N94- 34180

# TEST RESULTS FOR ROTORDYNAMIC COEFFICIENTS OF ANTI-SWIRL SELF-INJECTION SEALS

C.H. Kim and Y.B. Lee
Korea Institute of Science and Technology
Seoul, Korea

54-37
12848
p-9

## ABSTRACT

Test results are presented for rotordynamic coefficients and leakage for three annular seals which use anti-swirl self-injection concept to yield significant improvement in whirl frequency ratios as compared to smooth and damper seals. A new anti-swirl self-injection mechanism is achieved by deliberately machining self-injection holes inside the seal stator to partially divert inlet flow into the anti-swirl direction. The anti-swirl self-injection mechanism is used to achieve effective reduction of the tangential flow which is considered as a prime cause of rotor instability in high performance turbomachinery. Test results show that the self-injection mechanism significantly improves whirl frequency ratios; however, the leakage performance degrades due to the introduction of the self-injection mechanism. Through a series of the test program, an optimum anti-swirl self-injection seal which uses a labyrinth stator surface with anti-axial flow injections is selected to obtain a significant improvement in the whirl frequency ratio as compared to a damper seal, while showing moderate leakage performance.

Best whirl frequency ratio is achieved by an anti-swirl self-injection seal of 12 holes anti-swirl and 6 degree anti-leakage injection with a labyrinth surface configuration. When compared to a damper seal, the optimum configuration outperforms the whirl frequency ratio by a factor of 2.

## NOMENCLATURE

| | |
|---|---|
| $\underline{A}$ | Measured frequency response functions, introduced in Eq. (2) |
| C, c | Direct and cross-coupled damping coefficients, introduced in Eq. (1), (FT/L) |
| $C_d$ | Leakage coefficient, introduced in Eq. (5) |
| $\bar{C}$ | Radial clearance,(L) |
| $\underline{E}'$ | Identity matrix, introduced in Eq.(2) |
| $F_X, F_Y$ | Seal reaction-force components, introduced in Eq. (1), (F) |
| K,k | Direct and cross-coupled stiffness coefficients, introduced in Eq. (1) (F/L) |
| M | Added-mass coefficient, introduced in Eq.(1), (M) |
| $\underline{S}'$ | Error matrix, introduced in Eq.(2) |
| X,Y | Seal rotor to stator relative displacement, introduced in Eq.(1), (L) |
| Ra | Seal axial Reynolds number($2V\bar{C}/\upsilon$) |
| V | Fluid average axial velocity(L/T) |
| $\underline{X}$ | Unknown coefficients, $\underline{M}$, $\underline{C}$, and $\underline{K}$, introduced in Eq. (2) |
| f | whirl frequency ratio, introduced in Eq. ( 4 ) |
| $\omega$ | rotorational speed, introduced in Eq. ( 4 ) |
| $\upsilon$ | Fluid kinematic viscosity (L /T) |
| $\Omega$ | Impulse hammer frequency |

## INTRODUCTION

Throughout 1980's , one of the major concerns in seal dynamics is the consequence of a reduction in the tangential velocity inside the seal. Lower tangential velocities yield a reduction in the cross-coupled stiffness coefficient k of the following reaction-force/displacement model for the seal.

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

This linearized model is assumed to apply for small motion about a centered position and defines the reaction-force components ( $F_x$,$F_y$) in terms of (X,Y), the components of the displacement vector of the seal rotor relative to its stator. Reducing k reduces the destabilizing forces developed by the seal due to hydrodynamic action and should yield a higher stable operating speed for pumps.

von Pragenau(1982) originally proposed that annular seals for pumps which used rough stators and smooth rotors would lower tangential velocities than conventional seals which use smooth rotors and stators. A reduction of the tangential velocity was realized through damper seals(Childs and Kim, 1985,1986; Iwatsubo and Sheng,1990) which used rough stators and smooth rotors. The surface-roughness pattern retards the generation of the tangential flow inside the seal clearance. While damper seals are adopted in several industrial applications into high performance turbomachinery, a promising alternative in reducing the tangential velocity is introduced and tested, named as " swirl brake". The swirl brake which is used in the inlet section of the seal clearance brakes the inlet swirl inherent inside the seal clearance. The related research effort has been reported with a success in the SSME HPOTP turbine interstage seals.(Childs et.al,1990)

In this paper, a 'new' concept in reducing the tangential velocity is proposed and preliminarily tested with a series of the seal. Newly proposed is the anti-swirl self-injection seal which is illustrated in Fig. 1 along with smooth and damper seals for comparisons. Using the inlet high pressure, the inlet flow is partially diverted into holes or slots inside the seal stators and self-injected into the reverse direction of the shaft rotation. This concept can be a more aggressive measure to a reduction of the tangential flow inside the seal clearance. The resultant tangential velocity is presumably reduced, and the consequent improvement in stability can be obtained. This anti-swirl self-injection concept is shown to be a promising alternative to the conventional swirl brake. The present test program examines the process of the concept realization and refinement of the newly proposed seals. The consequence of the anti-swirl self-injection mechanism is tested and the side effect on leakage performance is investigated.

## TEST APPARATUS

A new test rig was designed and built, based on the basic design reported in the reference of Massmann and Nordmann(1985). The principle of the test rig is simple but complicated enough to satisfy the current purpose of the test. Fig. 2 illustrates the test rig. The movable housing is flexibly supported by springs and steel bars to the main structure of the test rig. The two identical seal inserts can be exchanged for new test seals. After the seal is inserted into the main housing, the pressure measurement holes are machined and connected to the Scani-valves.With a four stage supply pump and controlled flow rates, the supplied fluid enters the seal section in the middle through two supply holes. The inlet fluid then exits through two seal sections. The magnetic flow meter measures flow rates in the exit pipe line. The pressurized seal generate the seal forces which are the source of the dynamic coefficients. The movement of the main body relative to the rotor was measured by four eddy-current-type proximity probes. As will be explained later, seal coefficients can be measured with dynamic impact tests for each test seal.

The temperatures, flow rates, and pressures are measured and controlled through a personal computer.The temperature of the working fluid, water, was kept constant at 40 °C, using an automatically-controlled cooler inside the water tank. A modification to the apparatus of Massmann and Normann(1985) is the measurement of the static pressure inside the seal clearance through the Scani-valve mechanism illustrated in Fig. 3. The inlet pressure, five pressure readings inside the seal clearance, and the outlet pressure are measured through the Scani-valve mechanism which is connected to a pressure sensor.

With this test rig, a test matrix was set up to vary rotation speeds up to 6000 rpm and supply pressures up to 10 bars. Five seal inserts were sequentially tested for a range of rotation speed and supply pressures.

## TEST SEALS

As stated in the preceding section, five seals were tested serially for concept demonstration and refinement of the newly proposed seals. For comparison to conventional seals, a smooth seal and a damper seal were also tested. Fig. 1 shows geometries of a smooth seal, a damper seal, and three anti-swirl self-injection seals. The geometry of the damper seal was adopted from the test results in the reference of Childs and Kim(1986) for a possible optimum damping. For the anti-swirl pattern, a 12 holes injection was considered. Twelve supply holes were drilled into the stator body and then the anti-swirl pattern was realized by drilling holes against the direction of rotation. For leakage reduction, a 6 degree anti-leakage injection and a labyrinth surface were considered. Table 1 shows the configuration of tested seals. All tested seals have the same minimum clearance of 0.2 mm with a smooth rotor. Details are explained in the table. Seal 3 was tested first to see the effect of anti-swirl configurations. The effect of anti-leakage injection was tested in seal 4 with 12 holes of anti-swirl injection. Based on the above series of tested seals, the seal 5 of 12 holes with anti-leakage injection and labyrinth effect was tested.

## SEAL PARAMETER IDENTIFICATION

The impulse hammer technique was used for the dynamic testing of the parameter identification in the frequency domain(Massmann and Normann,1985) as illustrated in Fig. 4. As explained in detail in the reference of Massmann and Normann(1985), the instrumental variable method is used. In conjunction with the least square method, the instrumental variable method utilizes the measured frequency response function to iterate the least square algorithm for better curve-fittings.

Utilizing the fact that the product of the mobility matrix and the stiffness matrix should be the identity matrix, a complex equation is arranged into an overdetermined equation system in the case of a broad band excitation of the impulse hammer.

$$\underline{A}\underline{X} = \underline{E}' + \underline{S}' \qquad (2)$$

where $\underline{A}$ consists of the measured frequency response functions related to the exciter frequencies $\Omega$ , $\underline{X}$ represents the unknown coefficients $\underline{M}$ , $\underline{C}$, and $\underline{K}$, $\underline{E}'$ is a modified identity matrix and $\underline{S}'$ is the error matrix. After deriving the loss function with respect to the seal coefficients, the instrumental variable method is applied as shown in Fig. 5. A new matrix $\underline{W}^T$ with instrumental variables is built up, using the initial matrix obtained by the original least square matrix method.

$$\underline{W}^T\underline{A}\underline{X} = \underline{W}^T\underline{E}' \qquad (3)$$

As shown in Fig. 5 , this procedure is repeated and after each step the actual estimation is compared with that of the last step. The procedure stops if the correlation is satisfactory. The instrument variable method is less sensitive to noise in the measurement data and reduces the error of the identified coefficients(Massmann and Normann,1985) as illustrated in Fig.6.

## EXPERIMENTAL RESULTS

### Dynamic Coefficient and Leakage Test Data

For a given seal configuration, a test matrix is obtained by varying the axial Reynolds number and running speed. The Ra range varies between the maximum flow capacity of the supply pump and minimum $\Delta P$ sufficient to generate reasonable transient pressure signal amplitudes. For a given Ra value, the running speed is varied sequentially over the running-speed capacity of the drive motor.

An evaluation of the relative performance of seal configurations is expedited by extracting stiffness, damping, and mass coefficients from the impact force/displacement data in frequency domain. From a stability standpoint, the destabilizing tangential force is of most interst. A positive cross-coupled stiffness k is destabilizing because it drives the forward orbital motion of the rotor. Positive direct damping C and a negative cross-coupled stiffness are stabilizing because they oppose the orbital motion. A convenient measure of seal stability is the whirl frequency ratio of cross-coupled stiffness to direct damping forces with a circular orbit.(Childs et.al,1989)

$$\text{whirl-frequency ratio} = f = \frac{k}{C\omega} \qquad (4)$$

The stator inserts are to be evaluated based on k, C, whirl-frequency ratio, and leakage performance. Volumetric seal leakage is defined by

$$\Delta P = C_d \frac{\rho V^2}{2} \qquad (5)$$

where the leakage coefficient $C_d$ is a nondimensional relative measure of the leakage to be expected from seals with different radii.

### Relative Uncertainty

The uncertainty in the dynamic coefficients can be determined using the method described by Holman(1978). The uncertainty in the force and displacements are 0.5N,0.0016mm, respectively. Before normalization, the nominal calculated uncertainty in stiffness coefficients is 51.53 N/mm(4.67%) and 0.164 N-s/mm(3.82%) for the damping coefficients. These predicted uncertainty values are generally satisfactory in comparison to nominal dynamic coefficients.

### Relative Performance of Anti-Swirl Self-Injection Seals

As stated, relative performance of anti-swirl self-injection seals was compared with a smooth seal and a damper seal in terms of leakage and stability. The leakage performance was measured by the leakage coefficient, while the stability was measured by the whirl frequency ratio. The influence of rotor speeds on the direct damping and the cross-coupled coefficients was first examined for an axial Reynolds number 8,000 in Fig. 7. First, as expected, the direct damping is independent to the rotor speeds, while the cross-coupled stiffness is a strong function of the rotor speeds;viz. the current test results follow general trends of other seals as predicted in the previous reports in references of Childs and Kim(1985,1986) and Iwatsubo and Sheng(1990). For the damping coefficient, the damper seal is outstanding ; nearly six times the smooth seal and trifold the anti-swirl self-injection seals. For the cross-coupled stiffness coefficient, seal 5 outperfoms other seals; ten times smaller than the smooth seal and twenty times smaller than the damper seal. The anti-swirl self-injection seals have much smaller values than conventional seals in the cross-coupled stiffness. These results assure the anti-swirl concept ; the anti-swirl self-injection mechanism retards the tangential flow inside the seal clearance and therefore results in the reduction of cross-coupled stiffnesses.

In Fig. 8, the whirl frequency ratio and leakage coefficients were compared among tested seals. As expected in the measurement of dynamic coefficients, anti-swirl self-injection seals are much better than the smooth seal. Among the anti-swirl self-injection seals, seal 5 is the best; more than six times less than the smooth seal and less than half the damper seal. As far as the leakage performance is concerned, the damper seal outperforms the others. During the test program, seal 3 was tested first and found to have too much leakage as shown in Fig. 8. Therefore, the anti-axial flow pattern was introduced in the seal 4 . Seal 4 improves about 10 % over the seal 3, while showing about the same performance in the whirl frequency ratio. Results show that the anti-axial flow injection is not as attractive as first expected for leakage reduction; maybe because of the limited reverse angle(6 degree) due to the limit of the current test rig installation, or the sharp acute reverse turn make the diverted flow lose too much of its momentum to have a discernible effect on the anti-leakage concept. Finally, in this test series, seal 5 with labyrith effect is tested. The labyrinth effect was introduced to improve the leakage performance; however, it was not enough, compared with the damper seal. More tests are planned for improving the leakage performance. However, more importantly in stability view point, these test results of the anti-swirl self-injection seal concepts clearly show that the anti-swirl concept is very effective in improving the stability .

In Fig. 9, the cross-coupled stiffness and damping coefficients are compared with different pressure differences. As the pressure difference increases, the direct damping increases; this result follows general trends of other seals. The damper seal has the highest damping values. Seal 3 and seal 4 have 30-40% higher damping values than the smooth seal. Seal 5 has about the same damping values as the smooth seal. For the cross-coupled stiffness, the anti-swirl self-injection seals show a clear pattern of reducing cross-coupled stiffnesses: versus the smooth and damper seals which show increased cross-coupled stiffness patterns as the pressure difference is increased. The anti-swirl self-injection seals show decreasing or constant cross-coupled stiffnesses. Again this figure clearly confirms the anti-swirl self-injection concept.

In Fig.10, the whirl frequency ratios and leakage coefficients are compared. As expected from Fig. 9, anti-swirl self-injection seals show clear superiority in stability performance over smooth and damper seals. Among the self-injection seals, seal 5 has the lowest whirl frequency ratio; less than half that of smooth and damper seals. However, the leakage performance can be a drawback. Seal 4 and seal 5 are better than the smooth seal; however, the damper seal leaks 20% less than seal 5. These results show that better designs should be addressed in the anti-swirl self-injection seals . Along with this leakage performance, the dependency of pressure differences and hole numbers for better stability performance should be studied in more details. Current results demonstrate how the anti-swirl self-injection seals perform and introduce the concept of anti-swirl and self-injection mechanisms for annular seals.

## CONCLUSIONS

Newly proposed anti-swirl self-injection seals have been tested and compared with smooth and damper seals. Test results show that the anti-swirl self-injection concept can significantly reduce cross-coupled stiffnesses and to show a stability improvement by a factor of 2 in the whirl frequency ratio as compared with a damper seal. A minor drawback identified in this preliminary test is the leakage performance. More detailed tests for the leakage performance based on new designs are planned and could solve this problem.

## REFERENCES

Childs, D.W., Baskahrone, E., Ramsey. C., 1990" Test Results for Rotordyanmic Coefficients of the SSME HPOTP Turbine Intersatge Seal with Two Swirl Brakes," NASA

CP 3122, Proceedings of the workshop : Rotordynamic Instability Problems in High Perforamance Turbomachinery held at Texas A&M University, May 21-23, pp165-178

Childs, D.W., Elrod, D. and Hale, K., 1989,"Annular Honeycomb Seals: Test Results for Leakage and Rotordynamic Coeficients; Comparisons to Labyrinth and Smooth Configurations," ASME Trans. Journal of Tribology, Vol. 111, pp293-301

Childs, D.W. and Kim,C.H., 1985,"Analysis and Testing for Rotordyanmic Coefficients of Turbulent Annular Seals with Different, Directionally Homogeneous Surface-Roughness Treatment for Rotor and Stator Elements", ASME Trans. Journal of Tribology, Vol.107, No.3, July , pp296-306.

Childs.D.W. and Kim, C.H., 1986, "Test Results for Round-Hole-Pattern Damper Seals : Optimum Configurations and Dimensions for Maximum Net Damping," ASME Trans. Journal of Tribology, Vol. 108, Oct., pp605-611

Holman,J.P.,1978, Experimental Methods for Engineers, McGraw Hill, pp45

Iwatsubo, T., and Sheng, B.C., 1990,"An Experimental Study on the Static and Dynamic Characteristics of Damper Seals," Proceedings of 3rd International Conference on Rotordynamics, Lyon, France, Sept.10-12,pp99-104

Massmann, H. and Nordmann R., 1985," Some New Results Concerning the Dynamic Behavior of Annular Trubulent Seals," NASA CP 2409, Proceedings of the workshop : Instability in Rotating Machinery held in Carson City, June 10-14, pp179-194

von Pragenau,G.L., 1982,"Damping Seals for Turbomachinery,"NASA Technical Paper 1987

Table 1. Configuration of Test Seals

| Test seals | Type | Configuration |
|---|---|---|
| seal 1 | plain | |
| seal 2 | damper | hole pattern(Childs and Kim,1986) |
| seal 3 | ASIS | 12 holes one-line injection |
| seal 4 | ASIS | 12 holes one-line with anti-axial flow injection |
| seal 5 | ASIS labyrinth | 12 holes one-line with anti-axial flow injection and labyrinth effect |

* ASIS : Anti-Swirl Self-Injection Seal

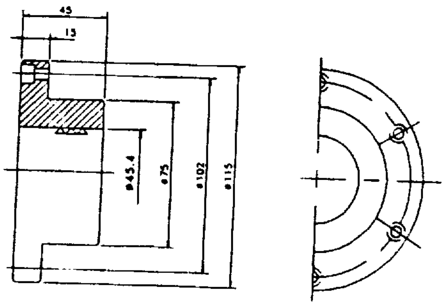



Fig.1(a) Smooth seal insert (seal 1)

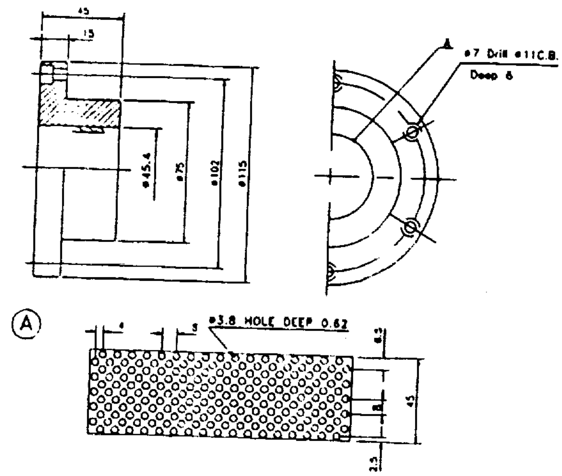



Fig.1(b) Damper seal insert (seal 2)

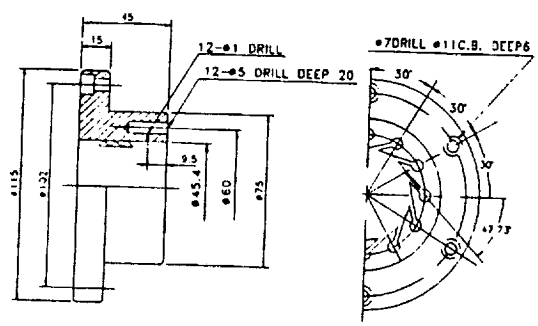



Fig.1(c) Anti-Swirl Self-Injection seal insert (seal 3)

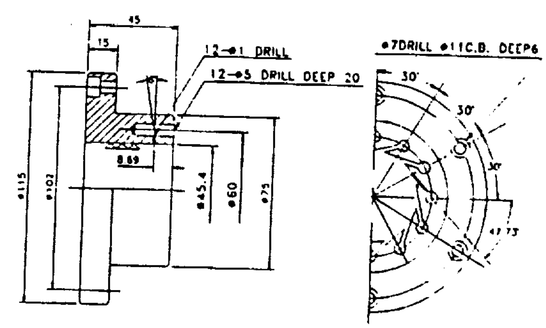



Fig.1(d) Anti-Swirl Self-Injection seal insert with anti-axial flow injection (seal 4)

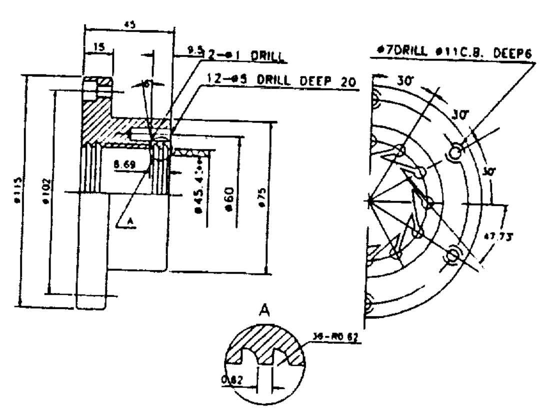



Fig.1(e) Anti-Swirl Self-Injection seal insert with anti-axial flow injection and labyrinth effect (seal 5)

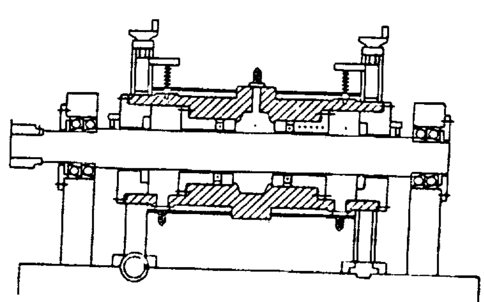


Fig.2 Test apparatus

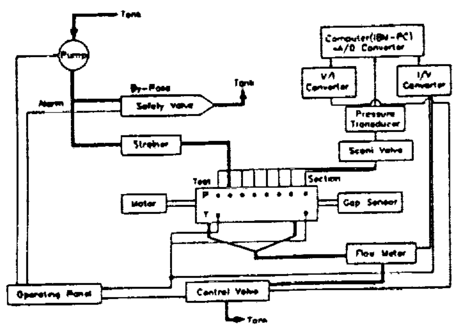


Fig.3 Schematics of the sealing test system

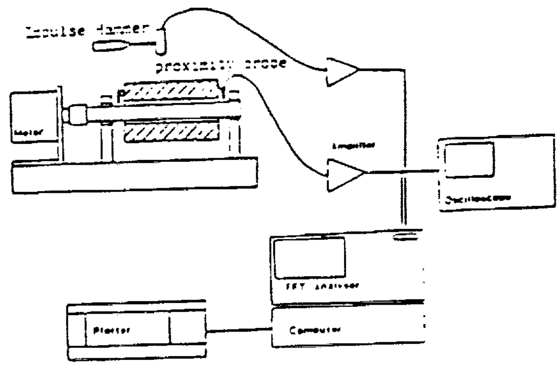


Fig.4 Data acquisition and processing

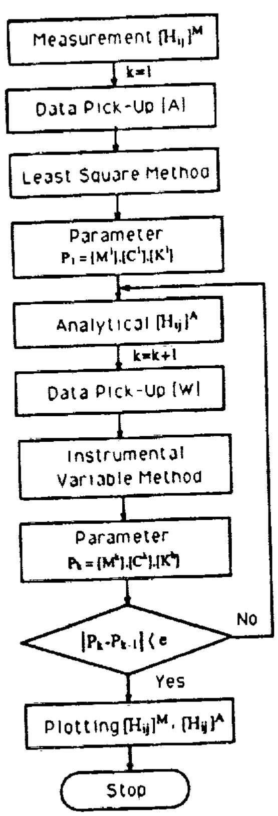



Fig.5 Flow chart of instrumental variable method

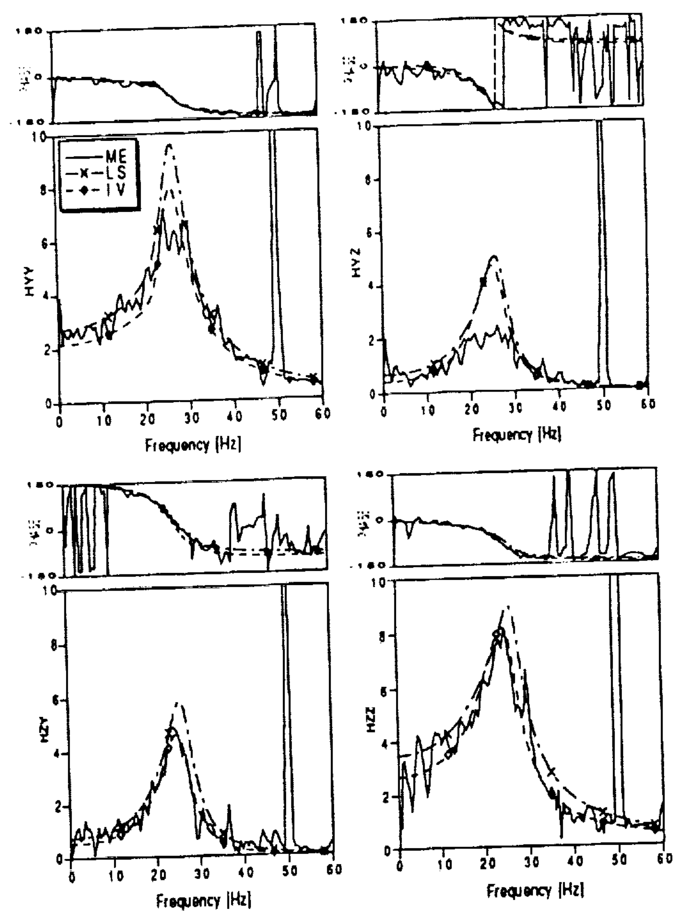
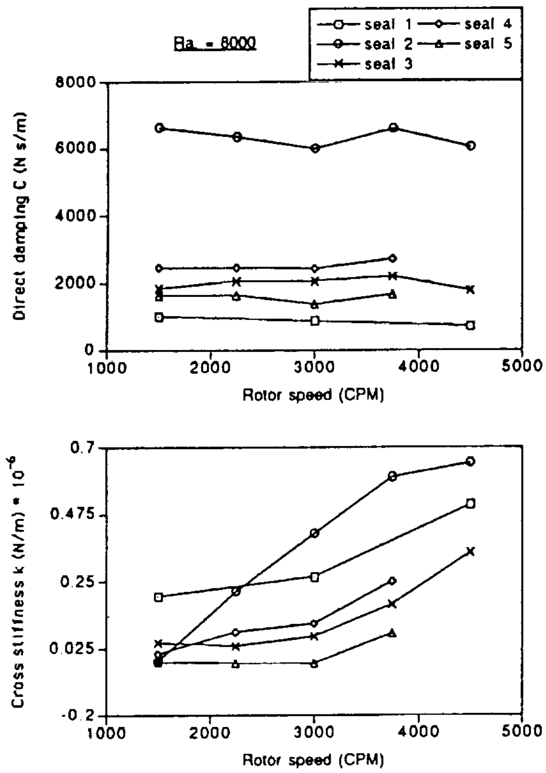



Fig.6 Example of measured and analytical frequency response functions ( ——— measured(ME), —X— least square(LS), - -◇- - instrumental variable(IV) )



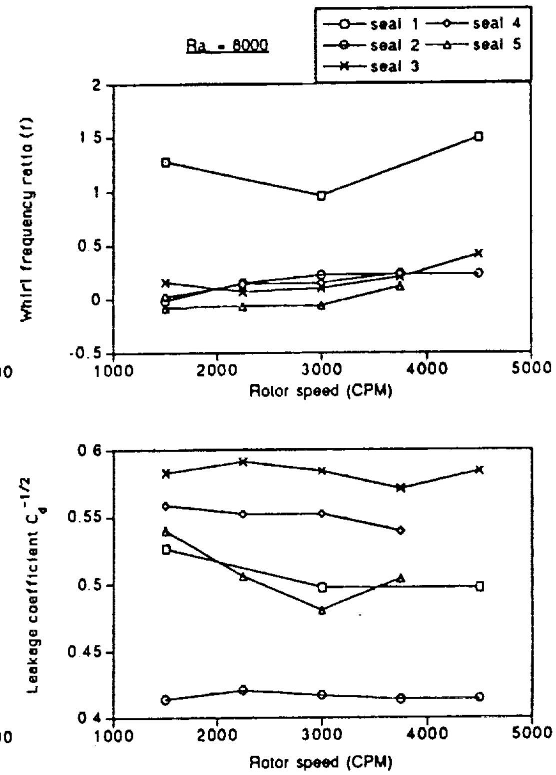
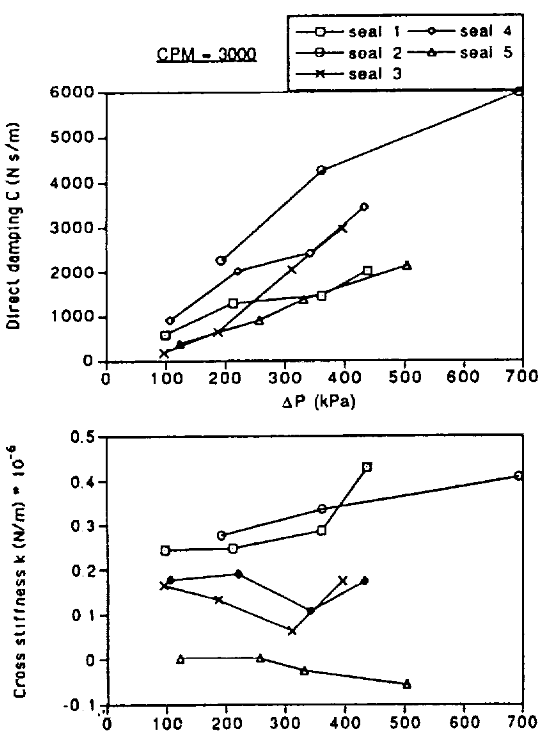




Fig.7 C and k versus CPM for the five seals of Table 1

Fig.8 f and $C_d$ versus CPM for the five seals of Table 1



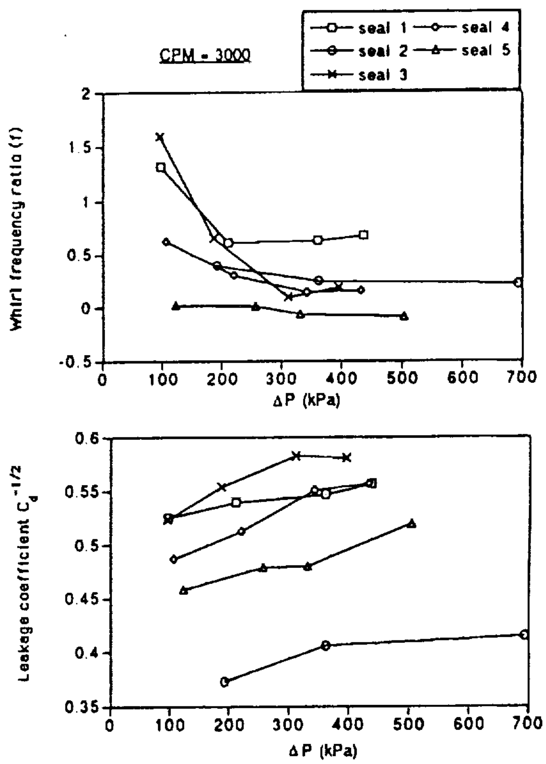




Fig.9 C and k versus $\Delta P$ for the five seals of Table 1

Fig.10 f and $C_d$ versus $\Delta P$ for the five seals of Table 1