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Performance of Scroll expander for CO₂ Refrigeration Cycle

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ABSTRACT

Since trans-critical carbon dioxide refrigeration cycles have a large throttling loss, recovering the throttling loss by an expander is effective for improvement of the cycle performance. There are many types of expander such as turbo, rotary vane, screw and scroll. Among them, the scroll machine is one of the expanders having good performance. In this study, in order to examine feasibility of the scroll expander for the carbon dioxide refrigeration cycle, the performance of the carbon dioxide scroll expander is investigated both theoretically and experimentally. Since a leakage in the expander influences its performance greatly and is a trans-critical flow from a supercritical to a two-phase region in the expander for the trans-critical carbon dioxide refrigeration cycle, a calculation method of the trans-critical leakage flow rate in the expander is discussed at first in the theoretical study. The calculation results showed that the total efficiency of the scroll expander becomes about 60% when the leakage gap size is 10 μm and the rotational speed is 3600rpm. On the other hand, a prototype of scroll expander was made using a mechanical element of a scroll compressor for a water heater and its performance was measured. The volumetric efficiency of 80 % was obtained and the total efficiency reached about 55 % although the scroll element of the compressor is used without any major modifications.

1. INTRODUCTION

Transition from synthetic refrigerants to natural ones is promoted in the refrigeration industry in recent years. Carbon dioxide (CO₂) is one of the promising candidate of the natural refrigerant because it has no flammability and no toxicity. The inherent performance of air cooled CO₂ cycles is lower than that of traditional cycles working with HFCs, and the efficiency of the CO₂ cycle should be improved. Since the CO₂ cycle has pretty large throttling loss at the expansion process, it is important to recover the loss in order to improve the performance of the CO₂ cycle. The use of an expander is one way to recover the throttling loss. Various types of the expander were proposed for the last decade. Stosic et al. (2002) developed a twin screw expander combined with a compressor, Beak et al. (2002) tested a piston-cylinder expansion device, Quack et al. (2004) proposed an integration of a three-stage expander in the series of their studies, Tøndell et al. (2004) introduced a concept of an impulse type turbine and Okamoto et al. (2005) presented the performance of a two-stage swing expander combined with a swing compressor. Authors (Fukuta et al., 2000, 2001, 2003) have been studying the feasibility of a vane type expander, and the performance of the expander for the CO₂ cycle is being clarified theoretically and experimentally.

Scroll machines can be operated as the expander and its performance is pretty high (Yanagisawa et al., 1988). However, reports of the scroll expander for CO₂ cycle were quit few, except Huff et al. (2003) showed the performance of the scroll expander experimentally. In this study, feasibility of the scroll expander for the CO₂

refrigeration cycle is examined both theoretically and experimentally. Since a leakage in the expander influences its performance greatly and is a trans-critical flow from a supercritical to a two-phase region in the expander for the trans-critical carbon dioxide refrigeration cycle, a calculation method of a trans-critical leakage flow rate in the expander is discussed at first in the theoretical study. Then, a prototype of scroll expander is developed and its performance is measured.

2. SCROLL EXPANDER

Figure 1 shows pressure-enthalpy ($P-h$) diagrams for the CO₂ trans-critical cycle with the expander and without the expander. The enthalpy difference between the enthalpies at an evaporator inlet with and without the expander is the available energy by the expander, and it is useful not only as the recovered work but also as the additional refrigeration effect. In order to recover the throttling loss effectively, the expander performance must be high. Authors developed the vane type expander and its total efficiency is about 60 % in a recent model. We set our goal at a primary stage to be 70 % and try to evaluate another type of expander, i.e. scroll type expander, as an available choice along with the improvement of the vane expander.

The scroll type expander was reported to have a good performance in a pneumatic system (Yanagisawa et al., 1988). Figure 2 shows an operating principle of the scroll expander. The scroll expansion mechanism is just reverse of a scroll compression mechanism. The scroll expander, therefore, can be realized with a scroll compressor mechanism without a major modification. The scroll expander does not need any control valves of inlet and outlet, there is no leakage path directly from the high pressure side to the low pressure side, an expansion volume ratio can be designed arbitrary and it has multiple expansion chambers simultaneously, which results in less torque fluctuation and an operation with low noise and vibration. On the other hand, the scroll mechanism is more complicated as compared with the vane expander and has larger mechanical loss.

3. THEORETICAL ANALYSIS

3.1 Pressure change in expansion chamber

Pressure change in the expansion chamber is analyzed with taking account of volume change of the chamber, flow restriction at an inlet port, flow restriction caused by tooth proximities at the inlet and the outlet, and a leakage. The chamber volume and the flow sectional area are calculated geometrically. Although it is desirable to calculate the pressure in the expansion chamber based on an energy equation (Fukuta et al., 2003), it is simply calculated by the following equation in this study.

$$P_{i+1} = P_i \left(\frac{V_i/m_i}{V_{i+1}/m_{i+1}} \right)^\kappa \quad (1)$$

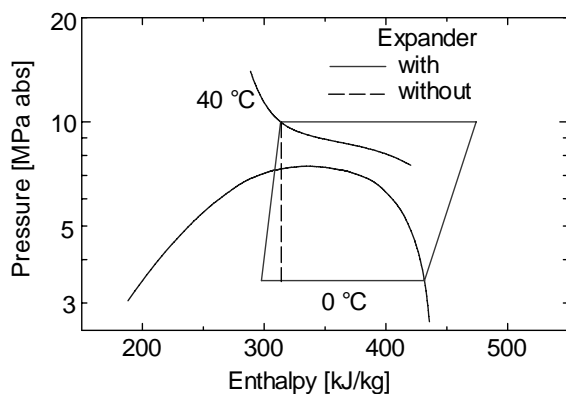


Fig.1 $P-h$ diagrams of CO₂ cycles

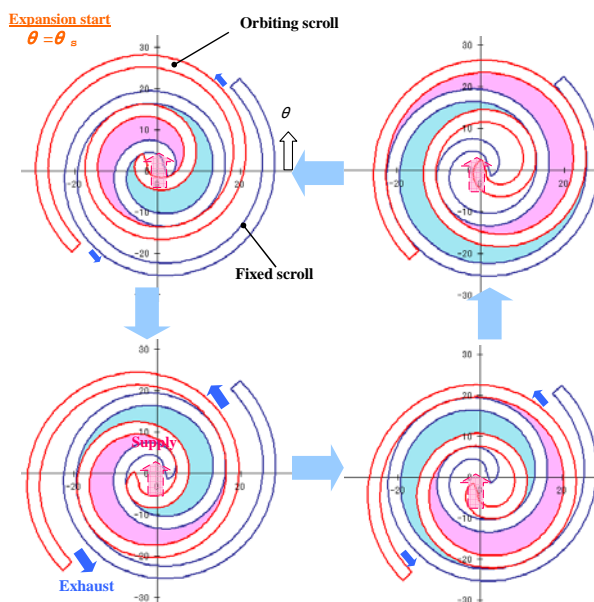


Fig. 2 Operating principle of scroll expander

where, P is the chamber pressure, V is the chamber volume, m is mass of refrigerant in the chamber, subscript i is a time step of the calculation and exponent κ is an adiabatic compression exponent. The refrigerant mass in the chamber is calculated by Euler method based on the inlet and outlet flows and the leakage flows into/from the chamber. In general, the exponent κ in the CO₂ expansion process is hardly decided because the condition of carbon dioxide changes from super-critical to two-phase. In this study, the exponent κ is defined so that the pressures and specific volumes, (P, v) and (P', v') , at slightly different pressures under constant entropy satisfy the relationship of an adiabatic change, i.e. $Pv^\kappa = P'v'^\kappa$. The exponent κ obtained in this manner is an apparent one and the value of κ under a given entropy is shown in Fig. 3. In this case, the entropy is 1.356 kJ/(kgK) which corresponds to 10 MPa and 40 °C.

In order to calculate the refrigerant mass in the expansion chamber, the mass flow rates into/from the chamber should be given. In a simple analysis, the mass flow rate is generally calculated as an isentropic nozzle flow using a constant adiabatic exponent. However as shown in Fig. 3, the adiabatic exponent is not constant against the pressure. In this study, the mass flow rate is obtained based on an energy equation as follows.

$$G = CA\rho_2 w_2 \tag{2}$$

$$w_2 = \sqrt{2(h_1 - h_2)} \tag{3}$$

where, G is the mass flow rate, C is a flow coefficient, A is the flow sectional area, ρ is density, w is flow velocity, h is enthalpy and subscripts 1 and 2 mean upstream and downstream respectively. h_2 and ρ_2 are calculated under the condition that the entropy at the downstream pressure is the same as the upstream. Figure 4 shows the mass flow rate per unit area against the downstream pressure under the upstream conditions of 10 MPa and 1.356 kJ/(kgK). Thick solid line shows the mass flow rate calculated by Eq. (2), in which the effective flow sectional area CA is 1. Since the velocity increases while the density decreases with decreasing the downstream pressure, the mass flow rate has maximum value against the downstream pressure. Therefore, when the downstream pressure is less than the pressure at which the mass flow rate becomes maximum, the flow is choked and the mass flow rate becomes constant against the downstream pressure as shown by the thick broken line. The choking pressure in this case corresponds to the point that the flow condition at the downstream becomes two- phase condition. In Fig. 4, the mass flow rate calculated by the general nozzle flow equation using the apparent exponent κ at the upstream condition is also plotted with thin line. If the downstream pressure is greater than the choking pressure, both mass flow rates have the almost same value. Even when the downstream pressure decreases, the flow is hardly choked because the two-phase condition at the downstream is not considered in the nozzle flow model. The choking pressure under different upstream pressure is shown in Fig. 5. It is shown that when the upstream pressure is greater

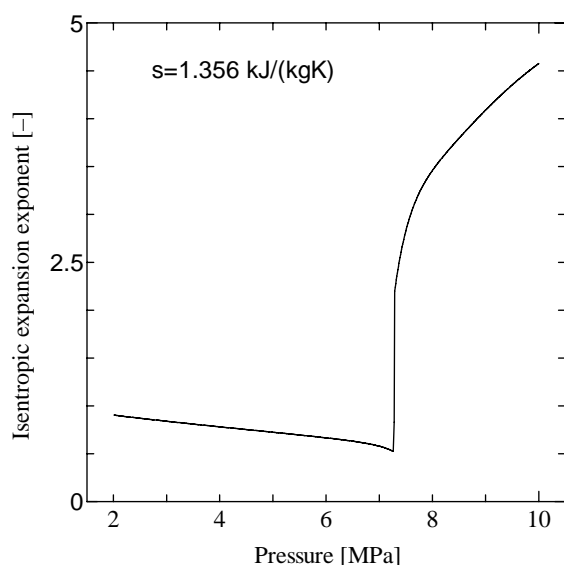


Fig. 3 Isentropic expansion exponent

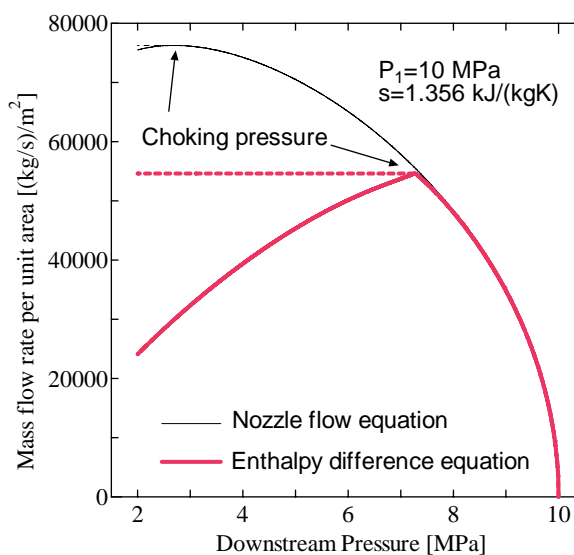


Fig. 4 Mass flow rate against downstream pressure

than a certain value, e.g. 9.3 MPa in this case, the choking pressure is constant because the choking phenomena is caused by an acceleration by the flash at the entrance of the two-phase region.

The pressure in the expansion chamber during the expansion process is greatly influenced by the pressure loss during the supplying process because the pressure loss reduces the pressure from which the expansion process starts. As a result, the pressure loss during the supplying process reduces a P-V work and the mass flow rate. The pressure loss occurs at the inlet port and is caused by the proximity of the scroll teeth just before the crescent-shape expansion chamber is formed individually. The teeth proximity is shown in Fig. 6 and is taken into account geometrically. In addition, a flow restriction occurs at the beginning of the exhaust process by the teeth proximity. Although it is also considered geometrically, it affects only on the pressure change at the beginning of the exhaust process.

Figure 7 shows the calculated P-V diagram taking the rotational speed as a parameter. The specifications of the scroll expander are; the chamber volume when the expansion process starts is 1.54 cm³ and that at the end of the

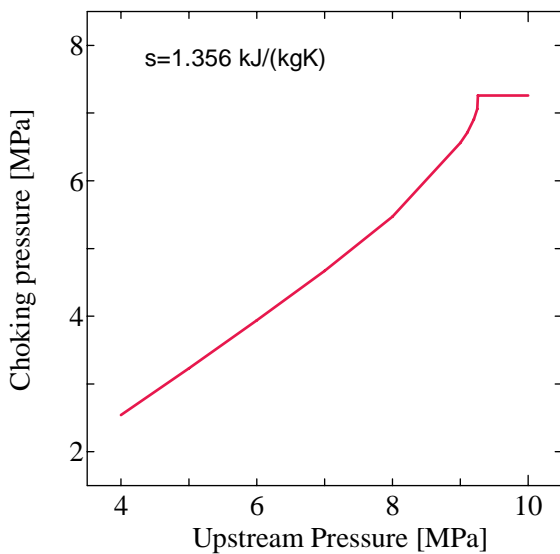


Fig. 5 Choking pressure under different upstream pressure

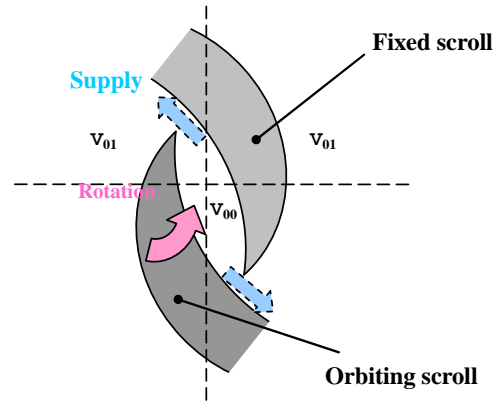


Fig. 6 Proximity of scroll teeth

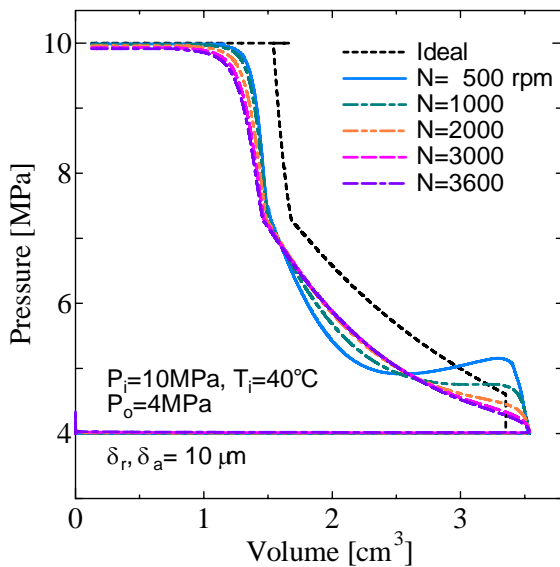


Fig. 7 Calculated P-V diagram

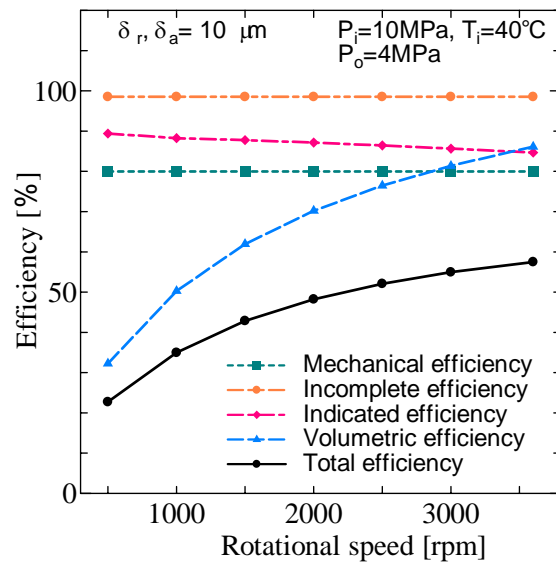


Fig. 8 Calculated performances

expansion process is 3.35 cm^3 , which results in a built-in expansion volume ratio of 2.18. The leakage clearances in both radial and axial direction of the scroll teeth are assumed to be $10 \text{ }\mu\text{m}$ in this calculation. The operating conditions are; the inlet pressure is 10 MPa, the inlet temperature is $40 \text{ }^\circ\text{C}$ and the outlet pressure is 4 MPa. The pressure starts to decrease before the individual expansion chamber is formed geometrically because of the flow restriction at the inlet port and the proximity of the scroll teeth. The pressure loss during the supplying process becomes larger as the rotational speed increases. In the super-critical region, the pressure decreases steeply with increasing the chamber volume, then the pressure change becomes slowly in the two-phase region. When the rotational speed is low, the pressure decreases steeply at the first part in the two-phase region. This is because the much leakage occurs to the outer expansion chamber. The pressure in the outer expansion chamber under the low rotational speed, on the contrary, increases due to the leakage before the discharge starts. Since the built-in expansion volume ratio is smaller than that of the ideal expansion, the working fluid starts to be discharged to the outlet before it is fully expanded to the outlet pressure. It is called 'under-expansion condition' and an under-expansion loss is caused. When the rotational speed is high, since the pressure at the beginning of expansion is low due to the inlet pressure loss and the leakage during the expansion process is small, the pressure at the end of expansion process approaches the outlet pressure and the under-expansion loss becomes small as a result.

3.2 Expander performance

The performance of the expander is examined based on the mass flow rate through the expander and the output power. The ideal mass flow rate is given as follows.

$$G_{ideal} = NV_s \rho_s \quad (4)$$

where, N is the rotational frequency, V_s is the chamber volume when the expansion chamber is closed and ρ_s is inlet density of CO_2 . The calculated mass flow rate is obtained by integrating the mass flow rate discharged from the expansion chamber during the discharge period.

$$G_{cal} = N \int_d G_{out} dt \quad (5)$$

The ratio of these flow rates is defined as the volumetric efficiency, η_v .

$$\eta_v = G_{ideal} / G_{cal} \quad (6)$$

The output power is calculated based on the indicated power. When the expansion pressure ratio corresponding to the built-in volume ratio of the expander does not coincide with the operating pressure ratio, the expansion process becomes incomplete one, i.e. under-expansion or over-expansion, and a certain loss occurs due to the mismatch between the pressure ratios. The ratio of area enclosed by the P - V diagram under the incomplete expansion to that under full expansion is termed the incomplete expansion efficiency, η_{ie} .

$$\eta_{ie} = \oint P_{ideal_ie} dV / \oint P_{ideal} dV_{ideal} \quad (7)$$

where, V_{ideal} is the volume of an imaginary expansion chamber in case of the full expansion. The indicated efficiency η_i is the ratio of area enclosed by the calculated PV diagram to ideal one with the incomplete expansion process.

$$\eta_i = \oint P dV / \oint P_{ideal_ie} dV \quad (8)$$

The ratio of shaft power L_s to the indicated power is mechanical efficiency η_m and is assumed to be constant ($=0.8$) in this study.

$$\eta_m = L_s / N \oint P dV \quad (9)$$

The ratio of shaft power to the ideal expansion power for the calculated mass flow rate is the total efficiency η_t and is product of each partial efficiency as follows.

$$\eta_t = \frac{L_s / G_{cal}}{N \oint P_{ideal} dV_{ideal} / G_{ideal}} = \eta_m \eta_i \eta_{ie} \eta_v \quad (10)$$

Fig.8 shows the volumetric efficiency, the incomplete expansion efficiency, the indicated efficiency and the total efficiency against the rotational speed. Here, the mechanical efficiency is assumed to be 80 % regardless of the rotational speed. As the rotational speed increases, since the leakage relative to the mass flow rate becomes small, the volumetric efficiency increases and it reaches about 85 % at the rotational speed of 3600 rpm. Although the indicated efficiency slightly decreases with increasing the rotational speed, this is mainly due to the relative

reduction of the leakage and it does not mean the increase of the energy loss. The pressure loss at the inlet port and that due to the teeth proximity also decrease the indicated efficiency. However, it contributes the reduction of the mass flow rate and increases the volumetric efficiency. The CO₂ trans-critical expansion process has relatively small expansion volume ratio (Fukuta et al. 2001) and the incomplete expansion efficiency is almost 100 % with the built-in volume ratio of 2.18. As a result, the total efficiency is expected to be about 60 % at the rotational speed of 3600 rpm. In this study, the influences of the axial and radial clearance magnitudes on the performance were examined too. It was found that the axial clearance has larger influence on the performance and a certain measure is needed to minimize the axial clearance to achieve the good performance.

4. EXPERIMENTAL SETUP

4.1 Scroll expander

Authors developed a prototype of the scroll expander using a mechanical element of a scroll compressor for CO₂ water heater manufactured by Matsushita Electric Industrial Co. Ltd.. The specifications of the scroll compressor are the same as those used in the calculation ($V_s=1.54 \text{ cm}^3/\text{rev.}$). A schematic view of the experimental scroll expander is illustrated in Fig. 9. It consists of the scroll element, an expander case, balancers and an extension shaft. A discharge valve attached in the original compressor is removed from the scroll element and bypass holes are plugged. The scroll element is enclosed in the expander case which can resist to the inlet pressure around 10 MPa. The inlet pressure is applied to the back side of an orbiting scroll to cancel a thrust force through a back-pressure port of the expander case. Two balancers are attached on the shaft to compensate the imbalance force and moment of the scroll assembly. In this study, the extension shaft is connected to the scroll shaft and comes outside of the expander case so that the expansion power is measured by a torque meter at the outside of the expander. A shaft seal and seals of the expansion case are done by O-rings. The moving parts are lubricated by the oil circulated with the CO₂ refrigerant.

4.2 Experimental cycle

The prototype of the scroll expander is connected to an experimental CO₂ refrigeration cycle shown in Fig. 10. The back-pressure which is the same as the inlet pressure is supplied to the back-pressure port of the expander case. Two compressors utilized as CO₂ water heater compressor are used to cover the wide range of mass flow rate. A gas cooler and an evaporator are double tube type. The extension shaft of the expander is connected to an oil pump as a load through a torque meter and the rotational speed of the expander is controlled by changing the discharge pressure of the oil pump. Pressures and temperatures at each point in the cycle are measured by Bourdon-type pressure gages and T-type thermocouples respectively. The mass flow rate flowing through the expander, which includes the flow rates through the expansion element and the back-pressure chamber, is measured by a Coriolis-type mass flow meter. In the experiment, the inlet pressure is 9 MPa, the inlet temperature is 40 °C and the outlet pressure is 4MPa. The mass flow rate and the shaft torque are measured by changing the rotational speed from 2000 to 4000 rpm.

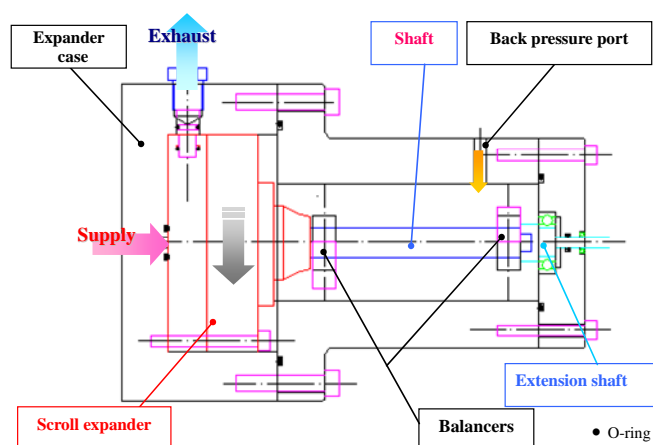


Fig. 9 Schematic view of scroll expander

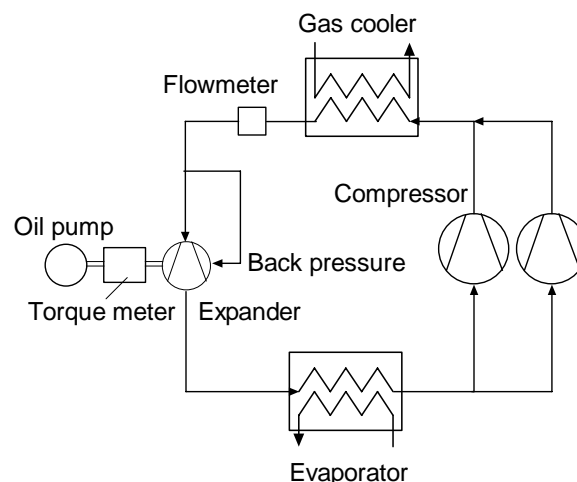


Fig. 10 Experimental cycle

5. EXPERIMENTAL RESULTS

The volumetric efficiency and the total efficiency are shown against the rotational speed in Fig. 11. Since the pressure change in the expansion chamber is not measured, the P - V work can not be obtained and the indicated and the mechanical efficiencies are not specified in the experiment. The product of these efficiencies is derived by dividing the total efficiency by the volumetric efficiency. It also includes the incomplete expansion efficiency, but the efficiency is almost 100 % in this case. The volumetric efficiency is about 80 % and does not decrease a lot even when the rotational speed decreases to 2000 rpm. This is because the scroll expander does not have a leakage path through which the leakage flows directly from the inlet to the outlet unlike the vane type expander. The product of the mechanical efficiency and the indicated efficiency increases with the rotational speed up to 3500 rpm, and decreases when the rotational speed exceeds 3500 rpm. This tendency is probably caused mainly by the change of mechanical efficiency although the mechanical efficiency is assumed to be constant in the calculation. The scroll expander has relatively large frictional loss and when the frictional loss is constant against the rotational speed, the mechanical efficiency increases with the rotational speed. Under high rotational speed, the increase of the frictional loss and the decrease of P - V work due to the inlet pressure loss cause the low efficiency. The total efficiency has a maximum value of 55 % around the 3500 rpm. This value is pretty good with considering that this is the first prototype of the scroll expander and it was made just using the scroll element of the compressor for the water heater without any major modifications.

6. CONCLUSIONS

A feasibility of scroll expander for carbon dioxide refrigeration cycles was examined both theoretically and experimentally. Pressure change in an expansion chamber was analyzed using an apparent expansion exponent and the mass flow rate of the trans-critical leakage flow was calculated based on the enthalpy difference along the isentropic change. The performance of the scroll expander was examined based on the indicated diagram and the mass flow rate. The calculation results showed that the total efficiency of the scroll expander becomes about 60% when the leakage clearance magnitude is 10 μm and the rotational speed is 3600rpm. In addition, the prototype of scroll expander was made using a mechanical element of scroll compressor for a water heater and its performance was measured. The volumetric efficiency of 80 % was obtained and the total efficiency reached about 55 % although the scroll element of the compressor is used without any major modifications.

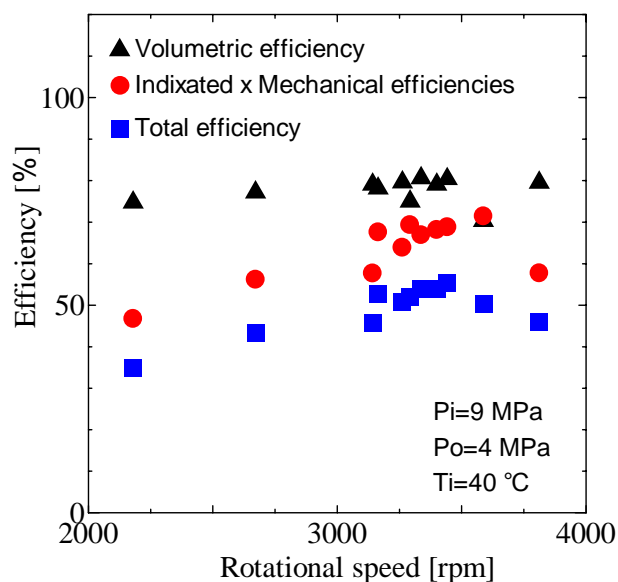


Fig. 11 Experimental results

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