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Phenomenon of Labyrinth Seal with Low Static Pressure Difference and Large Clearance

(Prediction of Performance and Experimental Result)

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Abstract : The low pressure axial flow fans with an outer ring, used for cooling automobile radiators, have a significantly large tip clearance between the ring tip and the fan shroud. It has been found that the turbulent reverse flow, or leakage flow, which occurs at the tip clearance, greatly affects the fan performance and noise level. Therefore, in order to improve the fan performance and noise level it is important to decrease the effect of leakage at the tip clearance. The authors investigated the performance of the straight-through type of labyrinth seal which operates in an extremely low static pressure difference with a large clearance. It was hoped that by sealing this clearance with the labyrinth seal the performance would be improved. It was verified that the labyrinth seal satisfied almost the same performance as that predicted by the previous theory. This theory was established by experimental studies in the condition of quite high static pressure difference when the labyrinth is stationary. However, it was later discovered that the leakage rate decreased significantly even though there was far lower ring speed in comparison to past research results where the ring rotated. This phenomenon is conspicuous in a lower differential pressure. However, the cause of this phenomenon has not been determined.

Keywords : Fan, Tip clearance, Labyrinth seal, Visualization

Nomenclature:

D	Diameter of the rings	(mm)	[constant : 362]
F	Opening area of the throttling	(m^2)	
G	Leakage rate	(kg/s)	
L	Axial length of the rings	(mm)	[constant : 49.7]
ℓ	Pitch of fins	(mm)	
n	Number of fins		
P	Absolute pressure	(Pa)	
Ps	Static pressure difference	(Pa)	
S	Inner distance between fins	(mm)	
U	Peripheral velocity of fin tip	(m/s)	
α	Flow coefficient of contraction at the throttling		
Γ	Specific leakage rate		$\Gamma=G_{Er}/G_{Es}$
$\delta 1$	Thickness of first and end fins	(mm)	[constant : 2.5]
$\delta 2$	Thickness of fins	(mm)	[constant : 1.5]
ε	Clearance	(mm)	[constant : 3.0]
θ	Jet expansion angle	(deg)	
λ	Pressure ratio		$\lambda=P_D/P_U$

ν	Specific volume of gas	(m^3/kg)
υ	Carry-over factor	$\upsilon = G_E/G_I$
Φ	Labyrinth function	
ϕ	Ideal labyrinth function	

Subscripts

a	Approximate	
D	Outlet of labyrinth	(Downstream)
E	Experimental	
I	Ideal	
R	Rotational	
S	Stationary	
U	Inlet of labyrinth	(Upstream)

1. Introduction

Two types of axial flow fans are used for automobile radiators. The first type is the axial flow fan which has nothing attached to its blade tip. The second type is the axial flow fan which has an outer ring attached to its blade tip. In the case of an axial flow fan without the ring, it is known that the reverse flow and the leakage, which occur at the tip clearance between the blade tip and the fan shroud, affect fan performance. Regarding this, it is reported that the difference of the tip clearance greatly influences both the performance and the noise level in several types of axial flow fan (Fukano et al., 1985). In addition, the authors have also shown that the increase of the tip clearance in the axial flow fan for a motorcycle radiator degenerates the performance and the noise level. Moreover, it is reported that the shroud shaped into a ring attached to the axial flow fan improves the fan performance in the case of large tip clearance (Shimada et al., 2001). However, in the case of the ring-attached axial flow fan for an automobile, there is clearance between the ring and the shroud, like the tip clearance of the fan without a ring. Also in this case, the reverse flow or the leakage flow influences the fan performance, like the case of the fan without a ring (Shimada et al., 2002). Therefore, the authors aimed at the labyrinth seal as a clearance seal, in order to seal the fan's tip clearance then improve the fan performance.

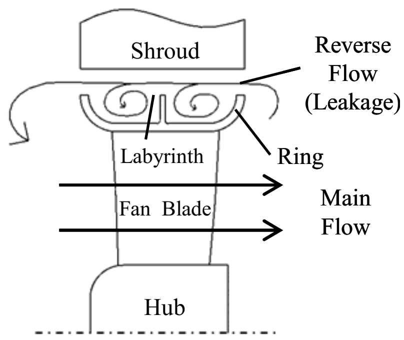
Generally, the labyrinth seal mechanism used in compressors or turbines operates in very high static pressure difference ($1.0x10^2 \sim 3.0x10^4$kPa, at maximum) and at a very small clearance (0.1 ~ 0.6mm). However, in the case of the ring-attached axial flow fans used in this research, they were mass-produced by injection molding with plastic which caused difficulties in keeping the desired clearance size. Therefore, it is difficult to keep the clearance very small. Generally, the clearance of these fans is large and is about 3mm. Furthermore, the static pressure difference at the operating region of these fans is extremely low, nearly 150Pa.

The performance of the labyrinth seals in the case of compressors or turbines was previously reported(e.g., Komotori and Mori, 1971; e.g., Komotori and Miyake, 1977). These experiments were conducted under high static pressure difference as the actual operating state. According to these reports, it is possible to predict the performance of the labyrinth seal using equations of approximation. Furthermore, the reports state the influence of the rotation on the performance is small. However, there are few reports about labyrinth seals which operate in the condition of an extremely low static pressure difference and a large clearance like the operating condition for the automobile's fans. The purpose of this report is to clarify the performance of the straight-through type labyrinth seal used in the unique condition mentioned above. The results of this study prove by using equations from past research that it is possible to predict the seal performance when it is stationary. The performance experiments and the flow visualizations were conducted in order to lead these results. However, a peculiar phenomenon different from results of past research was observed when the ring was rotating.

2. Operating Region and Configuration of the Labyrinth Seal

Figure 1 shows the structure of the labyrinth in the ring and the state of the reverse flow of the ring-attached axial flow fan, which is the subject of this research. By constructing the labyrinth in the ring, it seems to be possible to reduce the leakage flow occurring at the clearance area between the shroud and the ring and thus improve the performance. However, as it was mentioned in the previous chapter, the static pressure difference as the operational pressure of the research fan is significantly low and the sealing clearance is large. This is why the study area of this labyrinth is remarkably different from the area of past research. Figure 2 illustrates each study's area. The figure indicates the static pressure difference in its ordinate and the clearance in its abscissa. There are few examples researched and the results were publicized in the study area covered by this research.

(a) Bird's-Eye view (inlet side)



(b) Relation between structure and flow

Fig. 1. Axial flow fan with a ring and labyrinth seal.

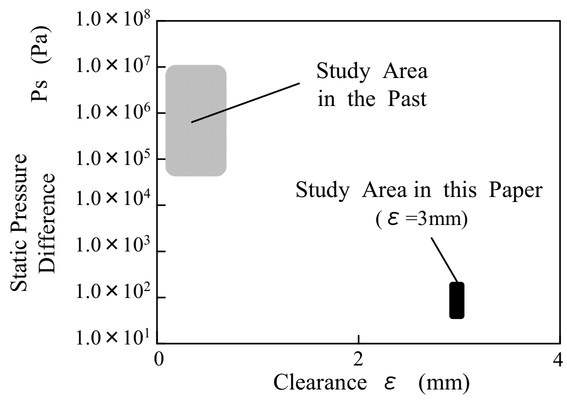
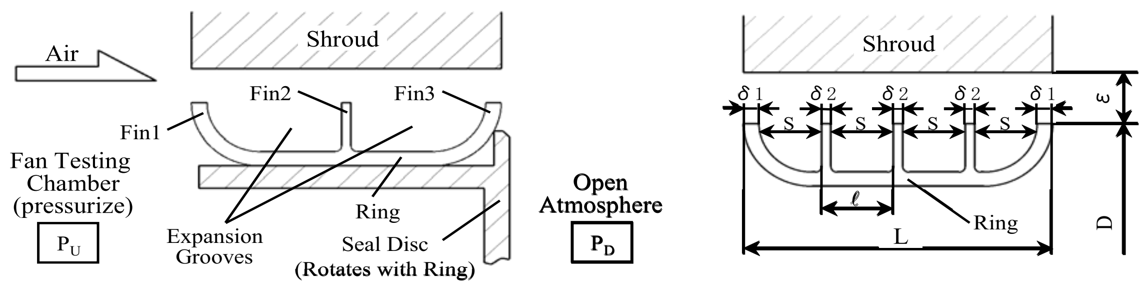


Fig. 2. Difference of study area.

Figure 3 shows an example of the form of labyrinth seal used in this research. This labyrinth seal is formed in the linear channel between the inlet and the outlet. Furthermore, the linear channel is not disturbed anywhere on the channel. This type of labyrinth seal is called a straight-through type. Such labyrinths are widely used because their construction and assemblage are very easy. The authors conducted research using the rings which have 2, 3, and 5 fins. Figure 3 (a) shows the case when the number of fins is n=3. It illustrates a labyrinth seal consisting of three fins and two expansion grooves between the fins. On the other hand, Fig.3 (b) shows the case when the number of fins is n=5. The radial height of each fin is constant. The tip end diameter of each fin is always 362mm. The clearance ε between the inner surface of the cylindrical shroud and the fins is 3mm. The axial length of the rings L does not depend on the number of fins and is a constant equal to 49.7mm. In the case of the same labyrinth, the pitch of fins ℓ is constant. The same condition applies to the inner distance S. The bell mouth parts at the inlet and outlet sides of the rings are uniform in their shapes.

In the experiments for the labyrinth's seal performance, the place for the fan is closed with a seal disc as shown in Fig.3 (a). There is no leakage in this part. Then, it was attached to the fan testing chamber, and the static pressure was increased at the upstream side while opening the downstream side to the atmosphere. The leakage rate was measured per given pressure in

Ps=50~250Pa. The ring and the seal disk were constructed as one whole and it was possible to rotate them together. Therefore, the leakage rate was measured both when the ring was stationary and rotating. The rotating speed was increased each time by 500rpm from 0rpm, till it reached 2500rpm in the constant pressure.



(a) Test rig (n=3) (b) Specifi cation of the ring (n=5)

Fig. 3. Configurations of the labyrinth seal and the test rig.

3. Prediction of Seal Performance and Experimental Results

3.1 Prediction of Seal Performance by Past Approximate Equations

In the case of an ideal labyrinth seal, when the fluid passes through the labyrinth throttling, complete adiabatic expansion occurs. The kinetic energy developed when passing through the throttling, is completely transformed to thermal energy in an expansion groove. As a result, there is no velocity right before the next fin, and also there is no loss of energy anywhere. This is the definition of the operating conditions of an ideal labyrinth seal. Approximation equations are often used because repeated computer calculations are necessary in order to solve the value of the ideal labyrinth function ϕ. Several approximation equations for the ideal labyrinth function ϕ have been proposed, but Komotori's equation (1) was used in this research as it offers better precision (Komotori, 1955).

$$\phi_a = \frac{\sqrt{1-\lambda^2}}{n^{2/5}}$$ (1) Komotori's equation for approximate ideal labyrinth function ϕ_a

In the case of a straight-through type labyrinth seal as shown in Fig.4, the clearance part is formed in a linear channel. Therefore, the flow in the labyrinth does not effectively expand in the expansion grooves but sometimes simply passes through the channel. This phenomenon is called a carry-over and it is known that this influence deteriorates the performance of the labyrinth (Komotori, 1957). When there is a carry-over in the straight-through type labyrinth, its influence can be expressed with the carry-over factor $\upsilon = G_E/G_I$. G_E is the leakage rate of a labyrinth seal with a carry-over which is obtained experimentally. G_I is the leakage rate of the ideal labyrinth without any carry-over. Equation (2) shows an approximation equation. When the static pressure difference is low and the pressure ratio λ is approaching 1, giving a solution very close to the strict solution (Komotori and Mori, 1971). In this report, λ is very close to 1. Therefore, Eq.(2) is very useful for this research. Here, θ is the angle of jet expansion which starts from the contraction near the fin tip as shown in Fig.4.

$$\left.\begin{aligned} \upsilon &= \sqrt{\frac{n}{1+(n-1)(1-A)^2}} \\ A &= \frac{\varepsilon\alpha/\ell}{(\varepsilon\alpha/\ell)+\tan\theta} \end{aligned}\right\}$$ (2) Komotori's equation for carry-over factor υ

Here, the experimental leakage rate is expressed with Eq.(3). The labyrinth function Φ is expressed with Eq. (4) as a function of α and υ. Therefore, it is possible to lead to Eq.(5), in order to express Φ with the experimental leakage rate G_E.

$$G_E = F\upsilon\alpha\phi_a\sqrt{\frac{P_U}{\nu_U}} \quad (3)$$

$$\Phi = \upsilon\alpha\phi_a \quad \text{(4) Predicted value}$$

$$\Phi = G_E\left(F\sqrt{\frac{P_U}{\nu_U}}\right)^{-1} \quad \text{(5) Experimental value}$$

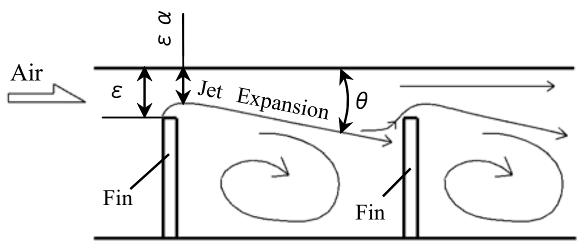


Fig. 4. Flow model of straight-through type labyrinth seal.

3.2 Comparison of Seal Performance between Prediction and Experimentation

These approximation equations proposed by Komotori are derived from the experimental data in the case of high static pressure difference and small clearance. Therefore, it is not clear whether these approximation equations can be used to predict the leakage rate for this study. In order to ensure the adaptability of these approximation equations to the study area of this research, the calculation results and the experimental results were compared. This was made under the following conditions: the number of fins n=2, 3, 5, the static pressure difference Ps=50Pa~250Pa, for the shape of the labyrinth seal shown in Fig.3.

Figure 5 shows the results obtained from Eq.(4). In this figure, the curves were calculated with the flow coefficient $\alpha = 0.7$ and the angle of jet expansion $\theta = 6°$ as the values used by Komotori (Komotori, 1957). It can be derived from Fig.5 that the labyrinth function Φ is the smallest when n=2, consequently, the leakage rate is the smallest. The leakage rate is also small when n=3. However, both results are very close and the performance is almost the same. When n=5, the value of Φ is the greatest. The increase of the stage number of the labyrinth is supposed to improve the performance of the seal. However, the reverse happens in this case and the performance decreases. The reason is that the carry-over factor υ expressed in Eq.(2) increases together with n, and this results in an increase in Φ.

Figure 6 shows the experimental results when the ring is stationary. The tendency of these results corresponds well with the calculation results of Fig.5. The relative performance for each number of fins can be closely predicted, even though Fig.5 shows lower values especially when the static pressure difference Ps is high. Therefore, it can be concluded that it is possible to use the approximation equations of past research even in the case of an extremely low static pressure difference and a large clearance when there is no rotation. The two-dotted chain line shows the characteristics of a simple annular passage without expansion grooves as the case when n increases boundlessly and the ring becomes solid. In this case, it shows the greatest Φ. Therefore, even though under the conditions of an extremely low static pressure difference and a large clearance, the expansion grooves operate effectively. Consequently, increasing n will lead to the inefficient operation of the expansion grooves. Then the performance gradually decreases and approaches the curve of an annular passage.

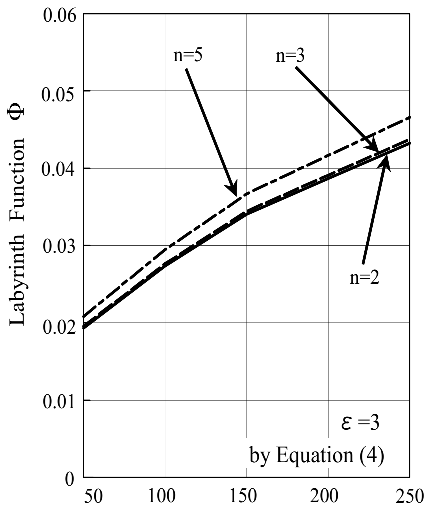


Fig. 5. Seal performance by prediction.

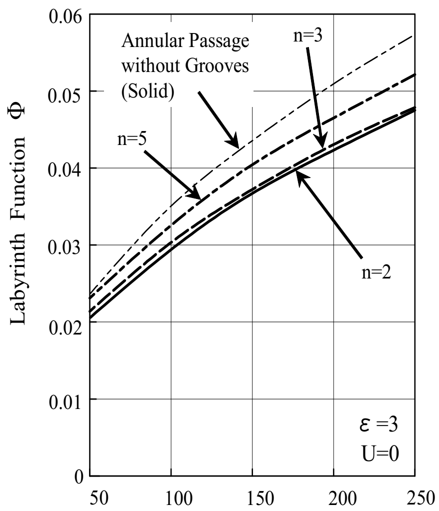


Fig. 6. Seal performance by experimentation.

4. Visualization of Internal Flow

It can be judged that the increase of n leads to the increase of Φ from the previous chapter both theoretically and experimentally. That means, when n increases, the carry-over also increases and as a result the leakage rate increases too. In order to clarify the internal flow when n increases, an experiment of flow visualization was conducted by the surface floating tracer method. The approximate equations for the performance prediction in the previous chapter have been lead from the test results in the case of the stationary condition. In this respect, the effect of rotation of labyrinth seal can not be given for this visualization experiment. Consequently, the two-dimensional models, which neglected the rotational effects, are used in order to depict the internal flow for the visualization. During the experiment, fine aluminum powder was left floating on the water surface of a pool. Next, the trace of the floating aluminum particles was shot with a slow shutter speed camera. The Reynolds number was made the same as in the actual device in order to be able to accurately reproduce the flow. The size of the visual model is ten times bigger than the size of the actual device.

Figure 7 shows the results of the visualization when n=2. According to the picture of Fig.7(a), the internal flow follows along the shroud wall's surface, but then separates at the point about 2/3 from the inlet. The separated flow diffuses in the expansion groove and while one part of it flows toward the outlet, another part of it forms an eddy in the central part of the expansion groove. In other words, the flow expands well and the expansion groove works effectively. Figure 7(b) shows the flow at the inlet throttling, and Fig.7(c) shows the flow at the outlet throttling. Contraction of the flow can be seen both in the inlet and outlet. Especially at the outlet side, the contraction followed by a large eddy on the fin tip can be seen clearly. As it was shown in the calculation results of Fig.5, the flow coefficient of contraction α is a general representative number for the whole labyrinth and is empirically set to 0.7. However, this does not mean that the flow coefficient of contraction is the same at each throttling. As clearly shown in Fig.7, the size of the contraction is different at each inlet and outlet.

Figure 8 shows the results of the visualization in the case of n=3. As shown in Fig.8(a), even though there is an eddy formed in each expansion groove, most of the flow goes through along the shroud wall surface and is carried over to the outlet. In the case of n=2, the flow separates from the shroud surface and expands well in the expansion grooves. However, in the case of n=3, such a

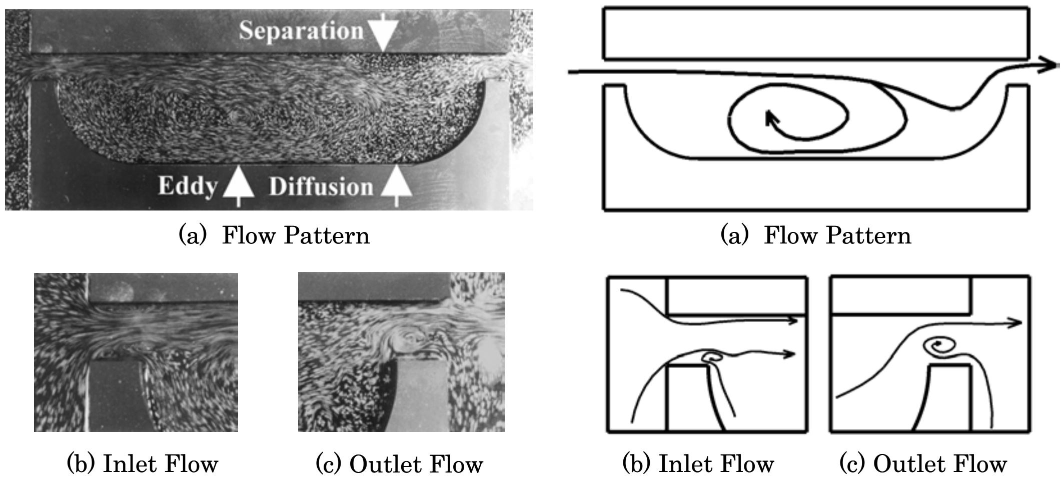


Fig. 7. State of internal flow by surface floating tracer method with illustrations (n=2).

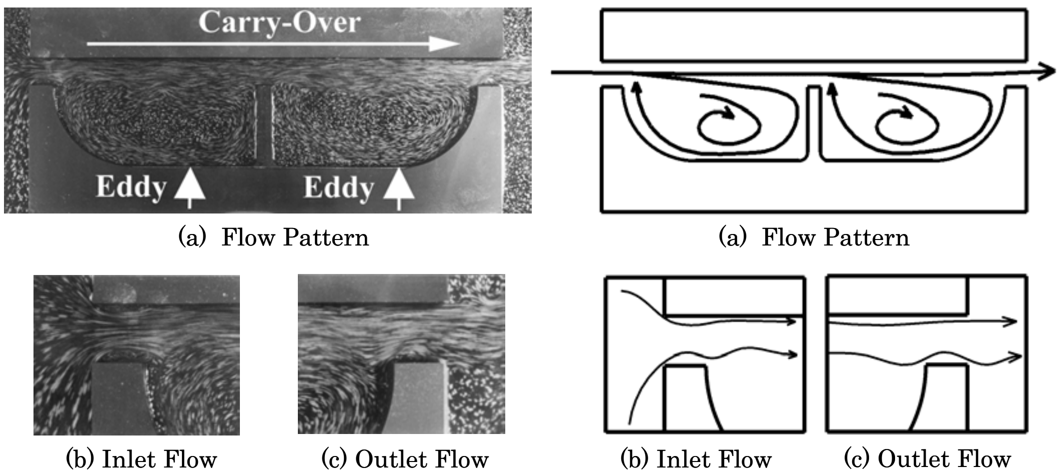


Fig. 8. State of internal flow by surface floating tracer method with illustrations (n=3).

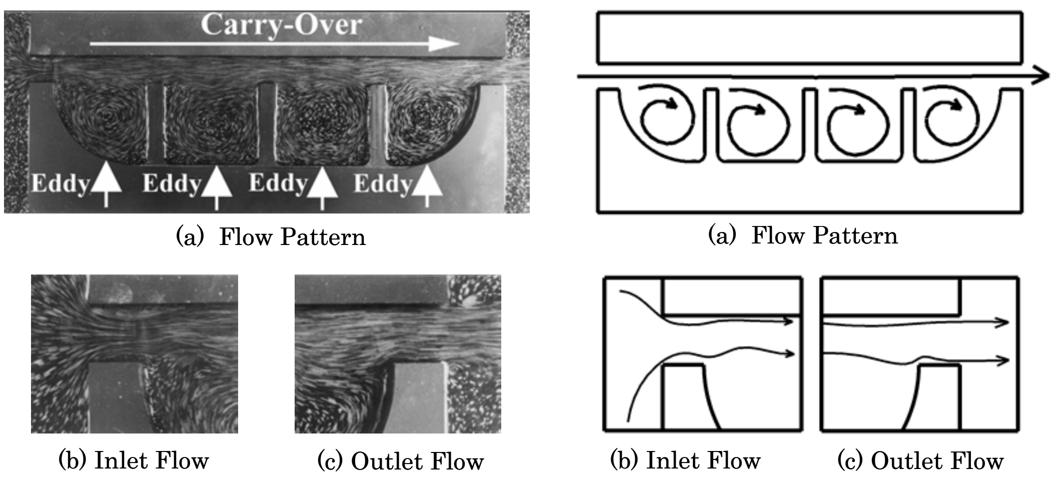


Fig. 9. State of internal flow by surface floating tracer method with illustrations (n=5).

phenomenon is not observed, but the carry-over increases instead. Figure 8(b) shows that there is no significant difference in the condition of the contraction at the inlet compared with the case of n=2. However, as shown in Fig.8(c), the contraction at the outlet becomes significantly small with the increase in the carry-over. In other words, an increase of n leads to an increase of the flow coefficient of contraction at the outlet side in this case.

Figure 9 shows the results of the visualization in the case of n=5. The form of n=5 is the form where each expansion groove of n=3 has been divided into halves by a fin. As shown in Fig.9(a), even though an eddy is formed in each expansion groove, almost all of the flow goes through along the shroud wall surface and carries over. The contraction at the inlet, which is shown in Fig.9(b), is not much different from the case of n=2 or n=3. However, the outlet contraction shown in Fig.9(c) almost disappears.

As the result of this visualization, it is clear that the increase of n leads to the increase of the carry-over. Therefore, this is the reason of the decrease in performance shown in Fig.6. On the other hand, it can be said that the calculation results in Fig.5 are basically correct, because they include the real phenomenon observed from this visualization. However, Fig.5 does not take into consideration the above-mentioned relationship that α changes with a change in n. As a result, the predicted value of Φ when n=5 is 7.7% higher than the value of Φ when n=2, in Fig.5. However, experimental value of Φ when n=5 is from 9.5% to 12.1% higher than the value of Φ when n=2, in Fig.6. In other words, when n is increased from 2 to 5, the performance decreases greatly in the case of experiment than in the case of prediction. Flow visualization shows that the value of α also increased with the increase of n in the actual case different from the case of prediction with Eq.(4). It is thought that this is the cause of the difference between the prediction and the experiment mentioned above.

5. Effect of Rotation

The rotating condition is the expected operating condition for the labyrinth seal because the labyrinth seal studied in this report is used for a ring-attached fan. The report of Komotori investigates the case of a rotating labyrinth seal and the experiment is conducted when the pressure ratio λ is about 0.5 (Komotori and Miyake, 1977). According to this report, when the peripheral velocity of fin tip U is low, no significant decrease of the leakage rate is observed. However, the leakage rate gradually begins to decrease when U=50 m/s. When U reaches 250 m/s, the leakage rate is reported to decrease about 10~20% in comparison with the stationary condition. Meanwhile, in the case of the labyrinth seal used in this author's report, the maximum number of revolution is 2500 rpm, and the peripheral velocity of fin tip is U=47.4 m/s. Therefore, even when the peripheral velocity is at the maximum, it is still lower than U=50 m/s. If predictions are made by using Komotori's research results, a significant decrease in the leakage rate can not be expected.

Figure 10 shows the seal performance when the ring rotates in U=47.4 m/s. According to the figure, in any cases of n, Φ decreases significantly when rotation is applied in comparison with Fig.6. For example when n=2, Φ drastically decreases in the whole region of Ps in comparison with the case of U=0 m/s. In Fig.6, when n=3, Φ was a little higher in the whole region than the case of n=2. However, in Fig.10, the value of Φ is lower than n=2 when Ps is below 100Pa. Furthermore, when n=5, the value of Φ approaches the results of n=2 more than the case shown in Fig.6, and then they have almost the same value when Ps=50Pa. Figure 11 shows the change of the leakage rate by rotation with U and the specific leakage rate Γ. Here, Γ is expressed as $\Gamma = G_{Er}/G_{Es}$. G_{Er} is the leakage rate when rotation is applied and G_{Es} is the leakage rate when the ring is stationary. Figure 11(a) shows the case of n=2. Γ decreases with an increase in U in the case of any Ps. Especially when the static pressure difference is low, Ps=50Pa, Γ decreases significantly. Figure 11(b) shows the case of n=3. There is no large difference in the value of Γ when Ps=250Pa compared with the case of n=2. However, when Ps=150Pa, the value of Γ is slightly smaller than the case of n=2. When Ps=50Pa, it becomes even smaller than the case of n=2, and when U=47.4 m/s, the value of Γ reaches 0.56. Next, Fig. 11(c) shows the case of n=5. The value of Γ begins decreasing even at the highest Ps=250Pa, compared with the case of n=2 or 3. When Ps=150Pa, the value of Γ decreases slightly and continuously. However, when Ps=50Pa, almost no change is observed in the value of Γ compared with the case of n=3. Therefore, it can be said that Γ saturates when Ps=50Pa.

In other words, with the maximum Ps=250Pa, Γ begins to decrease for the first time when n increases to 5. When Ps=150Pa, the increase in n leads to the gradual decrease of Γ. When the minimum Ps=50Pa, significant decrease of Γ is observed even when n increases from 2 to 3, but no change is observed when n=5. Consequently, the influence of the rotation is stronger when Ps is lower. Also, within the limits of this experiment, this tendency is more significant when n is larger. However, there is a possibility to have a saturation region where Γ no longer decreases even if n is increased.

As mentioned above, the influence of rotation is very strong and the significant decrease of the leakage rate is observed from much lower peripheral velocity than the result of the past research when the static pressure difference is extremely low. However, it is unclear why the influence of rotation can cause such considerable reduction in the leakage rate. It is necessary to analyze and depict the internal flow of the labyrinth seal in order to solve this problem. However, most previous visualization studies have been conducted using the two-dimensional test rig with a smoke (Komotori, 1957) and with a schlieren system to use of a Freon doping technique (Tipton et al., 1986) and so forth, to neglect the effect of rotation. Several previous visualization studies have been conducted in the rotating condition by the oil-paint method (Miyake et al., 1986) or the bubble in the water (Iwatani et al., 1983) for example. However, it is not very easy to clarify the detailed behavior of the complicated three-dimensional flow in the rotating labyrinth seal by these attempts of visualization experiment. Also, these studies were conducted in the condition of a high static pressure difference and a small clearance, different from the condition of this study. Therefore, the authors are going to conduct the visualization study using the computational fluid dynamics (CFD) in order to clarify the detailed internal flow on the rotating condition, as well as to solve the reason for the significant decrease in the leakage rate in the sequel (Shimada et al., 2004).

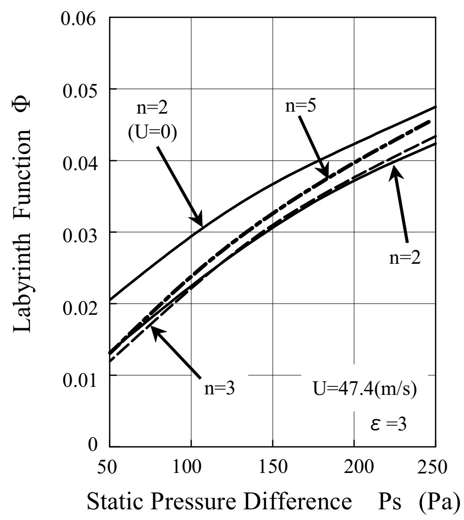
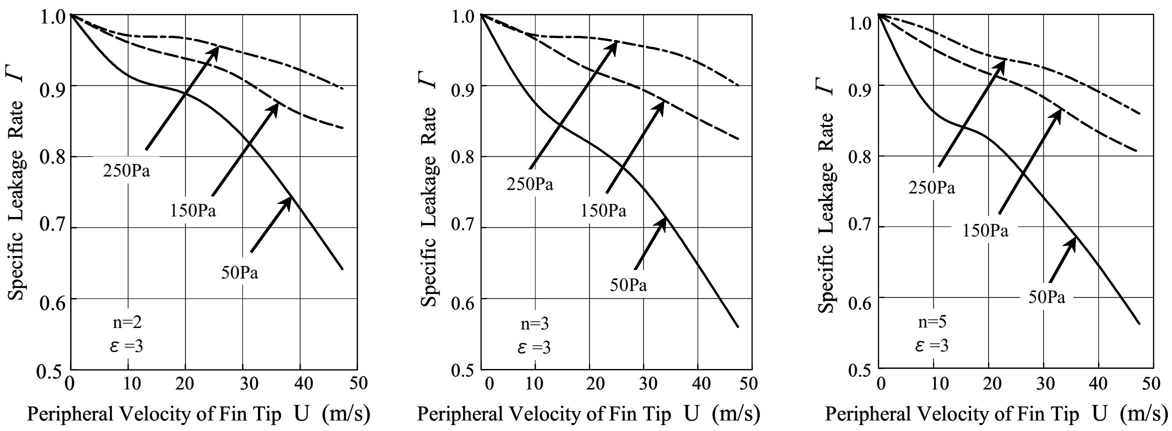


Fig. 10. Seal performance when the ring is rotating.



(a) In the Case of n=2 (b) In the Case of n=3 (c) In the Case of n=5

Fig. 11. Relation between Γ and U with different static pressure difference.

6. Conclusions

An experimental study of the Labyrinth seal which operates in the unique condition of an extremely low static pressure difference and a large clearance was conducted. This labyrinth seal is used in order to seal the clearance between the inner wall of the fan shroud and the ring which is fixed at the blade tip of an axial flow fan for automobile's radiator cooling. The following conclusions were reached in this study.

When the ring is stationary, it is possible to use the existing approximation equations of past research in order to predict the performance of the labyrinth seal. This was proven both by performance experiments and flow visualization.

However, when the ring is rotating, the seal performance was found to be greatly influenced by the rotation. When the static pressure difference Ps=250Pa and the number of fins n=3 then the peripheral velocity of the ring tip U=47.4 m/s, the specific leakage rate reaches Γ=0.90. Furthermore, when Ps=50Pa and the other conditions are the same as the above-mentioned, Γ decreases greatly then reaches Γ=0.56. Also, there is a tendency of Γ to decrease further when the number of fins n increased within the limits of the experiments. However, the results of the experiments suggest the possibility that the decrease of Γ saturates when n increases to some degree.

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References

Fukano, T., Kodama, Y. and Takamatsu, Y., Effect of Tip Clearance of Low Pressure Axial Flow Fan on Fan Noise, Trans. of JSME, (B), 51-463 (1985), 820.

Shimada, K., Hagiwara, H., Hasumi, H. and Ohkubo, T., A study of Radiator Cooling Fan for Motorcycles, Trans. of JSAE, 20014447, 32-3 (2001), 75.

Shimada, K., Kimura, K. and Watanabe, H., A study of Radiator Cooling Fan with Labyrinth Seal, Trans. of JSAE, 20024053, 33-1 (2002), 63.

Komotori, K. and Mori, H., Leakage Characteristics of Labyrinth Seals, Proc. of 5th International Conference on Fluid Sealing British Hydromechanics Research Association, E4 (1971), 45.

Komotori, K. and Miyake, K., Leakage Characteristics of Labyrinth Seals with High Rotating Speed, Proc. of Tokyo Joint Gas Turbine Congress, ASME-JSME-GTSJ, 45 (1977), 371.

Komotori, K., Leakage of Air through Labyrinth Packing(On the Straight-through Type) 1st Report, Trans. of JSME, (3), 21-105 (1955), 377.

Komotori, K., A Consideration on the Labyrinth Packing of Straight-through Type, Trans. of JSME, 23-133 (1957), 617.

Komotori, K., Leakage of Air through Labyrinth Packing (On the Straight-Through Type) 4th Report, Trans. of JSME, (3), 23-129 (1957), 330.

Tipton, D., Scott, T. and Vogel, R., Labyrinth Seal Analysis (Analytical and Experimental Development of a Design Model for Labyrinth Seals) Technical Report of Air Force Wright Aeronautical Laboratories, AFWAL-TR-85-2103, 3 (1986), 1.

Miyake, K., Watanabe, E. and Ariga, I., Leakage Characteristics of Rotating Stepped Labyrinth Gas Seals, SAE Paper , No.861848 (1986).

Iwatani, T., Motooka, N. and Kawai, R., Instability of a Rotating Shaft with Labyrinth Seal, Trans. of JSME, (C), 49-441 (1983), 719.

Shimada, K., Kimura, K., Ichikawa, S., Ohta, H. and Aoki, K., Phenomenon of Labyrinth Seal with Low Static Pressure Difference and Large Clearance, Analysis of the Internal Phenomenon with CFD, Journal of Visualization, 7-1 (2004).

Author Profile

Kota Shimada: He obtained his B.S. degree in mechanical engineering from Tokai University in 1992. He has worked for Toyo Radiator Co.,Ltd. as an engineer and has been engaged in the development of cooling fan systems for radiators since 1992. All japanese motorbike companies and some japanese automobiles companies currently use his fan design. His professional interests are focused on the design and analysis of axial flow fans which operate under unusual conditions.

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