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A STUDY ON THE LEAKAGE CHARACTERISTICS OF TIP SEAL MECHANISM IN THE SCROLL COMPRESSOR

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ABSTRACT

The tip seal as a sealing mechanism to reduce the leakage flow rate through the axial clearance has been widely used in scroll compressors, but the leakage characteristics of tip seal is not well clarified due to the complicated leakage path. This paper discusses the analytical results for four theoretical models used to predict the leakage flow rate and the relationship between the changes of tip clearance and leakage flow rate under various pressure ratios. In the experiment, it was found that the tip clearance increased due to reduction of the tip sealing force that pressed the tip seal against the opposite plate under small pressure difference between upstream and downstream. Among the four models, the predicted result for the nozzle flow model was most similar to experimental result. The equivalent clearance corresponding to the measured leakage flow rate was obtained by using the nozzle model. Additionally, the behavior of tip seal forced in the tip seal groove by pressure difference was measured with a laser displacement sensor to clarify the leakage phenomenon.

NOMENCLATURE

H_g : height of wrap tip groove
H_t : height of tip seal
L_g : width of wrap tip groove
L_t : thickness of tip seal
$\dot{m}$: leakage mass flow rate
P_{back} : groove pressure under tip seal
P_c : compression pressure
P_d : discharge pressure
P_{dn} : downstream pressure
P_s : suction pressure
P_{up} : upstream pressure
ΔP : differential pressure
r_i : inner diameter of circular wrap
r_o : outer diameter of circular wrap
T_{up} : upstream temperature
δ_a : clearance between tip seal and glass plate
δ_s : clearance between wrap tip and glass plate
ε : pressure ratio of P_{dn} to P_{up}

INTRODUCTION

In order to attain the high efficiency of scroll compressors widely used for the air-conditioning and heat pump systems, it is greatly important to minimize the leakage flow through the narrow clearances due to the pressure difference between compression chambers. Especially, when the alternative refrigerant having high-pressure properties such as R410A or CO_2 is applied to the scroll compressors instead of R22, the compressor efficiency decreases due to the increase of the leakage flow caused by large pressure difference between the compression chambers. Up to now, many kinds of the models to predict the leakage flow through the narrow clearances of the compressors have been proposed, and the experimental approaches to validate the analytical models have been tried [1-9].

The scroll compressors having superior compression mechanism to any other compressors have two kinds of leakage flows. One is the radial leakage flow through the axial clearance between the scroll wrap tip and the base plate of opposite scroll, the other is the tangential leakage flow through the radial clearance between the flanks of orbiting and fixed scroll wraps. In order to reduce these leakages in scroll compressors, the sealing mechanisms to control the narrow clearances have been used. Especially, it is more important to control the axial clearance for the high efficiency because the sealing length of the axial clearance is several times longer than that of the radial clearance corresponding to wrap height. Among the sealing mechanisms to control the axial clearance, the tip seal, which is placed in the groove machined on the tip of scroll wraps, has been the most widely used due to the advantages of simple structure and good sealing ability.

In the previous investigations for clarifying the leakage characteristics of tip seal mechanism, Inaba et al. [10] investigated the leakage losses about three leakage paths in the case of inserting the tip seal into a groove tightly or loosely, and they also developed the assembling techniques to insert the tip seal sufficiently close to the bottom surface of groove. On the other hand, Hirano et al. [11] clarified the sealing pressure forcing the tip seal against the base plate of opposite scroll by measuring the groove pressure under the tip seal along the scroll wrap length and measured the tip seal motion with eddy current sensors. Ancel et al. [12] investigated the dynamic behavior of two different types of tip seals, which are the multi-blade and the monobloc tip seals. Youn et al. [13] developed the experimental apparatus to measure the leakage flow using tip seal under the actual operating conditions. However, it is not enough to clarify the leakage characteristics of tip seal because of the complicated leakage path.

In this study, the leakage characteristics of scroll compressors having tip seal as the sealing mechanism for the axial clearance is investigated theoretically and experimentally. The calculation using four kinds of analytical models for the estimation of the leakage flow rate is conducted. The experimental apparatus having a simplified circular wrap is set up to measure the leakage flow rate, the axial clearance and the differential pressure between upstream pressure and groove pressure under the tip seal with the various pressure ratio conditions at the same time. The calculated results are compared with the experimental ones. Moreover, the behavior of tip seal in the tip seal groove is measured directly with a laser displacement sensor to clarify the leakage phenomenon.

THEORETICAL ANAYSIS

The several chambers formed by fixed and orbiting scrolls, which are eccentrically mated, are shown in Figure 1(a). Considering a pair of compression chambers having intermediate pressure in it as shown in Figure 1(b), there are two different kinds of leakage flows to suction pressure chamber through narrow clearances. One is the radial leakage flow through the axial clearance and the other is the tangential leakage flow through the radial clearance.

Several models can be used for calculation of the leakage flow rate through the narrow clearances in scroll compressors. In this study, we have taken into account the following four models that were proposed in the previous works. At first, the most commonly used model to estimate the leakage flow is the nozzle flow model, which is assumed as the compressible and isentropic flow without friction through the throttling area with an appropriate flow coefficient [10]. However, it is demanded to take viscous effects into account since the clearance of leakage channel is very narrow as compared to its length. Therefore the leakage channel that consists of the converging nozzle at flow entrance and the frictional channel having a constant rectangular cross section has been considered and the leakage flow rate in this channel is calculated with Fanno flow model, which is adiabatic flow with fluid friction [1]. Thirdly, we employed the leakage flow model (Ishii's model) based on a simple incompressible and viscous flow theory assuming a fully developed turbulent leakage flow [7]. Finally, the leakage flow is calculated with one-dimensional steady-state laminar flow model (Fagerli's model) with friction between parallel plates [8].

EXPERIMENTAL SETUP

Figure 2 shows the schematic view of an experimental channel formed by the simplified circular wrap, which has a tip seal to protect the radial leakage on the top of wrap. The assembled clearance δ_s exists between a glass plate and the tip of circular wrap, but the characteristics of leakage flow is governed by the tip clearance δ_a, which is much narrower than δ_s due to a tip seal that is pressed against the glass plate by pressure difference between upstream and downstream pressures. Experiments for the present study are performed using the following specification: r_i = 21 mm, r_o = 25 mm, r_{gi} = 22.025 mm, r_{go} = 23.975 mm and δ_s = 83 μm for the circular wrap; L_g = 1.95 mm, H_g = 1.52 mm for the groove machined on the tip of wrap; L_t = 1.9 mm, H_t = 1.4 mm for the tip seal.

Figure 3 shows the schematic diagram of the experimental setup. The working fluid, air, is supplied from the air tank to the test section shown in Figure 2 via a regulator that controls the upstream pressure P_{up}, and then the supplied high-pressure air leaks through the narrow clearance and the leaked air is released to the atmosphere. The downstream pressure P_{dn} is controlled with a control valve installed on the downstream line. The leakage flow rate, the differential pressure between upstream pressure and groove pressure P_{back} under the tip seal and the tip clearance on the tip seal are measured simultaneously with changing the pressure ratio P_{dn}/P_{up}. The upstream and downstream pressures and temperatures are measured by Bourdon-tube pressure gauges and T-type thermocouples, respectively. The leakage flow rate is measured by a floating-type rotameter ranging from 2 to 200 NL/min. A differential pressure gauge is installed between P_{up} and P_{back} to obtain the change of groove pressure under the tip seal according to changing the pressure conditions. The change of tip clearance is measured through the glass plate by using a laser displacement sensor with the resolution of 1μm.

In addition, the supplementary experiment is carried out to clarify the tip sealing phenomenon, as shown in Figure 4. The test section is mounted on the moving table, which is movable with 1μm precision by adjusting a micrometer. The reference surface on the lower surface of the glass plate is constructed with the vacuum evaporation of aluminum. The displacements between this reference surface and the tips of wrap and tip seal are measured along the wrap thickness as traversing the test section, while the laser displacement sensor is fixed.

RESULTS AND DISCUSSION

Calculation Conditions

In this study, the following conditions are used to predict the leakage flow rate through the narrow channel based on the above-mentioned four leakage models. Air (adiabatic index k = 1.4) is used as a working fluid. The narrow channel consists of the clearance δ = 20 μm, length L (= L_t) = 1.9 mm and width W (=$2\pi(r_{gi} + r_{go})/2$) = 144.5 mm. The upstream temperature T_{up} is 25 °C. The upstream pressure P_{up} and downstream pressure P_{dn} range from 0.101 to 0.984 MPa[abs], respectively as fixed the other one. The flow coefficient is set to 1.0.

Theoretical Results

Figure 5 shows the calculated results for the leakage flow rate with the change of pressure ratio ε, which is the ratio of downstream to upstream pressure under the constant upstream pressure P_{up} = 0.984 MPa. The nozzle model neglecting the viscous effects shows the largest leakage flow rate as compared to any other three models in the region of $\varepsilon > 0.8$, where the viscous effects are dominant since the flow velocity through the clearance is small. In this region, the leakages of three models taking account of the viscous effects are nearly the same. On the other hand, the inertial effect becomes greater than the viscous effect when the flow velocity increases with the decrease of pressure ratio, which means that the pressure difference between P_{up} and P_{dn} is large. Thus in the case of the models of Ishii and Fagerli that only consider the viscous effects, the leakage flow rates gradually increase, but in the case of Fanno and nozzle models assuming as the compressible flow, the choking phenomenon occurs when the pressure ratios are less than critical pressure ratio, 0.528 for nozzle and 0.35 for Fanno in this conditions. When choking occurs, the leakage flow rates through a narrow clearance no longer increase and are independent of the change of downstream pressure. Among the leakage models, the Fanno model shows the smallest leakage flow rate under almost all pressure ratio conditions.

Figure 6 shows the calculated results for the leakage flow rate with the change of pressure ratio under the constant downstream pressure P_{dn} = 0.101 MPa. The figure indicates that the flow rate increases with increasing the upstream pressure. In the region where the pressure ratio is greater than 0.4, the leakage flow rate calculated by the models of Fanno and Fagerli are nearly the same and a little smaller than Ishii's model, and the calculated result for nozzle model is larger than Ishii's model. However, the leakage flow rates for the models of Ishii and Fagerli steeply increase as compared with those of Fanno and nozzle models under $\varepsilon < 0.4$ due to the same reasons as mentioned above.

Experimental Results and Equivalent Clearance

The experimental results for the leakage mass flow rate $\dot{m}$, the differential pressure ΔP between the upstream pressure and the groove pressure under the tip seal and the tip clearance δ_a are shown in Figure 7, by using the circular wrap having the tip seal shown in Figure 2. In addition, the equivalent clearances corresponding to the measured leakage mass flow rate were calculated with the nozzle flow model that shows the closest tendency to experimental result among the four analytical models. The experiment has been conducted under the two conditions of case 1 and case 2, which indicate to change downstream pressure P_{dn} from 0.101 to 0.964 MPa increasingly and from 0.964 to 0.101 MPa decreasingly, keeping upstream pressure P_{up} of 0.984MPa. In the case1, both of the leakage flow rate and differential pressure slightly increase and then gradually decrease with the increase of downstream pressure. In the case 2, both rapidly increase and then slowly decrease with the decrease of downstream pressure. The differential pressure is about 5kPa when the downstream pressure is equal to atmospheric pressure and it varies with the change of the leakage flow rate. On the other hand, the tip clearance measured by the laser displacement sensor represents the relative value from the reference clearance, which is set to 0 μm under P_{dn} = 0.101MPa. The tip clearance is almost constant up to P_{dn} = 0.4 MPa, but this increases and then shows a maximum of 22 μm at the largest downstream pressure, which means the smallest pressure difference between upstream and downstream. With decreasing the downstream pressure, the tip clearance goes down.

When the calculated leakage flow rates are compared with the measured ones, the equivalent clearance is about 0.8 μm in the region of small downstream pressures but it increases up to 2 μm as the downstream pressure increases. As a result, the tip sealing force, which presses the tip seal upward against an opposite plate, is reduced with the increase of downstream pressure and the reduced tip sealing force causes the increase of tip clearance. The equivalent clearances are much smaller than measured tip clearances because the flow coefficient is ideally set to 1.0. Applying the flow coefficient of 0.1 based on the experimental results of Youn et al [13], the results would be much closer to measured results. The leakage mass flow rate of the case 2 becomes much lager than that of the case 1 since the tip sealing force is not restored yet in spite of decreasing the downstream pressure, but gradually decreases and then becomes the same value as the case 1 because the tip sealing force is completely restored.

Figure 8 shows the experimental results for the leakage flow rate, the differential pressure and the tip clearance under the conditions of case 3 and case 4, which indicate to change upstream pressure P_{up} from 0.984 to 0.121 MPa decreasingly and from 0.121 to 0.984 MPa increasingly, keeping downstream pressure P_{dn} of 0.101MPa. In the case 3, the measured leakage flow rate decreases linearly with the decrease of upstream pressure and is almost the same as compared with the calculated one with the tip clearance of 0.8 μm, but it is a little far from the calculated one because the tip clearance increases in the region of small upstream pressures. The measured tip clearance is 27 μm at a minimum upstream pressure condition. The differential pressure varies with the change of leakage flow rate as mentioned in Figure 7. The results of the case 4 can be explained with the same principle as the results of the case 2. Conclusively, it was found that the change of tip clearances in the region of small pressure differences has an unfavorable effect on the tip seal leakage.

Leakage Phenomenon

In order to understand leakage phenomenon and tip seal behavior, the change of tip clearances was measured by the laser displacement sensor as traversing the test section forth and back at 0.1mm spacing, as shown in Figure 9. The tip seal is placed in the bottom position in the groove when upstream and downstream pressures are equal to atmospheric pressure. However, in the case of the upstream pressure of 0.984 MPa, the tip clearance decreases since the tip seal is pressed against the glass plate. The surface of the tip seal has some roughness as shown in Figure 9, and the roughness has a direct effect on the equivalent clearance, which affects the leakage flow rate.

CONCLUSIONS

The experimental setup has been developed to clarify the leakage characteristics of the tip seal as an axial sealing mechanism in scroll compressors by measuring the leakage flow rate, the differential pressure between upstream pressure and groove pressure under the tip seal and the tip clearance on the tip seal. The four kinds of analytical models to predict the leakage flow rate were reviewed, and the calculated results were compared with the experimental ones. When the upstream and downstream pressures are increased or decreased respectively with fixing the other one, there was the difference between the measured leakage flow rates. The experimental results for the leakage flow rate have rather similar tendency with the analytical results calculated by the nozzle flow model with constant clearance, but become larger due to the increase of tip clearance in the region where the pressure difference between upstream and downstream is small. In addition, in order to clarify the leakage phenomenon and the tip seal behavior, the change of tip clearances was measured with a laser displacement sensor along the wrap thickness as traversing the test section. The measured tip seal surface has some roughness, and the roughness has a direct effect on the equivalent clearance, which may affect the leakage flow.

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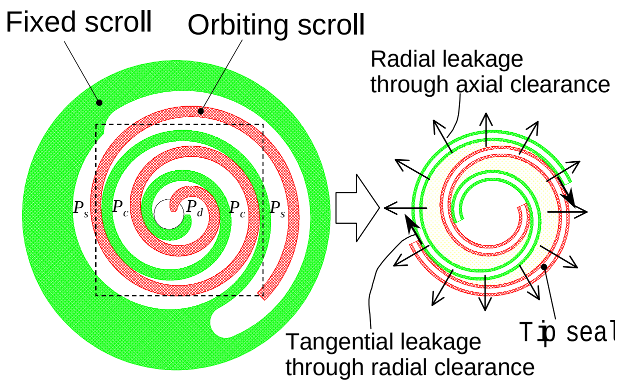

(a) (b)

Figure 1 Combination of scroll wraps (a) and leakage paths of a pair of intermediate compression chamber (b)

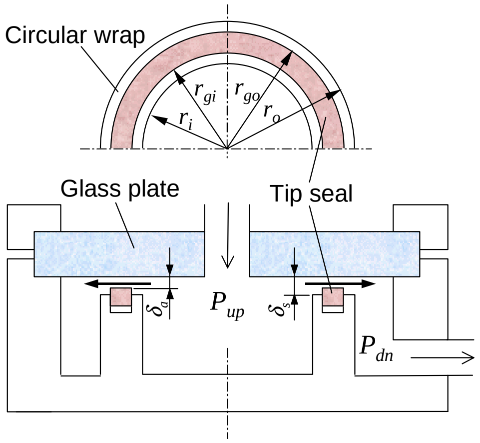

Figure 2 Schematic view of an experimental channel

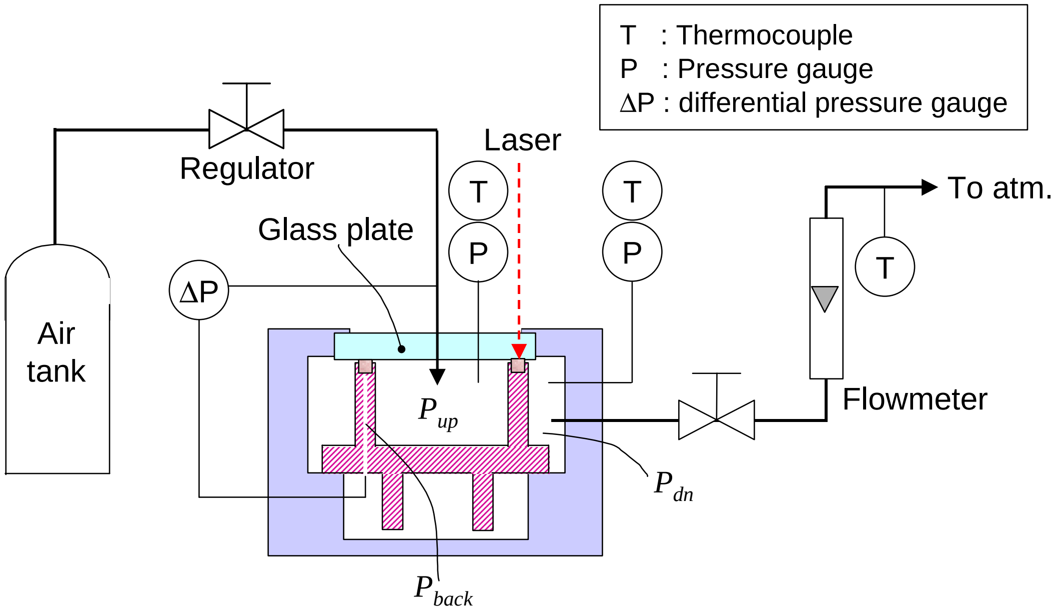

Figure 3 Schematic view of experimental setup

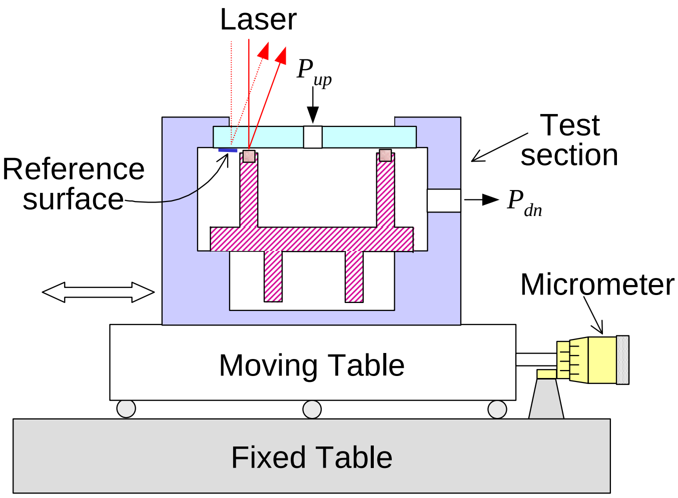

Figure 4 Experimental setup to measure tip seal behavior

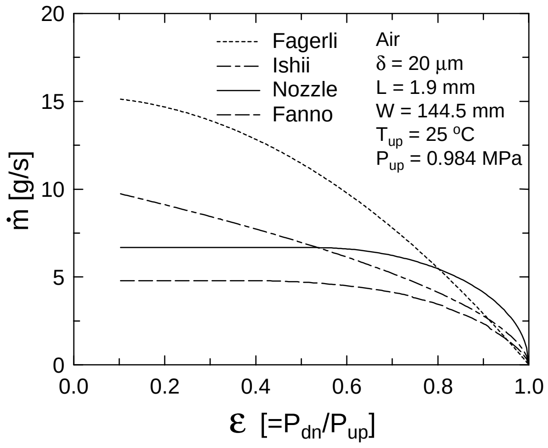

Figure 5 Calculated results for leakage flow rate under P_{up} = 0.984 MPa and P_{dn} = 0.101-0.984 MPa

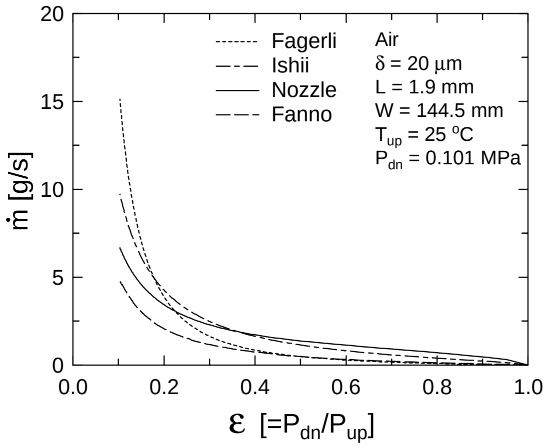

Figure 6 Calculated results for leakage flow rate under P_{up} = 0.984-0.101 MPa and P_{dn} = 0.101 MPa

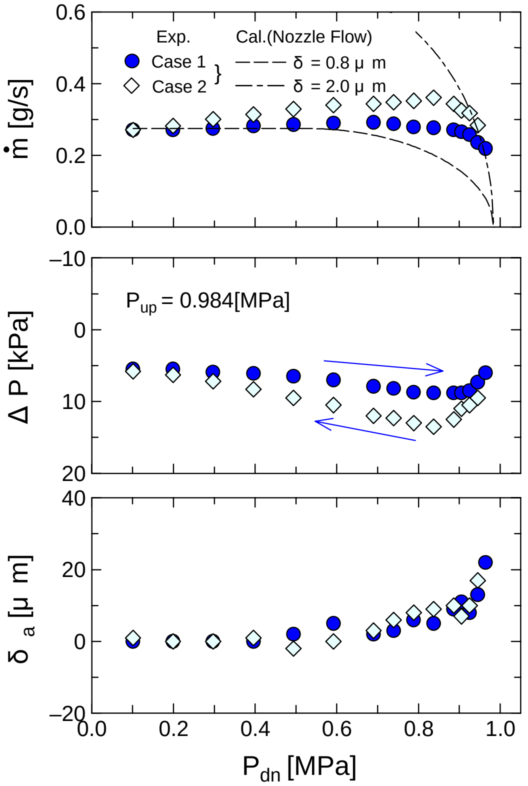

Figure 7 Experimental results for leakage flow rate, the differential pressure and axial clearance under the cases 1 and 2, and equivalent clearances calculated by nozzle flow

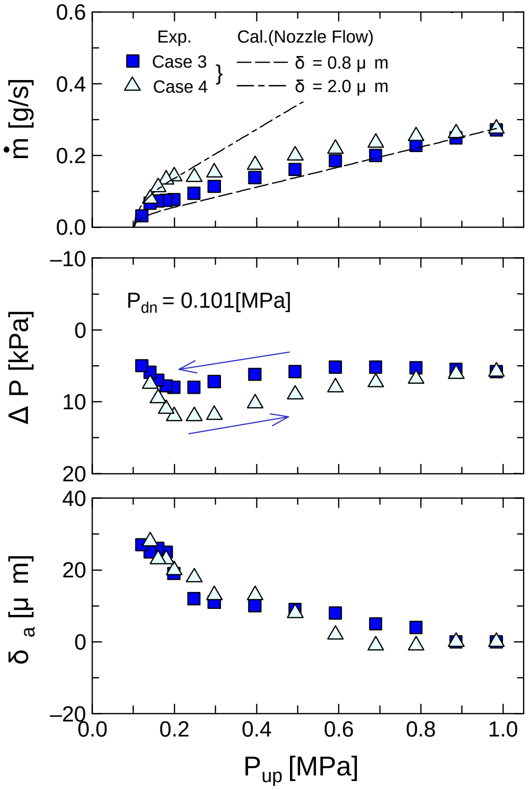

Figure 8 Experimental results for leakage flow rate, the differential pressure and axial clearance under the cases 3 and 4, and equivalent clearances calculated by nozzle flow

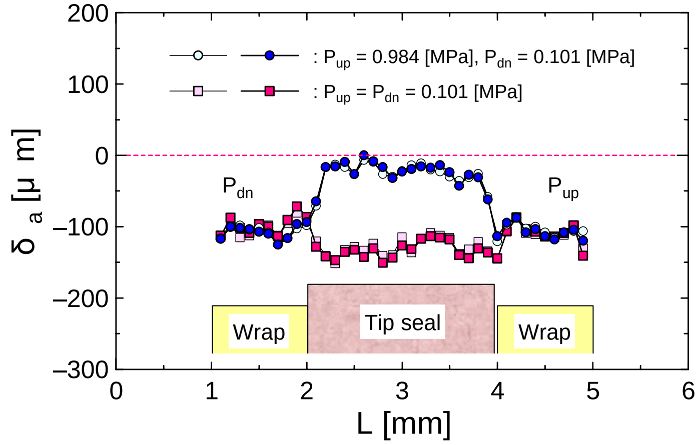

Figure 9 Change of axial clearances across wrap thickness