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AN EXPERIMENTAL STUDY ON THE STATIC AND DYNAMIC CHARACTERISTICS OF PUMP ANNULAR SEALS WITH TWO PHASE FLOW

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Abstract

A new test apparatus is reconstructed and is applied to investigate static and dynamic characteristics of annular seals leaked by two phase flow (gas and liquid) for turbopumps. The fluid forces acting on the seals are measured for various parameters such as void ratio, the preswirl velocity, the pressure difference between the inlet and outlet of the seal, the whirling amplitude, and the ratio of whirling speed to spinning speed of the rotor. Influence of these parameters on the static and dynamic characteristics is investigated from the experimental results. As a results, with regard to the two phase flow, as the void ratio increases, the flow induced force decreases. Another dynamic characteristics of two phase flow is as almost similer as that of the mono phase flow.

1. Introduction

A turbomachines such as a pump tends to be operated at high speed and high pressure in special sircumstances because of severe performance requirements. In the special sircumstances, the fluid force causes an instability of the rotating machine because of the noncontacting seal. So, it is necessary to make the dynamic behavior of two phase flow clear and to supply accurate data in order to predict and prevent the unstable vibration, and still more necessary to design a stable rotating machine.

Many theoretical and experimental analyses on the characteristics of noncontacting seals are reported for the monophase flow seal. As an experimental research on annular seals, Childs et al. (Refs. 2,3) investigated fluid forces acting on the seals and their characteristics with a test apparatus which has an eccentric rotor. Nordmann (Ref. 4) used an impulse force type test apparatus to carry out an experiment on the characteristics of annular seals. Kaneko et al. (Ref. 7), Kanki et al. (Ref. 8), and Hori et al. (Ref. 9) also reported their test results on the seals. All these results are important for the research on monophase flow annular seals.

The authors also investigated for the many types of seals, that is, annular seal, parallel groove seal, spiral seal and tri angle hale seal and also long annular seal for investigating the effect of moment force. The dynamic characteristics of annular seals leaked by the two phase flow have not been investigated yet in various operating conditions.

This paper shows the experimental results obtained by an experimental apparatus which is newly redesigned to obtain a dynamic characteristics of two phase flow. In the test the fluid force acting on the seals is measured then radial and tangential forces of the seal flow and the stiffness, damping, and inertia coefficients are obtained. Influence of the parameters such as whirling amplitude, pressure difference between the inlet and outlet of the seal, preswirl velocity, rotating and whirling speeds, and their directions is investigated. Then the effect of two phase flow on the dynamic characteristics of annular parallel seal is investigated.

2. Nomenclature

F_x, F_y	: Fluid forces in x and y directions
F_r, F_t	: Fluid forces in r and t directions
$K_{xx}, K_{yy}, K_{xy}, K_{yx}$	: Stiffness coefficients
$C_{xx}, C_{yy}, C_{xy}, C_{yx}$	: Damping coefficients
$M_{xx}, M_{yy}, M_{xy}, M_{yx}$	: Inertia coefficient
x, y, z	: Fixed coordinates (z: axial direction)
r, t	: Radial and tangential coordinates
$p(\phi, z)$	: Pressure distribution
$P(\phi)$	: Average pressure in axial direction
P_{in}	: Inlet pressure to the seal
P_{ex}	: Outlet pressure from the seal
ΔP	: Pressure difference between inlet and outlet of the seal
ω	: Spinning angular velocity of the rotor
Ω	: Whirling angular velocity of the rotor
e	: Whirling eccentricity of the rotor
ϕ	: Phase difference between principle force and displacement
R, D	: Rotor radius and diameter, respectively
L	: Seal length
C	: Seal clearance
V_t	: Preswirl velocity
V_{ts}	: Preswirl velocity without a rotating motion
V_a	: Fluid average velocity in axial direction
$\overline{\alpha}$	: mean void ratio
Q_g, Q_l	: flow rate of gas and liquid, respectively
$\overline{P}, P_{atm}$	: average pressure in the seal and outlet pressure

3. Experimental apparatus

3.1 Test seal apparatus

Figs.1 and 2 show assembly of the test apparatus and the layout of the test facility, respectively. In Fig.1, a working fluid, that is, water and gas, is injected through three pairs of swirl passages to accomplish the different inlet swirl velocities shown in cross section B. The water passes through the clearances of the seal and flows to outlets in both sides of the housing. Cool water and gas is continuously supplied to the tanks in order to maintain constant temperature. The inlet part consists of four tubes connected with swirl passages for injecting the water and gas, and swirl speed is adjusted by a valve for air and two valves for water in order to obtain the arbitrary swirl velocity in the range of 0 - 15.0m/s. These valves are used in order to change the void ratio.

The seal assembly consists of a seal stator and a seal rotor. The seal rotor, of which diameter is 70mm, is connected to the motor by a flexible coupling. The motor is controlled in the speed range 0 - 3500r.p.m. by an electric inverter, which also selects the rotational direction. The seal stator has holes to measure the dynamic pressure in the seal, as shown in cross section C. Both long and short seals have three holes in axial direction and four holes in x and y directions. The dynamic pressure is measured by the strain gauge type pressure gauge shown in cross section C. The fluid dynamic force acting on the stator is also directly measured by the load cells shown in cross section C for comparison with the data of the pressure transducers.

The bearing assembly has two ball bearings to make spinning and whirling motions. To make a whirling motion, an inside sleeve and an outside sleeve, which have a 0.05mm eccentricity to each other, are attached between the two bearings as shown in cross section A of Fig.1. The two eccentric sleeves can be rotated, relatively. So an arbitrary eccentricity can be adjusted in the range of 0 - 0.1mm. The sleeves of both sides are driven by a motor through the timing belts. The motor can also be controlled by an electric inverter in the rotating speed range of 0 - 3500r.p.m., and rotational direction can be selected by an inverter.

The two phase flow is made as shown in Fig.3, where gas is mixed by higher pressure a liquid. The inlet pressure of two phase flow is measured just before the seal.

The void ratio in two phase flow can be changed the pressures of water and gas by of valves and regulator.

3.2 Measuring instruments and methods

The measuring procedure illustrated in Fig.4 consists of four kinds of physical variables: that is, rotating and whirling speed, dynamic pressure in the seal, seal forces and displacement of the rotor. Signals from measuring instruments are recorded by a data recorder and analyzed by a computer.

The rotating and whirling speeds are measured by eddycurrent type pulse sensors and digital counters.

Dynamic fluid force can be directly measured by a load cell as shown in Fig.5. The force measured by the load cell is calibrated to revise the influence of the O - rings and the inertia of the seal.

An eddycurrent type displacement sensor is used to measure the displacement and this displacement is used as a reference signal to obtain the phase difference between the displacement and the flow induced force.

The test data are recorded by a data recorder and sent to the computer through an A/D converter. In the computer the pressure values are integrated in the circumferential direction to obtain the fluid force, then the characteristic coefficients are calculated.

A special pitot tube set is used to measure the preswirl velocity in the seal inlet, as shown in Fig.6. The static and total pressures are measured by the pressure transducers instead of the usual U - tube.

Axial flow velocity in seal is determined from the outlet flow velocity directly measured by a pitot tube.

The flux of water and gas is obtaind by measuring the leakage of unit period by cylinder.

The void ratio in the two phase flow is decided as average value between inlet and outlet of seal. Here, it is supposed that air is ideal gas and slip ratio between water and gas is 1 because the seal is set horizontally . At this time, we obtain flowing equation from the flux of water and gas and the pressure which averages one of inlet and outlet.

$$\overline{\alpha} = \frac{P_{atm} Q_g}{\overline{P}\left(Q_l + \frac{P_{atm}}{\overline{P}} Q_g\right)} \tag{1}$$

Seal dimensions that is used for the test and experimental conditions are shown in the table 1,2.

3.3 Calibration and measuring error

As a preliminary test, the static and the dynamic characteristics of measuring instruments, i.e. the pressure transducer, the load cell and the pitot tube, are calibrated.

Dynamic calibration of the load cell was done in various working conditions. A periodic force was excited on the seal by a shaker, and the force was simultaneously measured by strain gauges attached to the shaking rod and by the load cell fixed on the other side. These data were used to calibrate the measured data in the real test.

A pitot tube set was used to measure the inlet preswirl velocity. Its pitot tube coefficient for calibration was determined by means of a standard pitot tube.

4. Calculation of fluid force and characteristic coefficients

It is assumed that the motion equation of the rotor system is represented as follows

$$[m]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{f(t)\} - \{F\}$$

where [M], [C], [K] are the mass, the damping, and the stiffness matrix and { F(t) } is the force vecter. { F } = $\{ F_x \ F_y\}^t$ is the fluid reaction force acting on a rotor and is represented as a linear function of rotor displacement, velocity and acceleration, as follows.

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} \tag{2}$$

For a small whirling motion about a center position, the relation of the coefficients may be expressed by

$$M_{xx} = M_{yy}, \quad M_{yx} = M_{xy} = 0, \quad C_{xx} = C_{yy},$$
$$C_{yx} = -C_{xy}, \quad K_{xx} = K_{yy}, \quad K_{yx} = -K_{xy} \tag{3}$$

where the cross-coupled inertia coefficient is neglected because it is negligible small.

A rotating coordinate system rotating with the rotor is adopted to analyze the fluid force easily. The relation between a fixed coordinate system and a rotating coordinate system is illustrated in Fig.7, where F_x and F_y are represented by the radial force F_r and the tangential force F_t. Details to obtain the fluid force is shown in (Ref.11), so these are not written here.

5. Experimental results and discussions

5.1 Visualization

We visualize a flow patteern of two phase flow in the part of seal made of acryle. As a result, we can observe that bubble of air in two phase flow is fine and two phase flow is homogeneous bubbly flow because of high rotating speed of the rotor.

5.2 Relative Uncertainly

Since the discrepancy of the fluid forces measured by the pressure transducer and load cell are very few, for the void ratio $\overline{\alpha}$=0 as shown in Fig.8, it is decided that the accuracy of the test data is very good. So in this test the fluid force measured by the load cell is used as the experimental results in this discussion.

Fig.9 show the time history of fluid force measured by the load cell for mean void ratio $\overline{\alpha}$=0 and 0.45. From these data it is known that the two phase flow has inherent random noise.

In the measuring process as the void ratio was increased, random vibration due to the two phase flow became very large. So the variance of the measured vibration data become large, but these are not the measurement error.

5.3 Effect of inlet pressure

Fig.10 shows the effect of inlet pressure on the radial and tangential forces. From these data it is known that the flow induced force increases with increasing the inlet pressure and its tendency is not change with increasing the void ratio. From this result it is known that the experiment can be done by constant inlet pressure to investigate the effect of void ratio on the flow induced force.

5.4 Effect of spinning speed

Figs.11(a) to (d) show the effect of spinning speed on F_r/e and F_t/e for different void ratio. Fig.(a) shows for void ratio zero in order to compare the result with another void ratio. Figs.(b) to (d) show F_r/e and F_t/e for the void ratio $\overline{\alpha}$=0.25, 0.45 and 0.7 in the case of various spinning speeds. From these results it is known that the fluid force decreases with the increase of void ratio. The variation of radial force increases with increase of void ratio. These phenomena are seemed to be occurred by the compressibility of the gas. For the tendency of the tangential and radial forces for rotating speed is same, and as rotating speed is increased, the fluid force increases.

Fig.12 shows the equivalent spring coefficients and damping coefficients and added mass. From these figures it is known that as the void ratio is increased, the spring coefficients and damping coefficients decrease as discussed in the result of F_r/e and F_t/e. Fig.13 shows the whirl frequency ratio,

$$f = K_{xy}/C_{xx}\Omega \tag{4}$$

and fluid force F/e, which is important to evaluate the instability force ;

$$F/e = \frac{1}{e}\sqrt{F_r^2 + F_t^2} \tag{5}$$

From this figure it is known that the whirl frequency ratio decreases at $\overline{\alpha}$=0.25 and then increases for increase the void ratio. The whirl frequency ratio has large difference at $\overline{\alpha}$=0.7, because the variation of force due to two phase flow becomes large and the error of amplitude and phase may be large. In this void ratio region the fluid force F/e becomes small, then it seemed to be small effect on the rotor system. Conventionaly, the pump rotor is supported by bearings and seals in the operational condition, and the seal force act as a stabilizing force in the rotor system. In order to demonstrate the effect of two phase flow, we assume that a rotor system is stable and in this system the bearings act to unstable and the seals act to stable. In this case, if the seal force is decreased by the two phase flow as shown in Fig.13, this stabilizing seal force decreases and the rotor system may become unstable.

5.5 Effect of preswirl velocity

Figs.14(a) to (c) shows the effect of the whirl ratio ω/Ω on F_r/e and F_t/e for various preswirl velocities. In this case preswirl velocity ratio is 1.6 for ω=2500rpm. The results illustrate that the effect of preswirl on the fluid force is same tendency as the case of void ratio $\overline{\alpha}$=0 but it decreases as the void ratio increases.

Fig.15 shows the whirl frequency ratio versus preswirl velocity. It is known from this figure that the effect of preswirl velocity decreases as the void ratio increases. Therefore two phase flow of high void ratio does not affect on instability of rotor system.

6. Conclusions

The two phase flow induced force through annular seal are experimentally investigated for setting the parameter void ratio, rotating speed, whirl/spin ratio, preswirl velocity, inlet pressure. Then the following results are obtained,

(1) The void ratio is increased, random vibration due to the two phase flow becomes very large.

(2) The fluid force increases with increasing the inlet pressure and its tendency is not change with increasing the void ratio.

(3) The fluid force decreases with the increase of void ratio. These phenomena is occurred by the compressibility of the gas.

(4) Whirl frequency ratio which is proposed by D. Childs decreases a little at $\overline{\alpha}$=0.25, but in practical, it is not so much changed and as the fluid force decreases in high void ratio, instability force decreases.

(5) The effect of preswirl velocity decreases as the void ratio increases, so this effect dose not act as instability.

(6) Since the seal force is decreased by the two phase flow, this stabilizing seal force decrease, and rotor system may become unstable.

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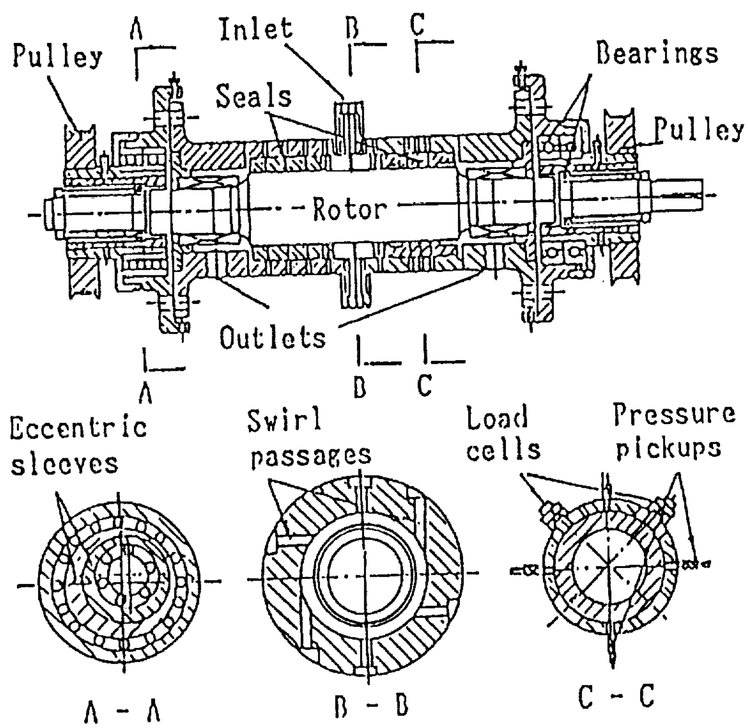


Fig.1 Test apparatus assembly

Table 1 Seal dimension

	Length (mm)	Diameter (mm)	L/D	Clearance (mm)
Plain Seal	70	70	1	0.5

Table 2 Experimental condition

Rotating speed	ω(rpm)	500 ~ 3500
Whirling speed	Ω(rpm)	±600 ~ ±2400
Whirling amplituide	e(μm)	50
Preswirl velocity without a rotating motion	Vts	-15 ~ 15
Temperature of water	T(°C)	19

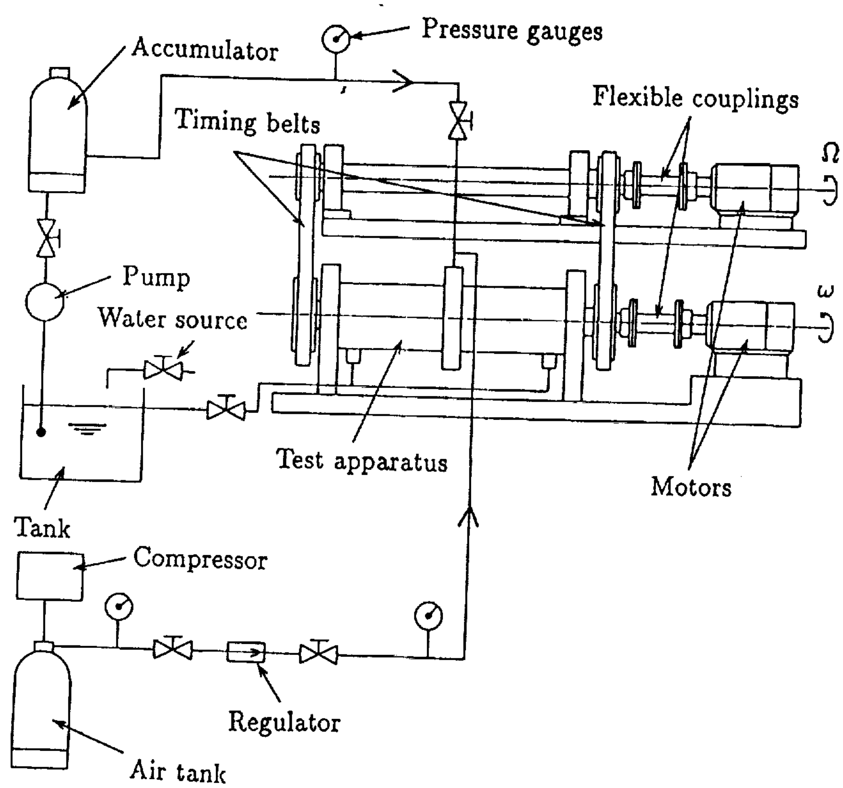


Fig.2 Test facility layout

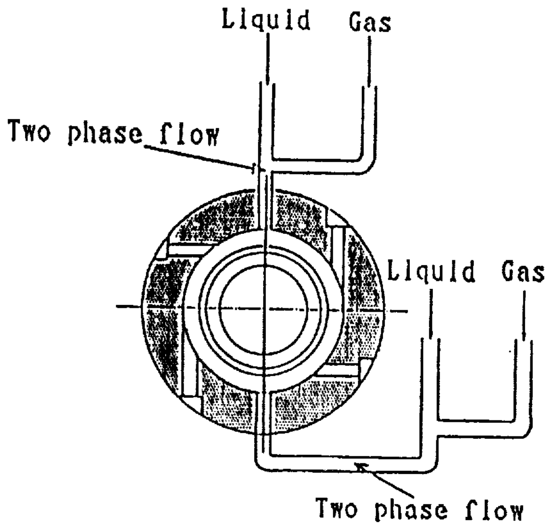


Fig.3 Detail of gas-liquid mixing device

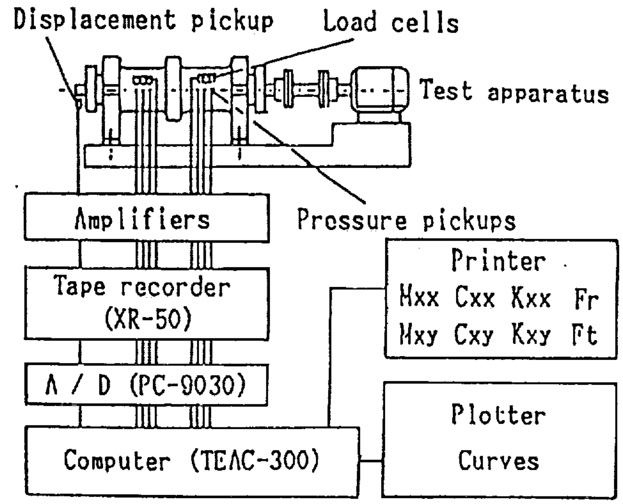


Fig.4 Analysis procedure

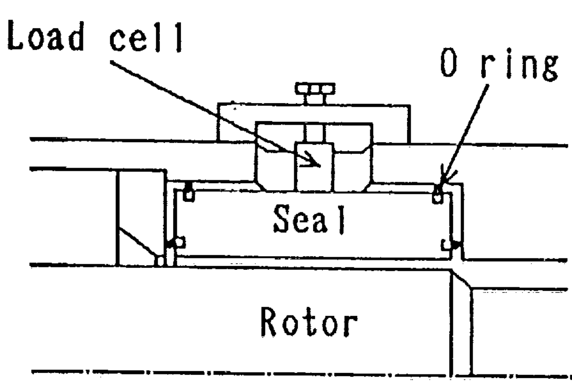


Fig.5 Detail of force measurement

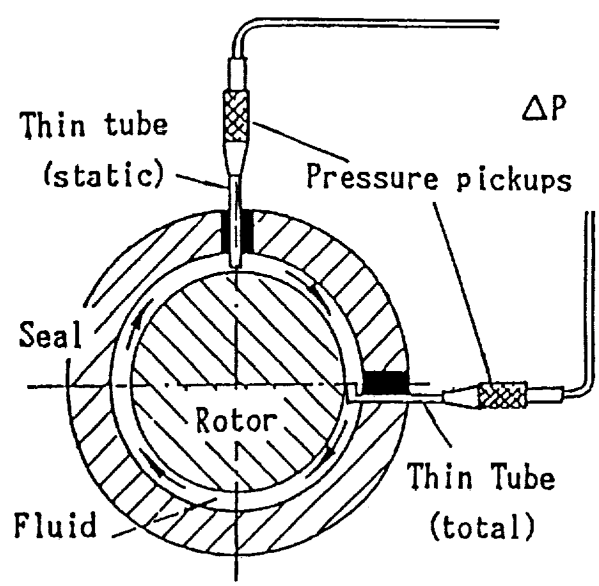


Fig.6 Detail of swirl velocity measurement

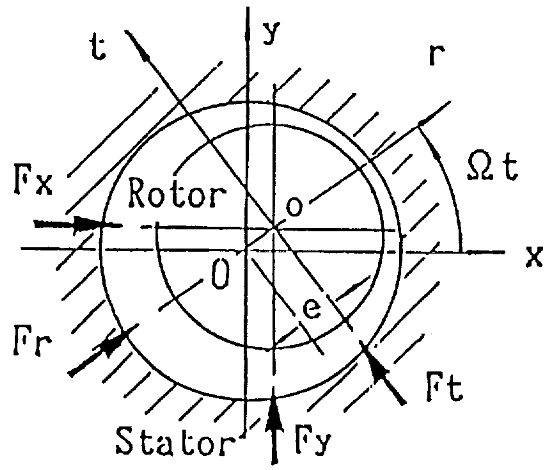
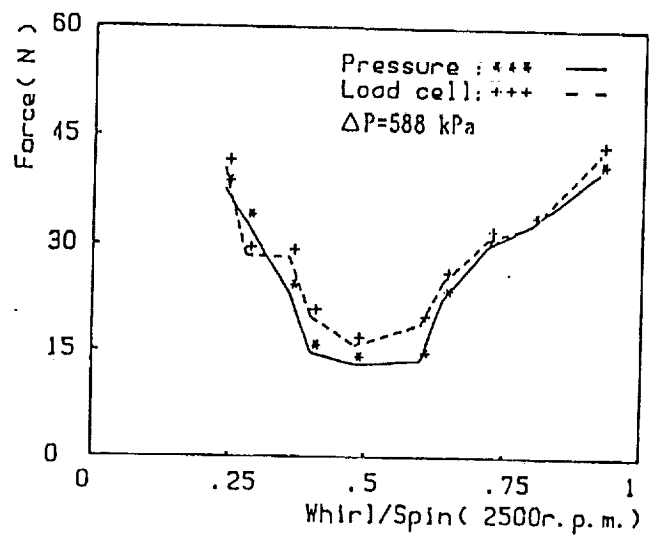


Fig.7 Model of annular seal



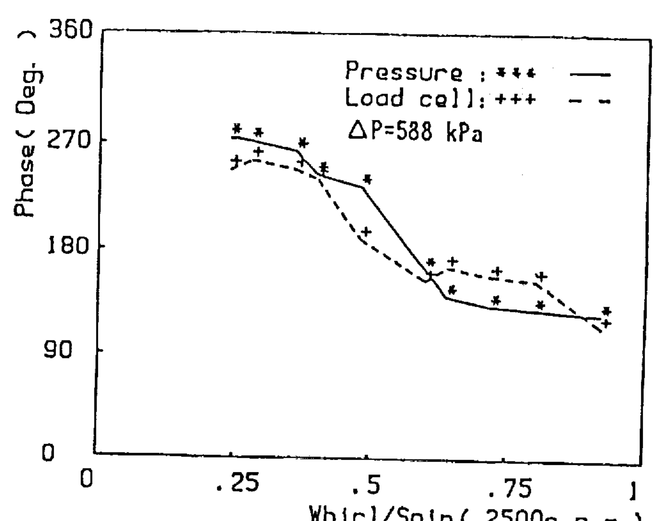


Fig.8 Results measured by two measuring methods

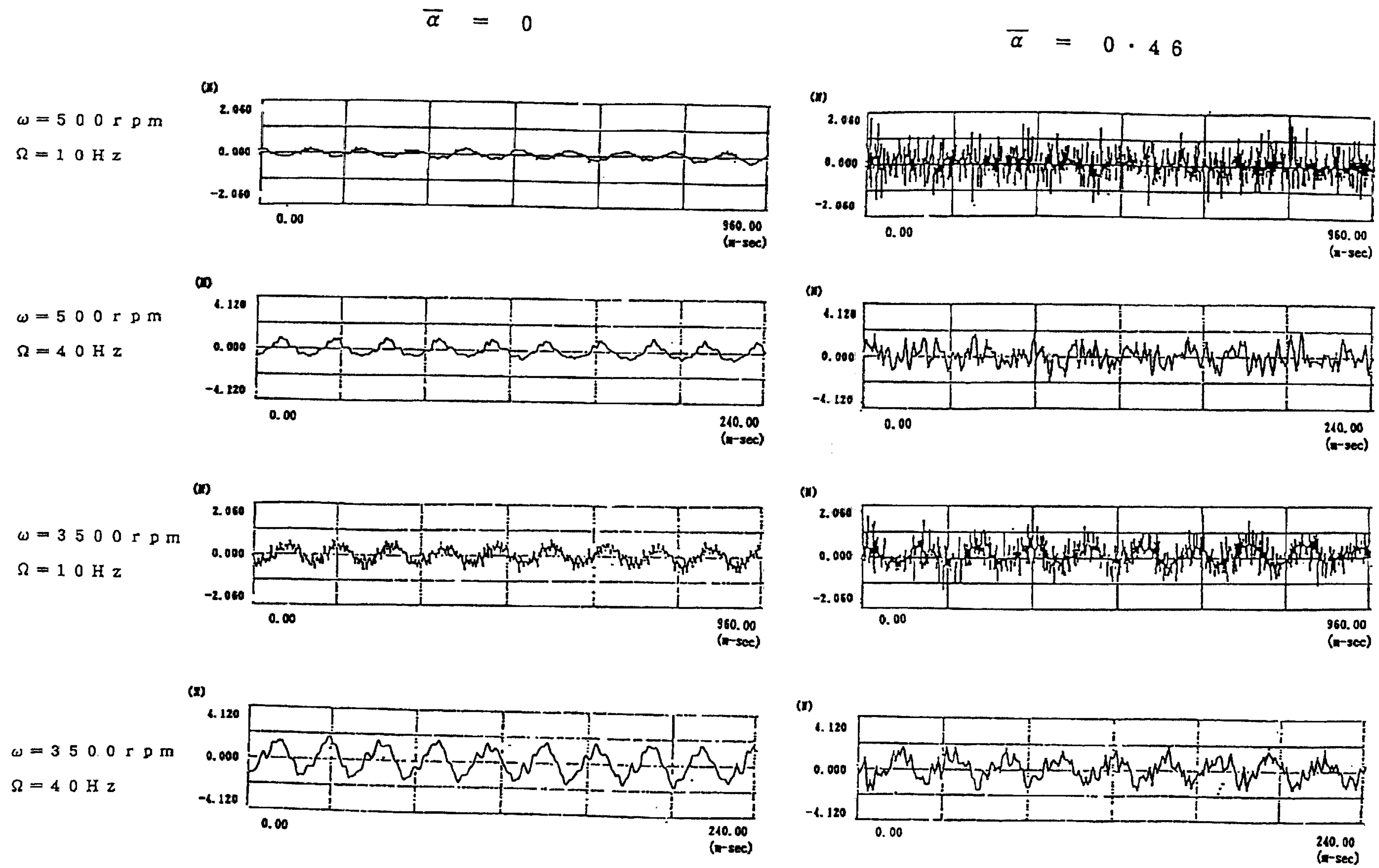
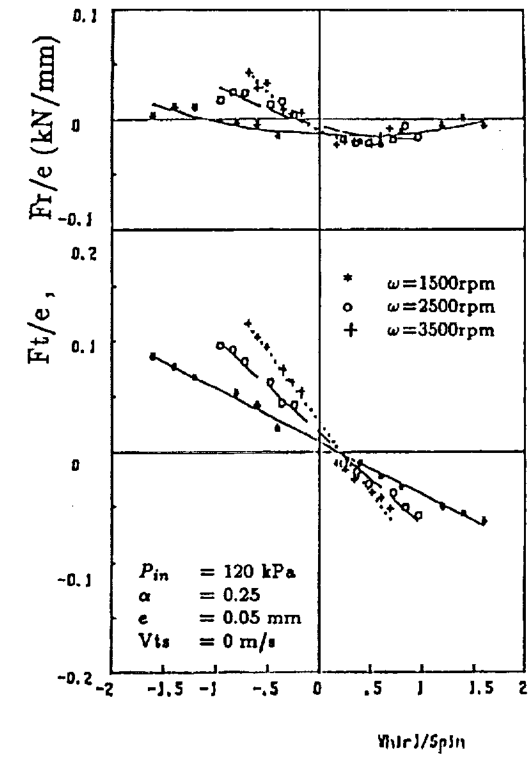
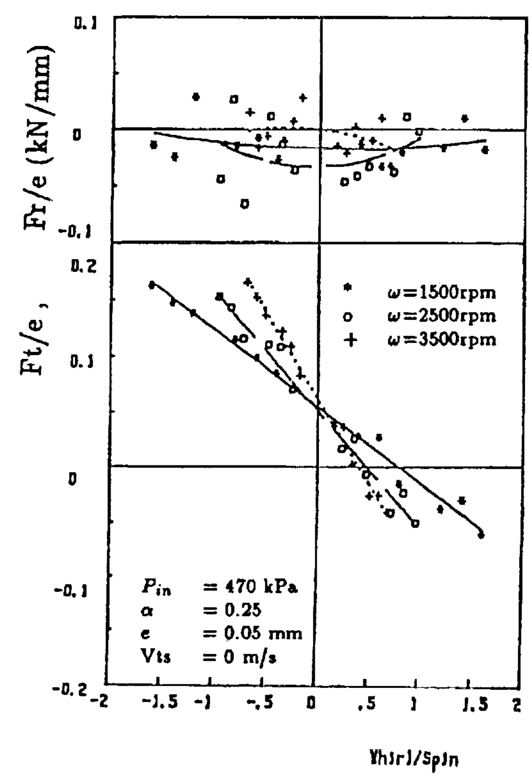


Fig.9 Time history of fluid force

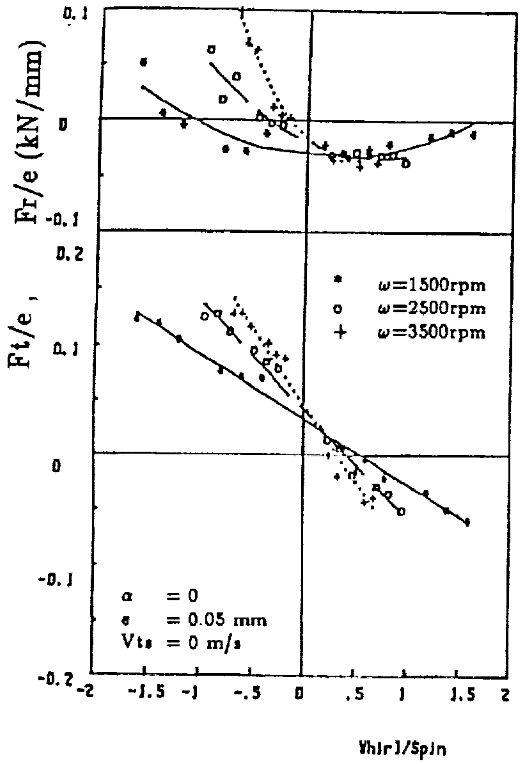


(a) $P_{in} = 120\mathrm{k}P_a$

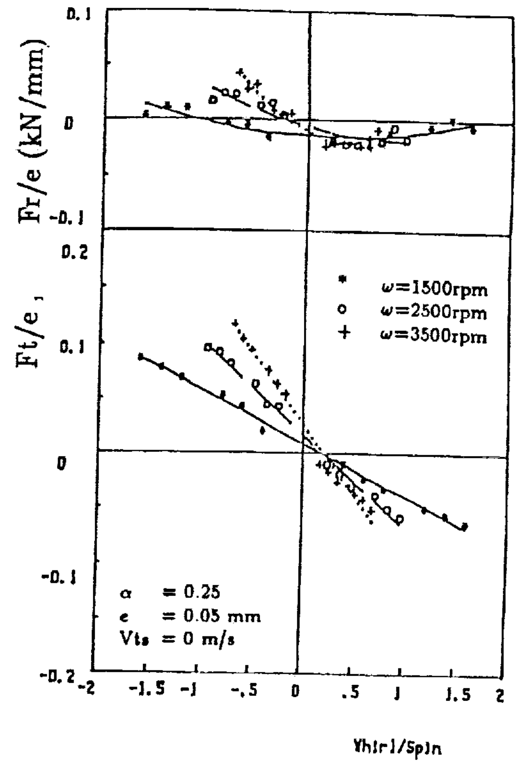


(b) $P_{in} = 470\mathrm{k}P_a$

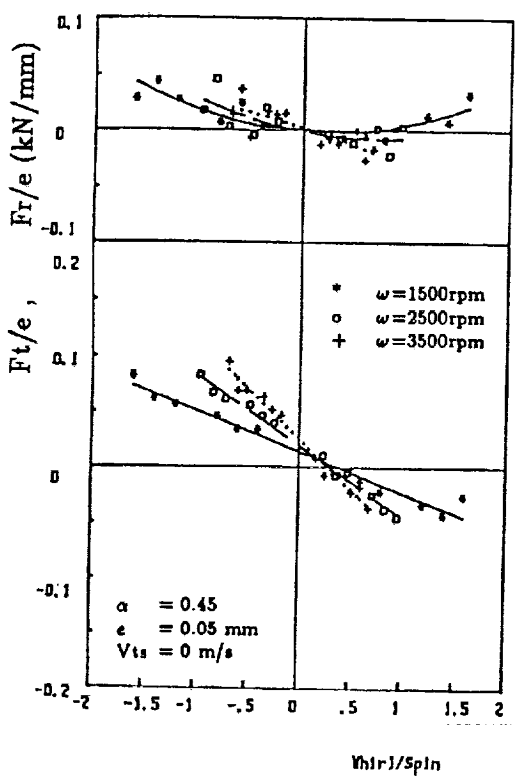
Fig.10 Radial and tangential forces for the constant void ratio



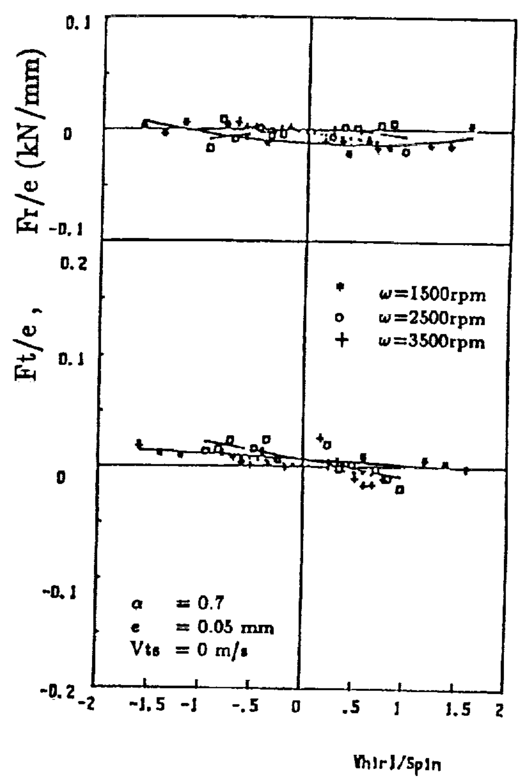
(a) $\bar{\alpha}$=0.0



(b) $\bar{\alpha}$=0.25



(c) $\overline{\alpha}$ = 0.45



(d) $\bar{\alpha}$=0.7

Fig.11 Radial and tangential fluid forces for void ratios

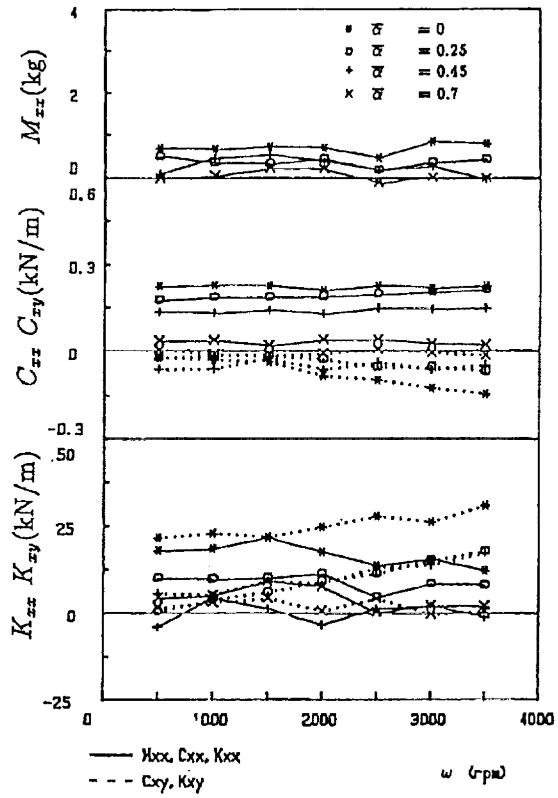


Fig.12 Fluid force cofficients for void ratios

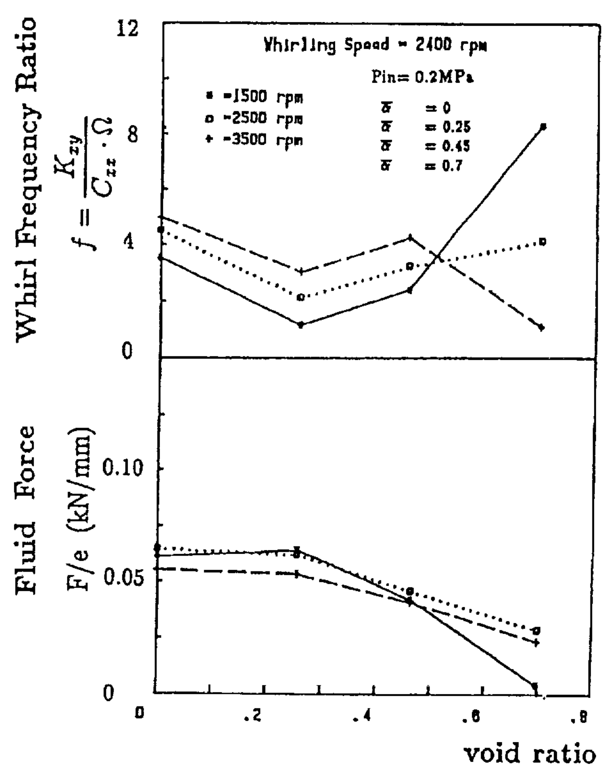
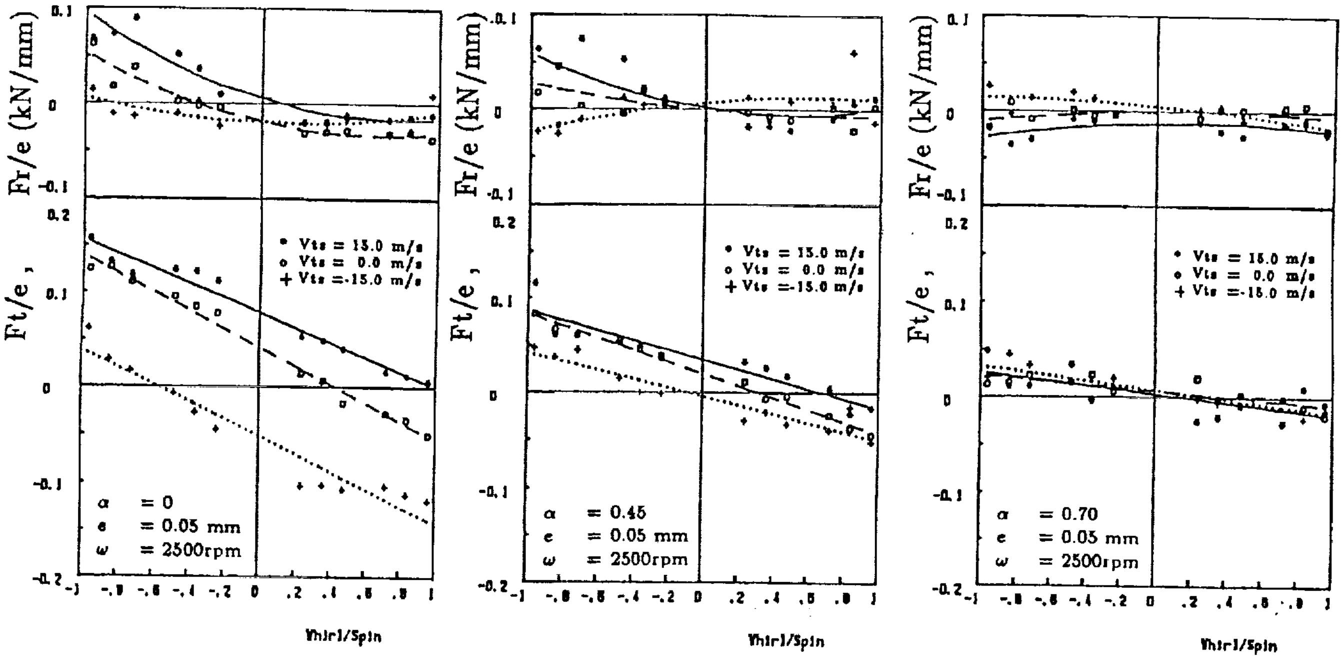


Fig.13 Whril frequency ratio and fluid force versus void ratio



(a) $\overline{\alpha} = 0$ (b) $\overline{\alpha} = 0.45$ (c) $\overline{\alpha} = 0.7$

Fig.14 Radial and tangential forces for preswirl velocity

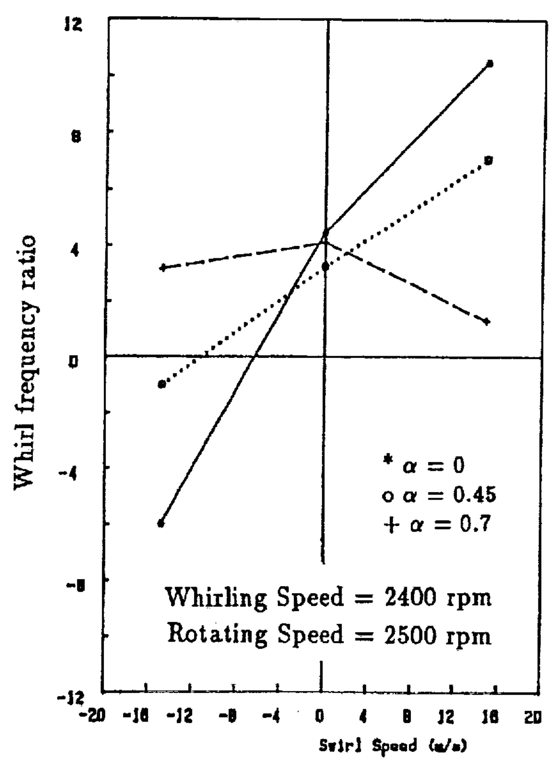


Fig.15 Whirl frequency ratio for void ratios