The design and development of a high speed journal bearing test rig for measuring dynamic characteristics of oil film bearings

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Abstract

This paper outlines the design and development of a test rig used to determine the dynamic characteristics of oil film journal bearings. Some background is given on the main design, operating parameters, and instrumentation systems. Recent developments include the ability to test at higher speeds using a specially designed drive motor and new test and analysis techniques. Newly commissioned software and detailed design improvements have also been incorporated to enable the rig to be used on a continuing basis for a range of bearing types and sizes. Some typical output plots are given for a 5 pad pivoted shoe bearing. A tribute is given to the late Keith Brockwell, a respected colleague and fellow researcher in the field.

Key words: Testing, hydrodynamic, journal bearing
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>$A_H$</td>
<td>Bearing housing acceleration, horizontal component</td>
<td>m s$^{-2}$</td>
</tr>
<tr>
<td>$A_V$</td>
<td>Bearing housing acceleration, vertical component</td>
<td>m s$^{-2}$</td>
</tr>
<tr>
<td>$b_{xx}$</td>
<td>Direct velocity (damping) coefficient</td>
<td>Nsm$^{-1}$</td>
</tr>
<tr>
<td>$b_{xy}$</td>
<td>Indirect velocity (damping) coefficient</td>
<td>Nsm$^{-1}$</td>
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<tr>
<td>$b_{yx}$</td>
<td>Indirect velocity (damping) coefficient</td>
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<tr>
<td>$b_{yy}$</td>
<td>Direct velocity (damping) coefficient</td>
<td>Nsm$^{-1}$</td>
</tr>
<tr>
<td>$F_1$</td>
<td>Dynamic force, right hand vibrator</td>
<td>N</td>
</tr>
<tr>
<td>$F_2$</td>
<td>Dynamic force, left hand vibrator</td>
<td>N</td>
</tr>
<tr>
<td>$H$</td>
<td>Static force, horizontal component</td>
<td>N</td>
</tr>
<tr>
<td>$p,q$</td>
<td>Local axes, aligned with orbit crossover portions</td>
<td>m,m</td>
</tr>
<tr>
<td>$R_1$</td>
<td>Displacement transducer channel characteristic</td>
<td>V m$^{-1}$</td>
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<tr>
<td>$R_2$</td>
<td>Displacement transducer channel characteristic</td>
<td>V m$^{-1}$</td>
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<tr>
<td>$R_3$</td>
<td>Displacement transducer channel characteristic</td>
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<tr>
<td>$R_4$</td>
<td>Displacement transducer channel characteristic</td>
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<td>$V$</td>
<td>Static force, vertical component</td>
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<tr>
<td>$x,y$</td>
<td>Local axes, y aligned with static load resultant</td>
<td>m,m</td>
</tr>
<tr>
<td>$X,Y$</td>
<td>Global axes, fixed, parallel to $H,V$</td>
<td>m,m</td>
</tr>
</tbody>
</table>
Introduction

The test rig was originally conceived and commissioned by Parkins in the early 1970's at Cranfield University, UK (1). It has been in regular use since, being modified and developed to suit many industrial and student research projects, also under the supervision of Parkins (2). In 1997 it was transferred to the Open University, UK, and has since been modified and re-commissioned for further studies under the supervision of Martin.

Although much modified since its early days the rig remains the same in principle. It comprises a stiff supporting framework, two orthogonally positioned vibrators (shakers), and systems for static load, oil supply, drive motor, instrumentation and control. In this respect it is similar to the rigs of Brockwell (3) and Flack (4) et al. Experimental work by these, and many others reported in English, have been neatly summarised by Swanson and Kirk (5).

In essence the main development changes to the test rig comprise the following:

- Realignment of the vibrator axes from 0 and 90° to ± 45° at ground level for reasons of support frame stiffness and to simplify oil drain arrangements.

- Extension of static load direction range from 0 to 90° to 0 to 180° to increase the range of direction.

- Substitution of remote belt drive by specially designed directly coupled drive motor rated at a speed of 25 000 rev. min⁻¹.

- Complete renewal of operating and data collection computing software.

Other changes have been implicit such as re-routing oil and electrical systems, new vibrator connector links, computing hardware etc. The detailed specifications have been reported by Rowan (6), but the rig has recently undergone a complete upgrade in terms of the operating and data collection software. For all measurement systems close attention is paid to observations and recommendations regarding variable uncertainties as reported by Kostrzewsky and Flack (7).

Test Rig Description

Figure 1 shows the main framework with the bearing housing assembly in the centre. In this configuration the two vibrators are mounted to the base each at 45° to vertical. They each connect to the bearing housing via a lightweight hollow aluminium rod and a bridge piece. The bridge pieces support dynamic force transducers and protect them from lateral loads but have negligible stiffness in the direct load path. Visible in Figure 1 are 3 lateral threaded rods used to prevent housing motions other than radial translations. Figure 1 also shows the oil circuit pump and its motor drive, and the static load vertical connection and one horizontal connection in place. Figure 2 shows the main rig scheme with both vibrators in place and, in this instance, the vertical and both horizontal static load systems connected.

Figure 3 shows the test bearing housing with the vertical and one horizontal static load component connected. Wire loops connect the floating bearing housing in the vertical and horizontal directions to trunnion blocks with pulleys. These blocks can be connected with further wire loops,
or pin jointed links, to static load cells via large springs. The purpose of the springs is to allow sufficient resilience in the load paths to maintain forces constant irrespective of the bearing housing motions. Such motions themselves, being of the order $0.05 \times 10^{-3}$ times the distance to fixed load points also have a negligible effect on the static load. All pulleys and pin joints have very small frictional resistances, so the bearing housing has freedom to rotate about 2 mutually perpendicular axes whilst under a steady static load, with negligible torsion effects. A separate restraining link reacts the oil film torque.

The static loads are effected by electrically driven screw jacks mounted at the far ends of the load paths, thus by appropriate vectoring a net load can be quickly set in any direction between 0 and $180^\circ$. The jack motors are speed controlled via variable frequency reversing inverters and, with gear ratios of 20 to 1, allow both a capacity of 10 kN, and very small increments of load, to be swiftly applied and monitored remotely.

Figure 4 is an assembly drawing of the latest direct drive and shaft support arrangement. The steel shaft is supported by 4 high precision grease lubricated angular contact bearings arranged as facing tandem pairs preloaded by 8 compression springs. The diameter of the journal in the test-bearing region is 80 mm nominal. A specially made high frequency induction motor is connected directly to the test shaft by an aluminium flexible coupling. The motor is water-cooled, has a rated maximum speed of 25 000 rev. min.$^{-1}$ and maximum output power of 51 kW. Motor control is by a variable frequency inverter and isolator unit. Test bearings are contained in the steel bearing housing which also contains mountings for the test instrumentation, static and dynamic load connections and the oil inlet and outlet. Visible in Figure 3 are 4 transducer mountings and the oil inlet.

The dynamic loads are generated by the large vibrator units (shakers). These are effectively wide frequency band force transducers, each capable of operating from sinusoidal or other wave-forms, or random signal inputs, in the range 5 to 4000 Hz. They are air-cooled by separate fan and blower units. Drive is by power amplifiers each safety tripped by cooling air pressure sensors from the vibrators. The vibrator stator frames are attached by very stiff struts aligned in the direction of the applied dynamic load and attached rigidly to the main frame thereby providing closed force loops.

Oil supply is from a reservoir and motor driven gear pump of capacity 29 L min$^{-1}$ and maximum supply pressure of 100 psi (689.5 kPa). Inlet pressure is adjusted as required with a bypass valve and recorded with a Bourdon type gauge. The inlet to the bearing housing is by flexible plastic tubing and the outlet a simple gravity drain arrangement back to the reservoir. The reservoir oil can be heated by immersion heater or cooled by an oil to water heat exchanger. A cartridge full flow filter is in series retaining particles in excess of 10 $\mu$m.

Figure 5 shows the main instrumentation processing and control units mounted separately but near to the main frame assembly. Electrical power is by 220 volt single phase or 410 volt 3 phase mains supply with emergency isolators as appropriate and all metalwork safety earthed.

**Instrumentation**

The primary variables to be measured are a) the relative displacement of journal and bearing; b) bearing static load; c) bearing dynamic load; d) bearing acceleration; e) shaft rotational speed and f) oil temperatures. Data derived are bearing equilibrium position, attitude angle, bearing
translation velocities and velocity (damping) coefficients. For some of the methods presently adopted, bearing displacement (stiffness) coefficients are obtained from separate experiments.

The bearing to journal horizontal and vertical displacements are monitored at both ends of the test bearing by 8 inductive proximity transducers, with reference to a global axis set parallel to \( H \) and \( V \). The transducers are set at an average distance of 1 mm from the surface of the journal and connected in push-pull mode to improve the linearity of their response characteristics. This type of transducer is commonly used in such applications primarily because they are not affected by the varying presence of oil in between the probe tip and target surface. It has been found, however, over many years of testing and with a variety of sizes and models that during the course of experiments there tends to be a drift in the effective location of the electrical origin. This applies even to quite large test installations (8).

It has also been found that the characteristic gradients in terms of \( V \ m^{-1} \) remain both constant and linear, so experimental techniques have been developed to take account of these features. Data for both static and dynamic conditions are derived from changes in relative displacements rather than absolute values, and are taken at the same experimental set up. These experimental techniques minimise the effects of possible drifting of electrical origins during an experiment. The transducers themselves are calibrated before and after testing using master capacitance transducers fitted and used whilst the shaft is stationary and surfaces cleaned of any oil. The master transducers themselves are calibrated remotely using precision blocks. In larger installations it is possible to use gauges blocks or dial test indicators directly to calibrate the inductive transducers (8).

The static load cells are resistive transducers with signal voltage outputs amplified by linear DC amplifiers. Calibration is by standard weights and not prone to drifting. The dynamic force transducers are of quartz type with a range of \( \pm 5 \ kN \) at frequencies between 2 and 4000 Hz. 2 identical charge amplifiers boost each channel transducer voltage. Each was set up by applying a sinewave of 1 volt amplitude to the amplifier input and adjusting the gain to produce a sinusoidal output of 10 volts amplitude. The dynamic transducers are factory calibrated for all main project tests and checked again manually prior to installation by the sudden removal of a known static weight load.

Accelerations of the bearing housing in the vertical and horizontal directions are sensed by piezoelectric accelerometers mounted directly to the housing in line with \( H \) and \( V \). Their signal voltage outputs are amplified by charge amplifiers. Each channel is adjusted to produce a sinusoidal output of 1 volt peak amplitude for 1g (9.81 m s\(^{-2}\)) acceleration.

Shaft speed is sensed by an optical encoder mounted at the shaft free end on the far side of the drive motor. A precision glass disc fitted to the shaft is etched with a circular grating of 180 divisions including one master division mark. An optical sensor outputs 1 rectangular pulse voltage for each division. The master mark signal activates a universal frequency counter and also provides the counter start point to trigger the data sampling. The dynamic forces can thus be applied at any point in the rotational cycle to begin an experiment as data collection will begin at the appropriate pass of the master mark. For safety's sake speed is occasionally checked with a hand held tachometer.

Temperatures are measured with standard T type thermocouples (range -50 °C to + 200 °C), usually for monitoring bulk oil, oil inlet, oil outlet, test bearing pad/surfaces and, for high speed work, the outer races of the support rolling element bearings.
As explained by Parkins in the accompanying paper (9) the main objectives of the exercise are the generation and analysis of suitable bearing orbits related to known net dynamic loads. The bearing orbit is measured at each end plane of the bearing housing assembly and the two orbits can be displayed and compared to check for any out of plane motion. Usually the processing is carried out for an averaged orbit at the bearing lateral centre, in plane with the dynamic force directions. Signal noise and interference are addressed in three main ways: 1) Low pass filtering of inductive transducer amplified signals; 2) full earthing and screening of all metalwork and motor drives, especially the main drive motor housing; 3) study, recording and subtraction of residual vibration signals from the observed orbits to produce a net orbit. The residual signal effect can be a problem at higher speeds if its magnitude approaches that of the orbit widths but even so can be safely subtracted provided it is of constant form during the brief data acquisition period.

**Data Acquisition System**

Control and data acquisition are provided by a dedicated personal computer using virtual instrument panels. Input and output are through an acquisition card. The hardware system layout is shown in Figure 6. The software is specially written and provides the following facilities: a) Calibration characteristics of 4 inductive displacement transducer channels; b) acquisition of bearing orbit data for evaluation of dynamic coefficients; c) on-line subtraction of residual vibration signals, enabling display of orbits due only to net dynamic loading; d) generation of vibrator command signals, cyclic or random form; e) setting and monitoring of static load vector to bearing housing; f) monitoring of shaft rotational speed and other parameters.

Bearing position and sampling rates are governed by the shaft speed encoder signal the frequency of which is 180 x the rotational speed. There are 10 channels of data comprising voltage signals related to: The bearing displacements $R_1$ to $R_4$, the dynamic forces $F_1$ and $F_2$, the static load components $H$ and $V$, and the bearing acceleration components, $A_H$ and $A_V$. Assuming a maximum possible test speed in excess of 25 000 rev. min$^{-1}$ the sample rate could exceed 75 000 signals per second for each channel. The board used can satisfy this requirement. A standard virtual instrument language was used to develop the graphic user interface and board function drive for control, acquisition and analysis functions. Figure 7 shows an overview of the main operation functions and programs.

**Testing Procedure**

Having completed a detailed check-list, and performed all the calibrations, the rig is run at the required shaft speed and bearing static load with the lateral support rods and vibrator struts disconnected. Thermal stability is considered achieved when the voltage readings from the displacement transducers remain unchanged for at least 15 minutes. The 3 lateral locating rods are then each secured in turn to the location plate as carefully as possible so as not to disturb the bearing to journal position and alignment. Then each vibrator strut is connected and tightened, again so as not to disturb the equilibrium position. Both these tasks require a degree of manual dexterity and experience from the operators. At this stage the bearing housing is ready for testing.
First the static data are obtained. Taking advantage of any symmetry of the bearing design (for example $72^\circ$ for a 5 tilting pad bearing, or complete $360^\circ$ for a plain bearing) the static force components, $H$ and $V$, are set so as to apply the same known resultant load in particular directions. From the changes in equilibrium position it is a simple matter to deduce the attitude angle and bearing centre for the warmed up stable condition, and hence the journal eccentricity. (It should be noted however that determination of the eccentricity ratio is not simple owing to difficulty in obtaining the actual clearance circle diameter for the same conditions).

The bearing is now ready for the dynamic experiment. First using the data generation program the dynamic force frequencies and amplitudes are set. This is done on a trial and error basis with usually one force at a frequency equal to the shaft speed and the other at twice the shaft speed. For each trial setting a dynamic orbit is generated for a few seconds and inspected for shape. The objective is to obtain a full well defined figure of 8 shaped orbit with a clearly defined cross-over point at the equilibrium position, ideally with the cross-over lengths being straight and orthogonal. If the orbit is suitable the settings are held for the experiment which lasts only for a further few seconds and proceeds as follows:

Static load conditions residual vibration signal recorded, designated A;

Dynamic forces applied two shaft rotations later;

After 10 shaft rotations under the dynamic loading, orbits for up to 30 rotations are recorded, along with the force and acceleration data;

Dynamic forces stopped approximately one second after starting;

After a further 200 shaft rotations the static residual signal is again recorded, designated B;

Rig continues to operate whilst post experiment inspections are carried out;

If inspections are satisfactory data are stored and immediately processed on line. The rig is then ready for another experiment.

Data stored for each orbit briefly comprises the applied dynamic forces resolved into components parallel to $H$ and $V$ and updated to include the mass x acceleration terms of the bearing giving the net dynamic forces reacted by the oil film for each data point. Also the bearing orbit positions in terms of co-ordinates parallel to $H$ and $V$ are held for each data point, and in terms of time from the master mark data point and known shaft speed.

Processing is done on line immediately with interactive commands from the operator. First the residual signal A is subtracted on-line from signal B. The result is stored and a judgement made to determine if the resulting signal is close to zero (showing that the residual signal has not changed as a result of the experiment). Figure 8 shows 1 example of the residual signals for an experiment at 4000 rev.min$^{-1}$ on a pivoted shoe bearing. The upper traces show the orbit amplitude trace against time before and after the experiment, the lowest trace shows the on-line subtraction result. There are no automated selection criteria but it is clear that in this case the change in residual signal can be considered negligible.
If the inspection result is considered acceptable residual signal A is then subtracted on-line from each of 5 orbits recorded to produce net orbits. Of the 30 orbits available usually numbers 10,11,12,13,14, 15 are selected. The net orbits are inspected for suitability for analysis by the $p,q$ method (i.e. still with complete, well defined cross over region, near to equilibrium position, with cross-over lengths being straight and orthogonal). Satisfactory orbits and their data are labelled and stored for immediate analysis.

The analysis program curve fits the displacements for the nearest 6 data points to the notional crossover point for the first and second paths around the orbit and determines the alignment of these straight line portions with respect to axes parallel to $H$ and $V$. The interpolated crossover is used as the origin of the $p,q$ co-ordinates system from which the coefficients are obtained using the method described by Parkins in the accompanying paper (9). Briefly for one orbit the bearing traverses the origin twice. Once at time $t_1$ when $q = \dot{q} = 0$ and $\dot{p}$ is significant, and once at time $t_2$ when $p = \dot{p} = 0$ and $\dot{q}$ is significant. The net dynamic forces at these times are known (interpolated if necessary between those for adjacent data point times) and resolved into components parallel to $p$ and $q$. The related velocity coefficients for the $p,q$ co-ordinates system are then determined. These are then transposed to the $x,y$ axis system (aligned with $y$ parallel to the static load resultant direction) to produce $b_{xx}, b_{xy}, b_{yx}$ and $b_{yy}$. The complete experiment and related processing takes only a minute or two, so the whole experiment can be repeated for the same warm up setting and operating conditions. Typical results for a 5 pad pivoted shoe journal bearing are given by Parkins in the accompanying paper (9).

**Conclusions**

A test rig described has been developed over a number of years and used by many research workers to determine the dynamic velocity coefficients of a variety of fluid film hydrodynamic journal bearings. Recent projects have combined newly devised analysis and processing techniques with the latest operation and control software to quickly produce on-line processing of the 4 velocity coefficients for a single vibration orbit for a given set of experimental conditions.

The process can be immediately repeated for a number of successive orbits very close together in terms of time, and generated for the same test conditions, but which are independent of one another. Thus several sets of results can be separately determined whilst minimising the effects of any changing conditions.

The experimental conditions themselves can be maintained for the short time needed to process and inspect results. This allows the whole experiment to be repeated if necessary at the same warm up and operating conditions. Equally for given warm up conditions particular parameters such as load or speed can be quickly changed in readiness for a new experiment.

It is hoped that this approach, combined with the use of experimental techniques which minimise the effects of possible drifts of electrical origins, will lead to more velocity coefficient data being determined with greater reliability for a whole range of bearing designs and operating parameters. The rig should therefore be in continued use for the foreseeable future making a contribution to the knowledge of the operating characteristics of fluid film bearings, the task first set out 30 years beforehand.
Acknowledgements

Dr D. Rowan carried out major work on the test rig whilst it was at Cranfield University, with support from Michell Bearings, and subsequently at the Open University. Some descriptive passages and figures in this paper have been adapted from his thesis. Prof. Y. Zhang of Xi'an Jiaotong University carried out an in depth study of Dr Rowan's test results and made some valuable observations and recommendations for further improvements to the rig. The authors are also indebted to the Open University for continued encouragement and support.

The authors are also pleased to remember the skills and contributions of Keith Brockwell who was very interested in their work and had seen the test rig a number of times. Indeed he had helped negotiate a collaborative research memorandum of understanding between his and the Open University Research Groups. He had also embarked on a part time PhD project with the Open University, working on the operating characteristics of pivoted shoe journal bearings, and had made a very promising and characteristically thorough start before his illness claimed him.
<table>
<thead>
<tr>
<th>Reference</th>
<th>Author(s)</th>
<th>Title</th>
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8  Martin, J.K.
   Parkins, D.W.

9  Parkins, D.W.
Fig. 1 Main frame assembly
Fig. 2 Main frame scheme
Fig. 3 Bearing housing and static load connection
Fig. 4 Bearing housing and shaft support assembly
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Fig. 8 Residual signals subtraction
Fig. 9 Typical orbit, displacements $m \times 10^{-6}$
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Fig. 2 Main frame scheme
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Fig. 4 Bearing housing and shaft support assembly
Fig. 5 Instrumentation and control stack
Fig. 6 Hardware layout
Start

Read calibration file

Static Equilibrium | Data Generation | Data Acquisition | Data View | Data Analysis

End

Fig. 7 Software overview
Fig. 8  Residual signals subtraction

Fig. 9  Typical orbit, displacements m x 10^-6