cause expansion ratio reduced. So, configuration design and clearance control were key issues to improve the expansion ratio of single screw expanders.

1. INTRODUCTION

Nowadays, research and development of low temperature heat efficient thermal power conversion system becomes a hot research field in the world. Among the research of different type thermodynamic cycles, ORC was considered the most potential technology and obtained the most attention in the world.

For ORC research, thermodynamic cycle analysis was the most popular field all over the world, and the most academic articles focused on it. Reviewing most articles in this topic, the result was still existed the problems of ideal and bad suitable for actual situation. Especially for small scale low temperature heat source, the restriction of expander was the key problem, and it was indicated overestimating expander efficiency and ignoring the influence of expansion ratio. Referring the efficiency of large scale steam turbine, expander efficiency was usually assumed above 0.8. Moreover, the actual performance of working fluid pump was another key problem, and the pump efficiency was significantly overestimated in many articles, the value was about 0.65-0.85. Hence, the performance indexes of ORC were overestimated obviously, and thermal efficiency of ORC was higher than 10%, even above 15%.

For experimental study of ORC system, we can get quite different situation. There are many articles carried out the research of this topic, we just took some examples. The result data of some experimental study articles was showed in Table 1. From the results of those articles, we can find that the types of working fluid pump were included diaphragm and multistage centrifugal, although most of the articles did not gave actual pump efficiency, but the data of Quoilin et al. (2010) could reflect the real situation, it was only 15%. In other words, the actual pump efficiency was very lower than assumption, but we had to choose diaphragm and multistage centrifugal pumps because of high pressure requirement. It was very difficult to obtain high pressure, small flow rate and high efficiency at the same time, so how to improve the efficiency of working fluid pump was a key issue for ORC, especially for small scale system. We also can find that the types of expander were included scroll, rolling piston, radial turbine and single screw. Actually, every type expanders have been studied. From the results, it was obvious that radial turbine had the biggest capacity, second was single screw, and scroll and rolling piston was smaller. The working conditions of each article were quite different, but the temperature difference of evaporation and condensation were all about 30-50℃. Because several articles did not mentioned the degree of supercooled, so the actual temperature difference between expander's inlet and outlet should below above value. It indicated that the real energy utilization percentage of ORC was significantly lower than analyzed results, it was also indicated that the limitation of expander was a key bottleneck for sufficiently utilization low temperature thermal energy. From the data of table 1, we can find that the expansion ratio was about 2.5 to 6.6. Considering the thermophysical property of working fluid, the temperature difference between inlet and outlet for expander was estimated about 40-60℃, and it was obviously lower than the temperature difference between low temperature heat source and surrounding. Meanwhile, if expander efficiency was not good enough, the performance of ORC would be worse than desired. Unfortunately, the real efficiency of expanders was not satisfied, it was just about 40%-60%. Kang et al. (2012) presented the tested efficiency of radial turbine beyond 80%, but its expansion ratio was below 2.72, so the maximum value of thermal efficiency was about 5.65%, lower than the case of higher expansion ratio. The results of unsatisfied expansion ratio and expander efficiency were unacceptable thermal efficiency of ORC.

Reviewing the research results related small scale ORC system, we can get three bottlenecks for ORC performance: expander efficiency, expansion ratio and the efficiency of pump and fan. For small scale expanders, improving efficiency was a tuff job, and working fluid pump had similar situation. So, improving expansion ratio was an easier measure for obtaining higher thermal efficiency at present. There are many factors influencing expansion ratio, such as working fluid types, expander configurations and working conditions, etc.

Single screw configuration is composed of a screw and several gaterotors, and generally divides into four types of PC, PP, CC and CP. C and P were the abbreviations of cylinder and plate, respectively. The first abbr. means the shape of screw and the second one means the shape of gaterotor. CP type is the most common configuration of single screw because of easily processing. A single screw expander mainly consists of screw, gaterotors and shell. The screw groove, the internal wall of shell and the profile surfaces of the gaterotor teeth constitutes a closed space, which was to change the volume with the rotation of screw and gaterotor. In this paper, we tried to analyze the influence factors of expansion ratio for single screw expanders, and try to provide the technical measures to improve it.

Table 1: List of the performance indexes of some experimental study articles

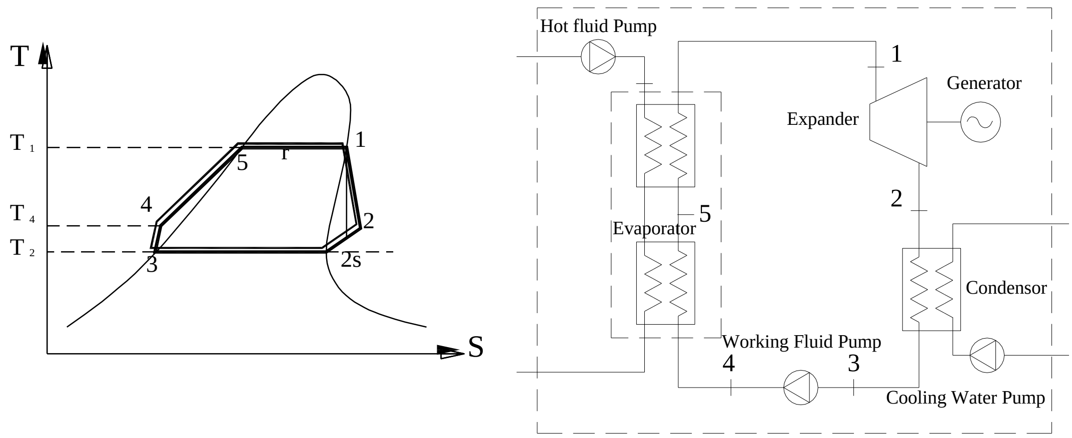
Author	Work fluid pump	Expander	Working fluid	W_G/kW	T_{eva}/℃	T_{con}/℃	η_E/%	ε	η_H/%	η_P/%
Li et al. (2013)	Diaphragm	Radial turbine	R123	6.07	90.7	39.4	58.53	4.19	7.98	No
Bracco et al. (2013)	Diaphragm	Scroll	R245fa	1.1-1.8	85.4-101.4	27-30	60-74	4.6-6.6	8.8-9.8	No
Quoilin et al. (2010)	Diaphragm	Scroll	R123	0.5-1.8	-	-	42-68	2.7-5.4	Max. 7.4	15%
Zheng et al. (2013)	Diaphragm	Rolling piston	R245fa	0.16-0.32	72.4-82.6	14.6-33.2	Max. 44	4	2-6	No
Zhou et al. (2013)	Multistage centrifugal	Scroll	R123	Max. 0.645	85.5-114.9	34.9-49.6	Max. 57	5	Max. 8.5	No
Kang et al. (2012)	Multistage centrifugal	Radial turbine	R245fa	24.5-31.2	77-83	44.5-47.3	76-82.2	2.62-.2.72	5.05-5.65	No
Gu et al. (2009)	Multistage centrifugal	Scroll	R600a	Max. 1.1	-	-	Max. 50.1	-	Max. 2.9	Assumed 85%
Zhang et al. (2014)	Multistage centrifugal	Single screw	R123	Max. 10.38	123.6-140.2	79-101	Max. 57.9	2.5-4.6	Max. 6.5	No

2. INFLUENCE FACTORS FOR ORC SYSTEM

Due to the limitation of expanders, low temperature heat energy could not make full use for small scale ORC system. A remarkable phenomenon was outlet temperature of working fluid higher than condensation temperature, and it was caused by relative lower expansion ratio of expanders. In this section, we tried to analysis expansion ratio how to influence the performance of ORC system.

2.1 Thermodynamic model

Firstly, a thermodynamic model was described, and specific information was showed in Figure 1.



(a) (b)

Figure 1: Organic Rankine cycle
(a) T-S chart; (b) Schematic diagram of thermal system

The thermodynamic model of power generation system was described as follow. Point 1, 3 and 5 were assumed to saturation state.

The absorption heat of working fluid in evaporator is calculated by

$$Q_1 = m(h_1 - h_4) \tag{1}$$

The generating capacity of expander is calculated by

$$w_G = (h_1 - h_{2s})\eta_a \eta_M \eta_G \tag{2}$$

The consumption of working fluid pump is calculated by

$$w_P = \frac{v_3(p_4 - p_3)}{\eta_P \eta_{EM}} = \frac{(h_4 - h_3)}{\eta_P \eta_{EM}} \tag{3}$$

The net generation of ORC is calculated by

$$W_{net,ORC} = W_G - W_P = m(w_G - w_P) \tag{4}$$

The thermal efficiency (thermodynamic first law efficiency) of ORC is defined as

$$\eta_{H,ORC} = \eta_{\mathrm{I},\mathrm{ORC}} = \frac{W_{net,ORC}}{Q_1} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} \tag{5}$$

The thermodynamic second law efficiency of ORC is defined as

$$\eta_{\mathrm{II},\mathrm{ORC}} = \frac{\eta_{H,ORC}}{1 - \frac{T_3}{T_1}} \tag{6}$$

Because of the limitation of expansion ratio (ε), when working fluid type and evaporation temperature was assumed, outlet pressure was calculated by expansion ratio. So, the relationship between inlet pressure and outlet pressure is defined as

$$\varepsilon = \frac{p_1}{p_2} \tag{7}$$

2.2 Result and discussion

Before calculated, we must assume some conditions. Here, R123 was selected by working fluid, because it had higher critical point. Generator efficiency and electric motor efficiency of working fluid pump was assumed 0.95, respectively.

2.2.1 Expansion ratio

Here, adiabatic and mechanical efficiency were assumed about 0.7 and 0.95, respectively. So the total efficiency of expander was about 0.65, and it was very near experimental results. Working fluid pump efficiency was assumed 0.8. Condensation temperature was assumed 30℃.

Figure 2 was the variation of thermal efficiency with expansion ratio of R123. At the same evaporation temperature, thermal efficiency increased with expansion ratio, until the condensation temperature was near environmental temperature. At the same expansion ratio, thermal efficiency decreased very slightly with the increase of evaporation temperature. However, with the absolute pressure rising, pump efficiency would be reduced in actual condition. So, it was indicated that lower evaporation temperature should be adopted at the condition of determined expansion ratio.

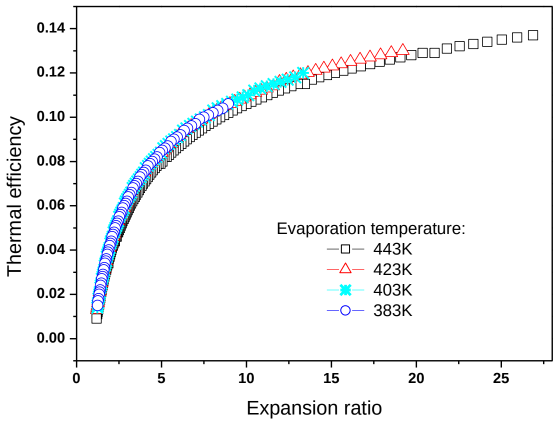


Figure 2: Variation of thermal efficiency with expansion ratio of R123

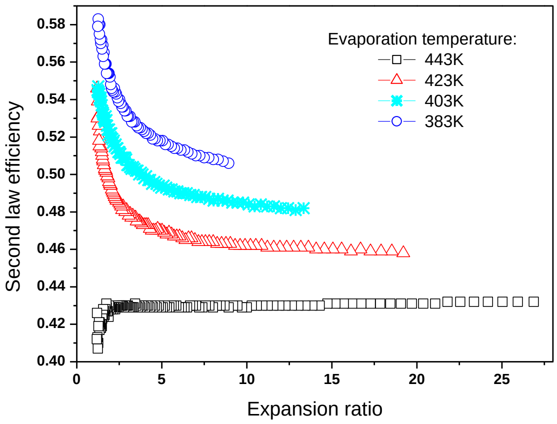


Figure 3: Variation of Second law efficiency with expansion ratio of R123

Figure 3 was the variation of second law efficiency with expansion ratio of R123. At the same evaporation temperature, second law efficiency decreased quickly with expansion ratio, until the condensation temperature was near environmental temperature. With the increase of evaporation temperature, the decrease velocity become gently. The reverse trend occurred with evaporation temperature closing critical point. At the same expansion ratio, second law efficiency accelerated declined with the increase of evaporation temperature. Summary above calculating result, it was indicated that improving expansion ratio was an effective method to increase the thermodynamic performance of ORC, however, if considering expander and pump efficiency would decrease with the increase of expansion ratio, so expansion ratio should existed suitable range.

2.2.2 Adiabatic efficiency

Here, mechanical efficiency was assumed 0.95. Expansion ratio was assumed 6. Working fluid pump efficiency was assumed 0.8.

Figure 4 was the variation of thermal efficiency with adiabatic efficiency of R123. At the same evaporation temperature, thermal efficiency lineal increased with adiabatic efficiency. At the same adiabatic efficiency, thermal efficiency decreased slightly with the increase of evaporation temperature, the trend was more obvious in the situation of lower adiabatic efficiency. So, it was indicated that improving evaporation temperature should not a reasonable method if expansion ratio fixed. Figure 5 was the variation of second law efficiency with adiabatic efficiency of R123. It has similar trend compared with thermal efficiency, just a little different. With the increase of adiabatic efficiency, the difference of second law efficiency was gradually added among different evaporation temperature. The phenomenon could explain that thermodynamic perfection will be improved with the increase of adiabatic efficiency and far away from critical point.

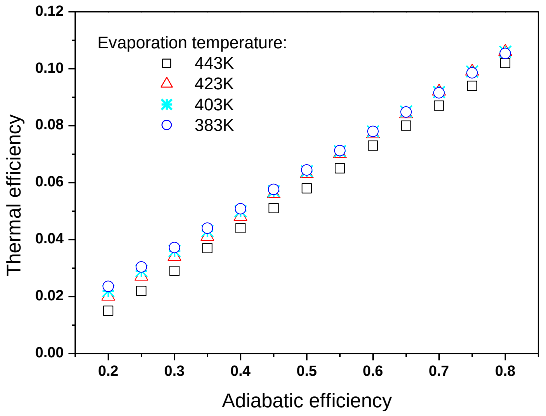


Figure 4: Variation of thermal efficiency with adiabatic efficiency of R123

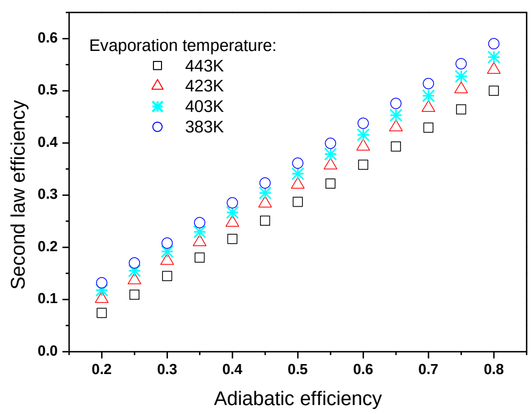


Figure 5: Variation of second law efficiency with adiabatic efficiency of R123

2.2.3 Pump efficiency

Here, adiabatic and mechanical efficiency were assumed about 0.7 and 0.95, respectively. Expansion ratio was assumed 6.

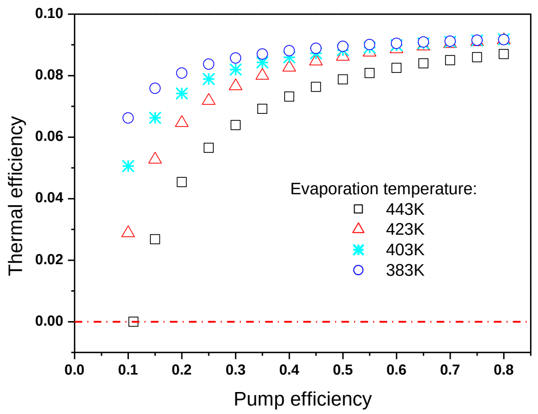


Figure 6: Variation of thermal efficiency with pump efficiency of R123

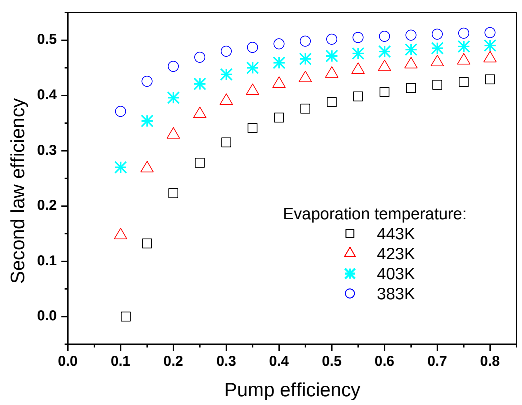


Figure 7: Variation of second law efficiency with pump efficiency of R123

Figure 6 was the variation of thermal efficiency with pump efficiency of R123. At the same evaporation temperature, thermal efficiency accelerated declined with pump efficiency decrease. At the same pump efficiency, thermal efficiency decreased accelerated with the increase of evaporation temperature, the trend was more obvious in the situation of lower pump efficiency. From the result, it was also found that the influence of pump efficiency was weakened at relative lower evaporation temperature, even lower than 0.3. So, it was indicated that improving evaporation temperature should not a reasonable method if low pump efficiency. Figure 7 was the variation of second law efficiency with pump efficiency of R123. It has similar trend compared with thermal efficiency.

Summary above calculating result, it could be found that improving expansion ratio could increase thermodynamic performance of ORC system. If expansion ratio could not be improved, increase evaporation temperature was no use to improve performance.

3. EXPANSION RATIO OF SINGLE SCREW EXPANDERS

According to the different working principles, expanders can be classified two types: velocity and positive displacement. Velocity expanders mainly include single stage and multi-stage axial turbine, single stage and multi-stage radial turbine, etc. Positive displacement expanders mainly include piston, scroll, twin screw and single screw, etc. According to the different movement forms, positive displacement expanders can be classified two types: reciprocating and rotary. Piston expander is the former, and scroll, twin screw and single screw are the latter.

For velocity expanders, increasing rotational speed could improve expansion ratio. But, if the rotational speed was increased too high, mechanical loss would be significantly raised. So, multi-stage configuration was adopted usually in order to improve expansion ratio. Generally speaking, expansion ratio of single stage for axial turbine was lower than radial turbine. So, the stage numbers of axial turbine should be more than radial turbine if obtained the same expansion ratio.
For positive displacement expanders, expansion ratio (ε) was influenced by internal volumetric ratio (τ) and adiabatic index of working fluid. The function existed in ideal condition:

$$\varepsilon = \left(\frac{V_{out}}{V_{in}}\right)^{\kappa} = \tau^{\kappa} \tag{8}$$

For reciprocating expanders, expansion ratio could be changed through controlling inlet and exhaust phases. However, if the expansion ratio was improved too high, adiabatic efficiency would greatly decrease due to the clearance volume. For rotary expanders, if the configuration and working fluid has been determined, expansion ratio was fixed. There are two methods to improve it: the one is modify the configuration of expanders, and the other is added capacity adjustment mechanism, for example, slide valve.
Single screw expander is positive displacement type, and belongs to rotary type too. Considering actual situation, mainly influence factors of expansion ratio included three aspects: internal volumetric ratio, working fluid type and working condition.

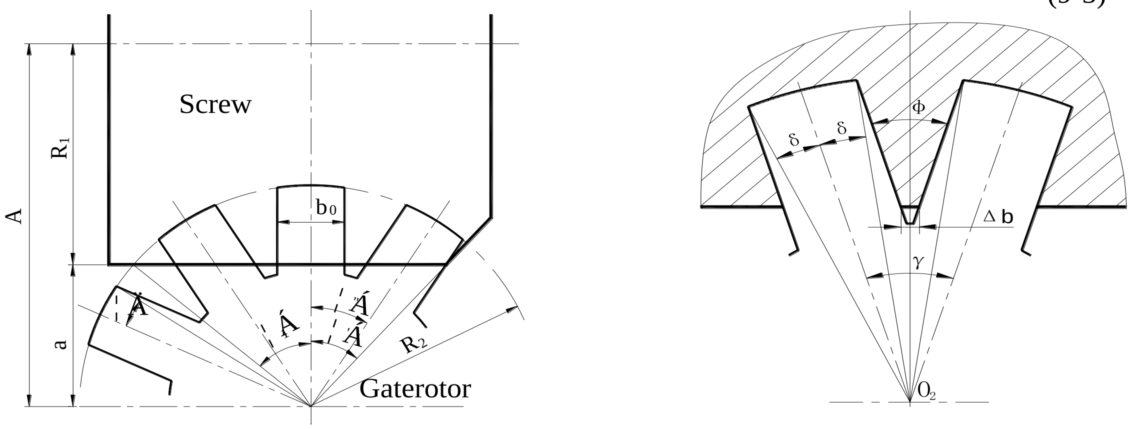
3.1 Internal volumetric ratio

Because CP type single screw configuration was the simple and widely use among different single screw types, so it was selected for the case to analysis. Geometric structure of single screw meshing pair was showed on Figure 8. Mainly parameters include screw radius R_1, gaterotor radius R_2, the center distance between screw and gaterotor A, meshing angle α_1, discharge angle α', half angle of teeth width δ, teeth width of gaterotor b_0, minimum thickness of screw rib Δb. When R_1, R_2, A, b_0 were assumed, other parameters could be calculated.
In order to obtain internal volumetric ratio, intake and exhaust volume should be calculated. Exhaust volume was the maximum volumetric element. Making reference to the handbook (Yu et al, 2012), the maximum volumetric element can be calculated by

$$V_1 = \int_{\alpha_1-\delta}^{\alpha_1+\delta}\int_{-\frac{b_0}{2}}^{R_2\sin(\alpha_1-\alpha)} \frac{R_1^{\,2}-(A-\sqrt{R_2^{\,2}-b^2}\cos\alpha+b\sin\alpha)^2}{2\cos\alpha}\omega db d\alpha \tag{9-1}$$

$$V_2 = \int_{\alpha''}^{\alpha_1-\delta}\int_{-\frac{b_0}{2}}^{\frac{b_0}{2}} \frac{R_1^{\,2}-(A-\sqrt{R_2^{\,2}-b^2}\cos\alpha+b\sin\alpha)^2}{2\cos\alpha}\omega db d\alpha \tag{9-2}$$

$$V_t = V_1 + V_2 \tag{9-3}$$



(a) Meshing pair (b) Gaterotor teeth and screw grooves

Figure 8: Geometric structure of meshing pairs

Intake volume was determined by intake triangle port and screw groove width (equal to gaterotor teeth width). When the arc length of the circumferential direction of intake triangle port was assumed, the rotary angle of screw could be calculated. Then, it was converted into the rotary angle of gaterotor. So, the rotary angle of gaterotor of intake process was the angle above sentence mentioned plusing teeth width angle (2δ). Thus, intake volume could be calculated by equation (9). Of course, the limits of integration should be changed.
Obviously, if we want to improve expansion ratio, intake volume should be reduced. According to above describe, there were two measures to realize it: the one was reduced the dimension of intake triangle port, and another was reduced screw groove width. The minimum dimension was zero, so the maximum internal volumetric ratio should be existed when screw groove width as certain value. Reducing screw groove width also can improve internal volumetric ratio, but it was limited by the requirement of gaterotor mechanical strength.
Here, we give an example to analysis. R_1, R_2, A, b_0 were assumed as 58.5mm, 58.5mm, 96mm and 17.1mm, respectively. Variation of maximum volumetric element and maximum internal volumetric ratio with minimum thickness of screw rib was showed on Figure 9. With the increase of minimum thickness of screw rib, maximum volumetric element was declined linearly, and maximum internal volumetric ratio was increased accelerated. From calculation result, appropriate reducing screw groove width could make maximum internal volumetric ratio to reach above 30. Considering the dimension of intake triangle port, internal volumetric ratio could be easy to realize above 20. Summery above analysis, for single screw expanders, internal volumetric ratio could be improved theoretically by configuration adjustment to satisfy the requirement of ORC system.

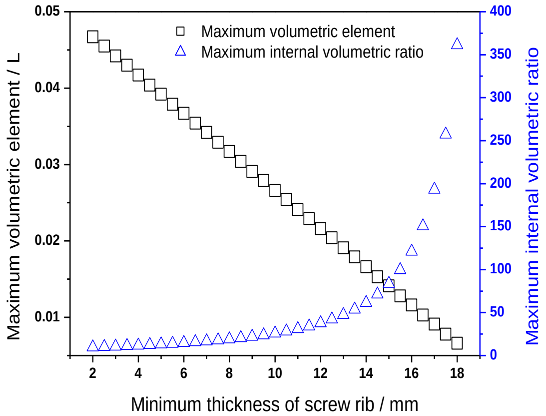


Figure 9: Variation of maximum volumetric element and maximum internal volumetric ratio with minimum thickness of screw rib (Screw diameter = 117mm)

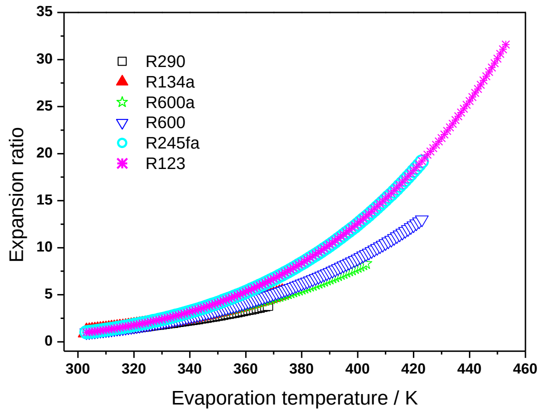


Figure 10: Variation of expansion ratio with evaporation temperature for six working fluids

3.2 Working fluids

Thermophysical properties of working fluid influencing expansion ratio was reflected in two aspects. The one was pressure ratio between evaporation and condensation temperature. Assuming condensation temperature was 303K, R290, R134a, R600a, R600, R245fa and R123 were selected to analyze. Figure 10 was the variation of expansion ratio with evaporation temperature for six working fluids. From the results, at the same temperature difference between evaporation and condensation, working fluid of higher critical point has higher expansion ratio, however, for working fluid of lower critical point, the absolute pressure was significantly higher than the former. In addition, expansion ratio of higher critical point working fluids can beyond 20, even 30 in subcritical region. So, if we want to sufficiently use low temperature heat source, expansion ratio of expanders should be improved.
On the other hand, adiabatic index of working fluids was another important factor. In common sense, working fluid has more complex of molecular structure, the adiabatic index has smaller. Furthermore, from some research result of, adiabatic index of many Freon was significant declined with pressure increase, and it was slightly changed with temperature increase in the situation of lower pressure

(about 200kPa), but it was decreased with temperature increase in higher pressure. For example, the adiabatic index of air was 1.4, R22 was about 1.12-1.19 and R600a was about 1.02-1.05. It was showed that adiabatic index of Freon was obviously lower than air. For refrigeration, it was a good characteristic because of reducing compression power consumption. But for expansion, the situation was opposite. Lower adiabatic index caused lower expansion ratio at the same configuration of expanders.

3.3 Working conditions

The above analysis was based on ideal conditions. But in actual working condition, the situation was more complex. For single screw expanders, many leakage passes were existed due to clearance fit requirement, and leakage will cause the increase of discharge temperature and pressure, so the possess index was lower than adiabatic index, at last, expansion ratio reduced. However, experimental study of this field was much lacked.

4. CONCLUSIONS

In this paper, expansion ratio influencing single screw expanders was discussed. Firstly, a simple thermodynamic model was described, and then expansion ratio influencing ORC system was analyzed. Secondly, the influence factor of expansion ratio for single screw expanders was analyzed. Through those works, five conclusions are obtained:

(1) With the increase of expansion ratio, thermal efficiency of ORC was improved, and improving speed was fast firstly, and then slowed down. Considering the actual efficiencies of working fluid pump and expander, appropriate improving expansion ratio was the effective technical measure to improve ORC performance.
(2) At the same expansion ratio, increase intake temperature was no use improving thermal efficiency, and could reduce second low efficiency. So, increase intake parameters were not a good measure to improving ORC performance when the configuration of expander was fixed or regulation system was not existed.
(3) For a certain configuration of single screw expander, maximum internal volumetric ratio was existed. Through adjusting the demission of intake triangle port and screw groove width, internal volumetric ratio could be changed.
(4) Thermophysical properties of working fluid would influence expansion ratio, especially adiabatic index. In next research, process index should be obtained by experimental study.

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