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Testing for the Effect of Groove Depths on the Rotordynamic Coefficients and Leakage of Circumferentially-Grooved Seals

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Circumferentially-Grooved Seal의 동특성 계수 및 누설량에 미치는 Groove 깊이의 영향에 관한 실험

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요 약

서로 다른 그루브 깊이를 갖는 circumferentially-grooved 씨일들의 동특성 계수 및 누설량에 대한 실험결과를 분석하였다.

실험된 결과를 분석해 보면, 그루브 깊이가 증가함에 따라, 누설량이 서서히 증가함을 알 수 있다. Smooth 씨일 및 damper 씨일과 비교할 때, smooth 씨일은 최대의 direct stiffness를 가지며, 평균 씨일 간극의 증가함에 따라 direct stiffness가 감소한다. Effective damping에 있어서, circumferentially-grooved 씨일은 최소의 값을 가지는 반면, 누설량은 가장 적다. 이론과의 실험치 비교는 일반적으로 만족할만한 결과를 보인다.

I. INTRODUCTION

One of the most common seal designs in commerical pumps is a circumferentially-grooved seal. Unlike smooth seals and damper seals[1,2] which employ deliberately homogenous-roughened stators and smooth rotors, grooved seals are characterized by surface roughness pattern of inhomogeneous directivity, viz., groove patterns generate directivity in wall shear stresses. The authors developed an analysis for grooved turbulent annular seals in reference[3] which combines a simple extension of Hirs' turbulent lubrication theory[4] with a "fine-groove" theory[5] for bearings which can handle both homogeneous(smooth and damper seals) and inhomogeneous(grooved seals) surface-roughness patterns easily by merely choosing empirical coefficients properly for each case. Preliminary comparisons between analysis and test data were carried out on tests for circumferentially-grooved pump seals[6]. The agreement between test results and analysis for the circumferential-

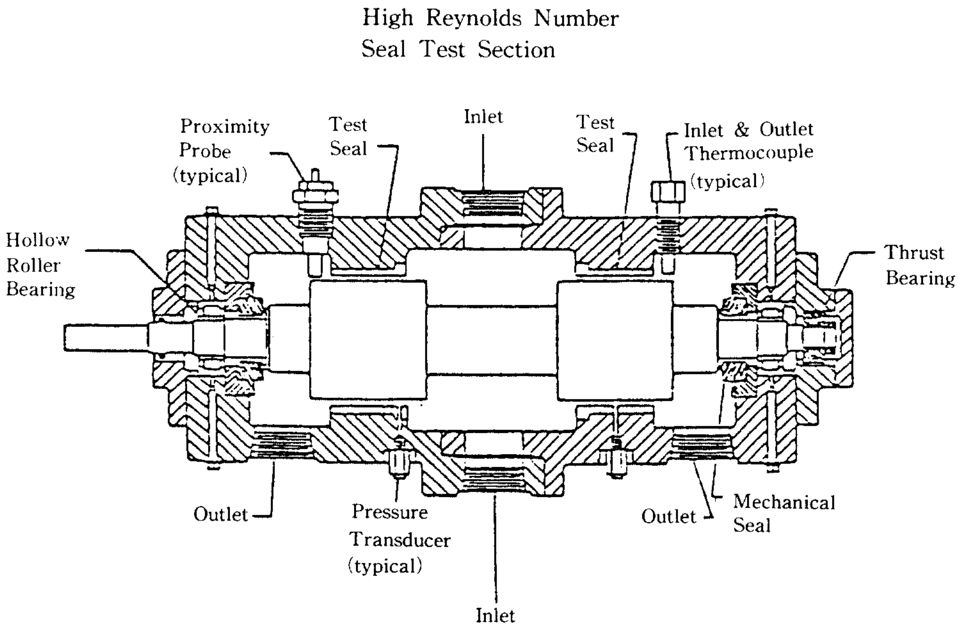

Fig.1. Test Section.

ly-grooved seals was satisfactory for the net-damping coefficients, but the stiffness predictions were mixed. Test results for circumferentially-grooved seals showed superiority in leakage performance, but inferiority in net-damping coefficients as compared to the smooth and damper seal. Grooved seals had lower stiffness than smooth seals.

Based on these preliminary results, a series of tests was newly conducted in which the groove-depth is systematically increased out to approximately three times the minimum clearance. Dynamic coefficients and leakage coefficients, which are based on measured radial and tangential forces and leakage data, are measured and compared to analysis. Measured data for the rotordynamic and leakage performance of these circumferentially-grooved seals are compared to smooth and damper seals in terms of the effective stiffness, damping, and leakage coefficients. More importantly, the effect of groove-depths on the rotordynamic and leakage performance of these circumferentially-grooved seals is investigated.

2. EXPERIMENTAL RESULTS

2-1. Introduction

The test results reported here were carried out in a test apparatus and facility which was developed as part of a high-Reynolds-number test program of pump seal configurations in support of the SSME(Space Shuttle Main Engine) development program. High Reynolds numbers, which are comparable to those achieved in the cryogenic turbopumps of the SSME, are achieved by using $CBrF_3$ as a test fluid. This is a Dupont-manufactured refrigerant and fire extinguisher fluid(Halon) which combines a high density and low kinematic viscosity, actually less than liquid hydrogen. Fig.1 illustrates the test apparatus. The test fluid enters the center and discharges axially across the two test seals. The rotor segments of

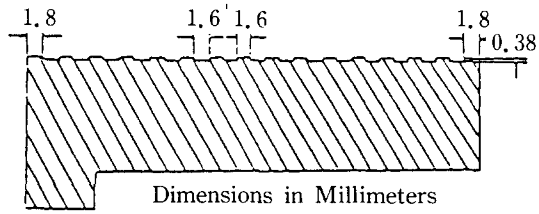

Fig.2. Circumferential-grooving details of stators.

the seal are mounted essentrically on the rotor with an eccentricity A. Hence, rotor rotation generates a synchronously-precessing pressure field. Axially-spaced, strain-gauge type pressure transducers are provided to measure the transient pressure field. Capacitance-type proximity probes are provided to simultaneously measure the rotor motion X(t), Y(t) relative to the stator. The transient pressure measurements are integrated to define Fr/A and F_θ/A, the force coefficients parallel and normal to the seal eccentricity vector. In any test, five to ten cycles of data, containing on the order of 2,000 data points, are analyzed, yielding a calculated average and standard deviation for Fr/A and F_θ/A. The test results used here were provided to answer the following questions:

(a) How do test results for grooved seals compare to the theory which was developed in reference[3]?

(b) How do the test results for grooved seals compare to each other, to the smooth seal and to the damper seal configuration with respect to rotordynamic coefficients and leakage?

(c) How do groove depths influence the rotordynamic and leakage performance of circumferentially-grooved seals?

2-2. Seal Configuration

All stators tested have a smooth rotor with a radius Ṙ=50 mm and a constant minimum clearance(0.5271 mm(20 mil)). All stators have 15 circumferential-groovings, as shown in Fig.2. Tests were carried out on four seal configurations which have 15 mil(0.381 mm), 25 mil(0.635 mm), 35 mil(0.889 mm) and 45 mil(1.143 mm) groove depths, respectively. By increasing groove depths, it is intended to investigate how groove depths influence rotordynamic leakage performance in circumferentially grooved seals.

2-3. Empirical Turbulence Coefficients

Following the iteration procedure discussed in[6], the rough-direction empirical turbulence coefficients, m_{sr}, n_{sr}, are obtained. These coefficients must be calculated from the static test data before a theoretical prediction can be made for Fr/A and F_θ/A. The results for empirical turbulert coefficients are provided in Table 1.

2-4. Dynamic Test Data

For a given seal configuration, a test matrix is carried out with variations in flow rate and shaft rotational speed. In a given test series, the axial Reynolds number is held constant and the rotational speed is incremented from approximetely 1000 rpm to 7200 rpm.

Fig.3 through 6 illustrate measured and theoretical results for Fr/A and F_θ/A versus Rao and ω for constant-minimum clearance, circumferentially-grooved stators.

2-5. Comparison to Theory

A quantitative comparison between theory and experiment is obtained by the following procedure. For small motion about a central

Table 1. Empirical turbulent coefficients for circumferentially-grooved stators.

Case	m_{sr}	n_{sr}
15 mil Groove Depth	−0.0268	0.1747
25 〃	−0.0890	0.3227
35 〃	−0.0810	0.4394
45 〃	−0.0782	0.5688

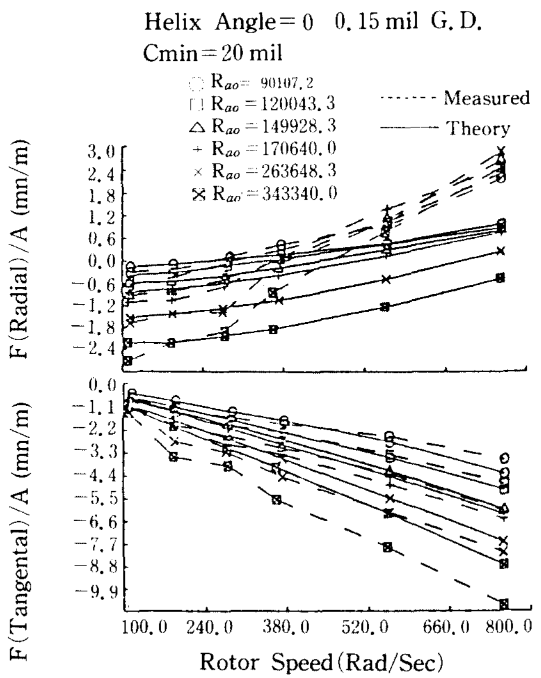

Fig.3. Measured and theoretical results for Fr/A and F_θ/A: 15 mil groove depth.

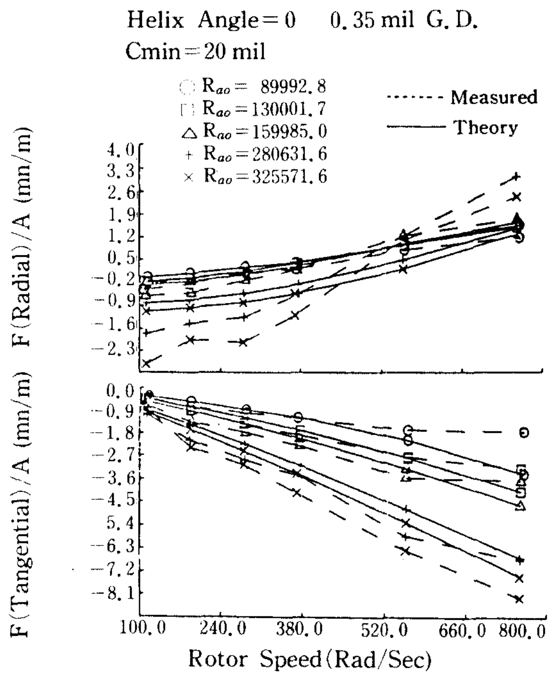

Fig.5. Measured and theoretical results for Fr/A and F_θ/A: 35 mil groove depth.

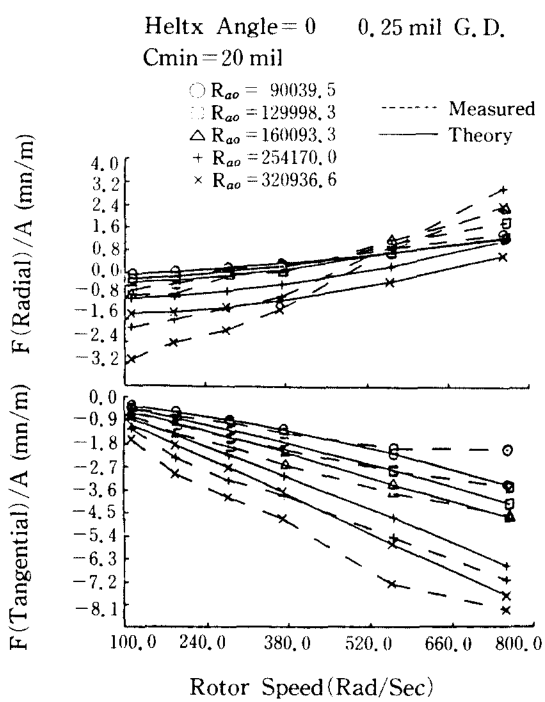

Fig.4. Measured and theoretical results for Fr/A and F_θ/A: 25 mil groove depth.

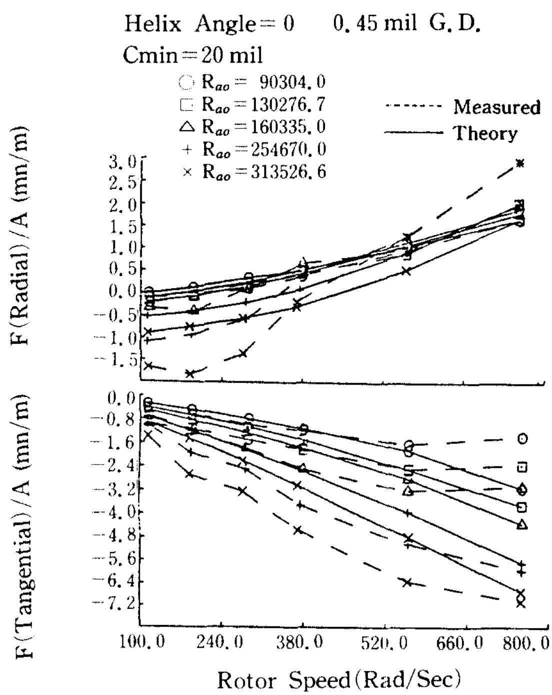

Fig.6. Measured and theoretical results for Fr/A and F_θ/A: 45 mil groove depth.

position, the force-motion model for an annular seal can be expressed as

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{pmatrix} K & k \\ -k & K \end{pmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} + \begin{pmatrix} C & c \\ -c & C \end{pmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} + M \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \quad (1)$$

where X(t), Y(t) define the relative position of the rotor relative to its stator, and $F_x(t)$, $F_y(t)$ are components of the reaction force. If a circular orbit of the form

$$X = A\cos\omega t,\quad Y = A\sin\omega t \quad (2)$$

is assumed, Eq.(1) yields the following definition of force coefficients which are, respectively, parallel and perpendicular to the rotating displacement vector.

$$F_\gamma / A = -K - c\omega + M\omega^2$$
$$F_\theta / A = k - C\omega \quad (3)$$

Eq. (3) provides the basis for a quantitative comparison between theory and experimental results. At first glance, these equations suggest that sufficient independent equations could be obtained to calculate all the rotordynamic coefficients by simply testing at three running speeds. However, the fact that the coefficients depend on ω precludes this approach. While K, C, and M are weak functions of ω through their dependence on ω, the cross-coupled coefficients k and c are linear functions of ω. In fact, if the fluid is prerotated prior to entering the seal such that the inlet tangential velocity is $U_{\theta 0} = R_\omega/2$, then theory predicts that $K = C_\omega/2$, $c = M_\omega$, and

$$F_\gamma / A = -K,\quad F_\theta / A = -C\omega / 2 \quad (4)$$

Table 2. Measured and theoretical predictions for K_{ef}, C_{ef}, and M_{ef}

Case	Ra_0	K_{ef} (N/m)		C_{ef} (NSec/m)		M_{ef} (kg)	
		Experiment	Exp./Theory	Experiment	Exp./Theory	Experiment	Exp./Theory
Circumf.-	90, 110	410,900	1.893	4, 203	0.723	3.748	2.430
Grooved	120, 000	837,700	1.479	5, 092	0.861	3.323	4.510
	149, 900	1,113,600	1.504	7, 405	0.906	5.319	3.640
(15mil G. D)	170, 600	1,731,000	1.822	7, 986	0.944	2.383	1.250
	263, 600	2,172,000	1.312	10, 350	1.021	5.064	2.200
	343, 300	3,471,000	1.494	13, 740	1.186	2.576	0.854
	90, 040	443,400	2.681	2, 597	0.548	1.167	0.901
	130, 000	863,200	2.206	4, 427	0.788	1.813	1.610
(25mil G. D)	160, 100	1,132,000	2.141	5, 976	0.935	2.847	1.790
	254, 200	2,089,000	2.180	8, 845	0.997	7.750	2.300
	320, 900	3,855,000	2.583	10, 510	1.005	2.256	0.664
	89, 990	402,100	2.833	2, 029	0.428	0.619	0.358
	130, 000	734,800	2.181	4, 209	0.752	0.416	0.407
(35mil G.D)	160, 000	1,124,000	2.655	4, 808	0.772	0.286	0.172
	280, 600	2,098,000	2.770	8, 974	1.029	6.802	1.940
	325, 600	3,351,000	3.523	11, 180	1.159	3.682	0.991
	90, 300	255,100	2.138	1, 717	0.388	2.395	0.965
	130, 300	257,100	0.766	2, 713	0.518	3.393	2.990
(45mil G.D)	160, 300	744,700	1.783	3, 758	0.630	1.266	0.670
	254, 800	1,544,000	2.518	7, 812	1.009	3.976	0.939
	313, 500	2,442,000	2.562	8, 913	0.979	4.552	1.050

The present test apparatus provides no intentioinal prerotation, and the expected result is of the form

$$k = b_1 C\omega/2, \quad b_1 < 1 \tag{5}$$
$$c = b_2 M\omega, \quad b_2 < 1$$
$$F_\gamma / A \cong -K_{ef} + M_{ef}\omega^2 = -K + M(1-b_2)\omega^2$$
$$F_\theta / A \cong -C_{ef}\omega = -C(1-b_1/2)\omega$$

The term C_{ef} denotes the "net damping coefficient" resulting from the drag force $C_\omega A$ and the forward whirl excitation force kA. A direct comparison between theory and experiment is obtained by curvefitting the theoretical and experimental results for F_r/A and F_θ/A to obtain predictions for K_{ef}, C_{ef}, and M_{ef}. Note that the procedure of curvefitting the data with respect to ω eliminates the running speed dependency. Further, K_{ef} is the zero-running-speed intercept of the F_r/A versus ω curve, and C_{ef} is the slope of the F_θ/A versus ω curve.

Comparisons among the curve-fitted results are provided in table 2.

A review of all stators tested supports the following conclusions:

(a) Theoretical direct stiffness values are generally smaller than measured.

(b) Theoretical predictions of C_{ef} are generally in good agreement with test results. The agreement generally improves with increasing Rao.

(c) Predictions for M_{ef} are mixed.

These results are generally consistent with previous predictions[6].

2-6. Performance Comparisons of Different Groove-Depth Stators

Four stators with different groove-depths are to be evaluated based on K_{ef}, C_{ef}, and leakage performance. The K_{ef}, C_{ef} parameters of the preceding sections are used as a measure of the direct stiffness and net damping. The nondimensional leakage coefficient C_L is defined by

$$\Delta P = Cd\frac{\rho\bar{V}^2}{2}$$
$$Q = 2\pi R\bar{C}\bar{V} = \frac{\bar{C}}{R} C_d^{-0.5} 2\pi R^2 \sqrt{\frac{2\Delta P}{\rho}} \tag{6}$$
$$= C_L \cdot 2\pi R^2 \sqrt{\frac{2\Delta P}{\rho}}$$

where Q is the volumetric flow rate

Tested stators are compared to the smooth and damper seal in terms of K_{ef}, C_{ef}, C_L in Fig. 7. Especially, the performance comparisons due to variations of groove depth are investigated.

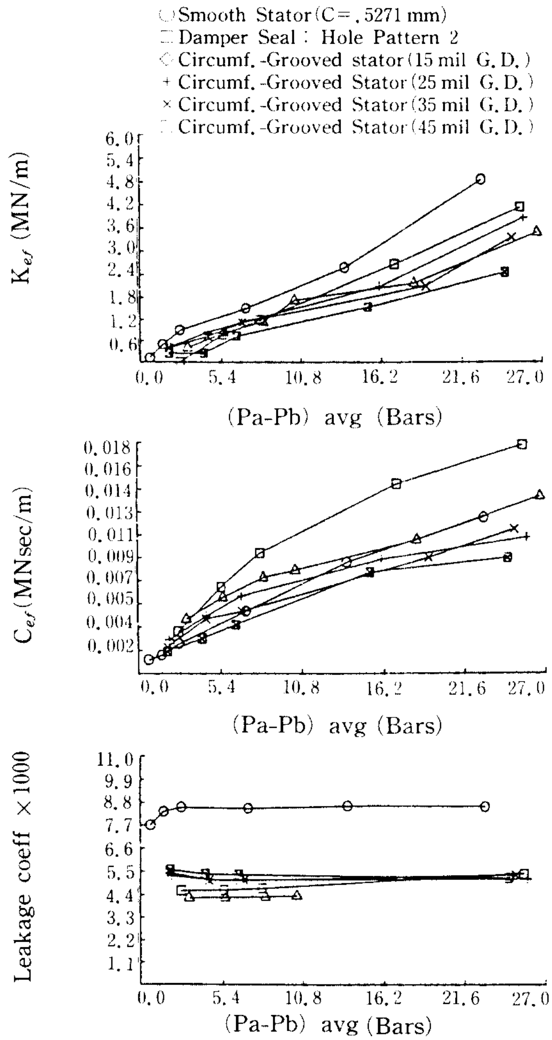

Fig.7. K_{ef}, C_{ef}, C_L vs ΔP for all stators tested and comparison with smooth and damper seals.

All seals have the same minimum clearance and radius. The damper seal employs a smooth rotor and deliberately-roughened stator elements to enhance the stability of the various surface roughness patterns which were tested in reference[1]. The hole pattern roughness maximized net-damping. Subsequent testing of hole pattern seals with variations of hole depth, and the ratio of hole-relieved area to total surface area demonstrated that 34% area ratio(γ) with a hole depth 3 times the minimum clearance has the best performance[2]. The results of the optimum-configuration hole-pattern damper seal are used for the following comparisons.

As shown in Fig.7, the average clearance has a dominant factor in K_{ef}. The smooth stator which has a minimum average clearance(0.5271 mm) has a maximum K_{ef}. Other seals show the significant decrease in K_{ef} as the average clearance increases by increasing in hole area or in groove depth. From a rotordynamics stability view point, C_{ef} should be maximized to improve the total system stability. In this view points, the damper seal is remarkably superior. The increased flow resistance in the tangential direction by the damper seal has reduced the destabilizing effect and increased C_{ef}. Since the smooth and circumferential-grooving patterns create no additional resistance in the tangential-direction, they have considerably lower values for C_{ef}. Generally, in all stators tested, C_{ef} is decreased by increasing the groove depth and average clearance. This result coincides with previous test results[6] for grooved seals. Note that leakage is directly proportional to C_L; hence, a minimum C_L is generally to be desired, and the circumferentially-grooved seal with 15 mil groove-depth is the best and the smooth stator the worst of the seals tested. As the groove depth is increased, the leakage is slowly increased. Note that the damper seal has very competitive leakage performance compared to grooved seals. Based on the results tested, a optimum groove depth for circumferentially-grooved seals exists and simple increase in the groove depth slightly deteriorate stability and leakage performance. Coupled with the previous test in reference[6], it can be concluded that the leakage can be effectively controlled by groove profiles rather than by groove depths.

3. CLOSURE

Additional new tests for circumferentially-grooved pump seals with groove-depth variations supports the following conclusions:

(a) The theoretical predictions of the net damping coefficient are generally in good agreement with test results, but the stiffness predictions are smaller than measured. All these result are generally consistent with previous predictions.

(b) Consistent with previous test results, circumferentially-grooved seals show superiority in leakage performance, but inferiority in net-damping coefficients as compared to the smooth and damper seal. Grooved seals have lower stiffness than smooth seals.

(c) An optimum groove depth for circumferentially-grooved seals exists and simple increase in the groove depth slighly deteriorate stability and leakage performance. The leakage can be effectively controlled by groove profiles rather than by groove depths.

Nomenclature

$\bar{C}$: average seal radial clearance. (L)

C, c : direct and cross-coupled damping coefficients defined by Eq.(1). (FT/L)

K, k : direct and cross-coupled stiffness coefficients defined by Eq.(1). (F/L)

M : inertia coefficient defined by Eq.(1). (M)

$R_{ao} = \frac{2\rho\bar{V}\bar{C}}{\mu}$: centered-position, axial Reynolds number

$\bar{V}$: centered-position, average axial fluid velocity.(L/T)

X, Y : radial seal displacement.(L)

ω : shaft angular velocity.(T^{-1})

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