Identification of Dynamic Bearing Parameters: A Review

R. Tiwari, A.W. Lees and M.I. Friswell

ABSTRACT—In this paper, we present a review of the experimental identification of dynamic parameters of bearings in a rotating machine. Major emphasis is given to vibration-based identification methods and the review encompasses descriptions of experimental measurement techniques, mathematical modeling, parameter extraction algorithms and uncertainty in the estimates applied to a variety of bearings. The parameter extraction algorithms include the descriptions of governing equations of the rotor-bearing system and identification methods in both time and frequency domains. The identification techniques have been classified based on methods used to excite the system. The review includes a variety of bearings and similar components, which play an active link between the rotating and stationary parts of a machine. Based on the state of the art in bearing identification, conclusions are made and future directions are suggested.

KEYWORDS—bearing, identification, rotor-dynamics

1. Introduction

In many industries the demand for high power and high speed together with uninterrupted and reliable operation is increasingly important. The accurate prediction and control of the dynamic behavior (unbalance response, critical speeds and instability) is another vital requirement. The bearings clearly constitute a vital component in any turbomachine and a good understanding of their dynamic behavior is a pre-requisite to the prediction of the machine's properties. Whilst there are many types of bearing, in this paper the focus of attention is fluid-film journal bearing. This restriction is imposed by the sheer volume of published work rather than rigorous logical demarcation. Similar techniques to many of those discussed below have also been applied to other types of bearings and seals. Some mention of these issues is given where appropriate.

Historically the theoretical estimation of the dynamic bearing characteristics has always been a source of error in the prediction of the dynamic behavior of rotor-bearing systems. Obtaining reliable estimates of the bearing operating conditions in actual test conditions is difficult and, consequently, physically meaningful and accurate parameter identification is required in actual test conditions to reduce the discrepancy between the measurements and the predictions. Experimental methods for the dynamic characterization of rolling element bearings, fluid-film bearings, magnetic bearings and seals have some similarities. In general, the methods require input signals (forces) and output signals (displacements/velocities/accelerations) of the dynamic system to be measured, and the unknown parameters of the system models are calculated by means of input–output relationships. The theoretical procedures and experimental measurements depend upon whether the bearing is investigated in isolation or installed in a rotor-bearing system.

In the present paper, a review has been made of the identification methods of bearings dynamic parameters. The main focus of our attention is fluid-film journal bearings, but because the techniques are common to various types of bearings and seals, reference to these are made where appropriate. A chronological list of source material on the experimental dynamic parameter identification of bearings with brief details can be found in Table 1.

2. Previous Literature Reviews and Surveys

The influence of bearings on the performance of rotor-bearing systems has been recognized for many years. One of the earliest attempts to model a journal bearing was reported by Stodola (1925) and Hummel (1926). They represented the fluid film as a simple spring support, but their model was incapable of accounting for the observed finite amplitude of oscillation of a shaft operating at a critical speed. Concurrently, Newkirk (1924) and Newkirk and Taylor (1925) described the phenomenon of bearing induced instability, which he called oil whip, and it soon occurred to several investigators that the problem of rotor stability could be related to the properties of the bearing dynamic coefficients. Ramsden (1967–68) was the first to review the papers on the experimentally obtained journal bearing dynamic characteristics. He concluded that a designer would require known stiffness and damping coefficients of the bearings. Since most of the data available at that time were experimental only, he stressed the need for accurate scaling laws to be evolved to avoid full-scale tests. In the late 1970s, Dowson and Taylor (1980) conducted a survey of the state of knowledge in the field of the influence of the bearings on rotor dynamics. They appreciated that a considerable amount of literature was

R. Tiwari, Department of Mechanical Engineering, Indian Institute of Technology Guwahati, 781039, India.
A. W. Lees (a.w.lees@swansea. ac.uk), School of Engineering, University of Wales Swansea, Swansea SA2 8PP, Wales, UK.
M. I. Friswell, Department of Aerospace Engineering, Queens Building, University of Bristol, Bristol BS8 1TR, UK.

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Table 1. Summary of experimental investigations. A chronological summary of papers on the experimental dynamic parameter identification of bearings.

References	Bearing type*	Type of excitation	Vibration response measured	Identified dynamic parameters
Hagg and Sankey (1956, 1958)	HDJ	Unbalance	Displ. (frequency)	Direct damping and stiffness
Mitchell et al. (1965–66)	HDJ	Incremental static load	Displacement	Stiffness
Duffin and Johnson (1966–67)	HDJ	Unbalance	Displ. (frequency)	Direct damping and stiffness
Glienicke (1966–67)	HDJ	Sinusoidal	Displ. (frequency)	Damping and stiffness
Woodcock and Holmes (1969–70)	HDJ	Unbalance	Displ. (frequency)	Damping and uncertainty
Black and Jenssen (1969–70)	PLS	Incremental static load	Displacement	Stiffness
Morton (1971)	HDJ	Sinusoidal	Displ. (frequency)	Damping and stiffness
Williams and Holmes (1971)	SQF	Static load	Ecc., alt., angle, vel.	Damping
Thomsen and Andersen (1974)	SQF	Unbalance	Displ. (frequency)	Damping
Morton (1975a)	HDJ	Step function	Displ. (frequency)	Damping and stiffness
Bannister (1976)	HDJ	Unbalance	Displ. (frequency)	Damping, stiffness and 28 non-linear coefficients
Tonnesen (1976)	SQF	Unbalance	Displ. (frequency)	Damping
Fleming et al. (1977)	GJ	Unbalance	Displ. (frequency)	Damping and stiffness
Wright (1978)	LS	Impact	Displ. (time)	Damping and stiffness
Parkins (1979)	HSJ	Bi-directional sine	Displ., vel. (time)	Damping and stiffness
Stanway et al. (1979a, 1979b)	SQF	PRBS	Displ. (time)	Damping
Childs et al. (1980)	Seals	Eccentric shaft	Displ. (frequency)	Mass, damping and stiffness
Diana et al. (1980)	HSJ	Inertial exciter	Displ. (frequency)	Damping and stiffness
Dogan et al. (1980)	HDJ	PRBS	Displ. (frequency)	Damping and stiffness
Hisa et al. (1980)	HDJ	Sinusoidal	Vel. (frequency)	Damping and stiffness
Iino and Kaneko (1980)	PLS	Synchronous	Displ. (frequency)	Mass, damping and stiffness
Nordmann and Schöllhorn (1980)	HDJ	Impact	Displ. (frequency)	Damping and stiffness
Walford and Stone (1980a, 1980b)	Ball	Sinusoidal	Displ., acc. (frequency)	Damping and stiffness
Burrows et al. (1981)	SQF	PRBS/SPHS	Displ. (frequency)	Damping
Parkins (1981)	HSJ	Bi-directional sine I	Displ., vel. (time)	Damping and stiffness
Burrows and Sahinkaya (1982a)	SQF	PRBS	Displ. (frequency)	Damping and uncertainty
Diana et al. (1982)	PLS	Synchronous	Displ. (frequency)	Stiffness
Stone (1982)	RE	Impact and Sine	Displ. (time, frequency)	Damping and stiffness
Goodwin et al. (1983)	HSJ	Unbalance	Displ. (frequency)	Damping and stiffness
Wright (1983)	GS	Synchronous	Displ. (frequency)	Damping and stiffness
Burrows et al. (1984)	SQF	SPHS	Displ. (frequency)	Damping
Falco et al. (1984)	PLS	Sinusoidal	Displ. (frequency)	Damping and stiffness
Goodwin et al. (1984)	HYJ	Unbalance	Displ. (frequency)	Damping and stiffness
Kanki and Kawakami (1984)	PLS	Sinusoidal	Displ. (frequency)	Mass, damping and stiffness
Nordmann and Massmann (1984)	ANS	Impact	Displ. (frequency)	Mass, damping and stiffness
Sahinkaya et al. (1984)	SQF	SPHS	Displ. (frequency)	Damping and uncertainty
Sahinkaya and Burrows (1984a)	SQF	SPHS	Displ. (time)	Damping and uncertainty
Burrows and Sahinkaya (1985)	SQF	SPHS	Displ., vel. (time, frequency)	Damping and uncertainty
Chang and Zheng (1985)	HDJ	Step function	Displ., acc. (frequency)	Damping and stiffness (0/4/4)
Childs and Kim (1985)	ALS	Eccentric rotor	Displ. (frequency)	Mass, damping and stiffness
Kaushal et al. (1985)	SQF	Unbalance	Displ. (frequency)	Damping (0/1/0) and uncertainty
Childs and Kim (1986)	DS	Eccentric rotor	Displ. (frequency)	Mass, damping and stiffness
Childs and Scharrer (1986)	GS	Sinusoidal	Displ. (frequency)	Damping and stiffness
Nelson et al. (1986)	ALS	Sinusoidal	Displ. (frequency)	Mass, damping and stiffness
Roberts et al. (1986)	SQF	Step function	Displ. (time)	Mass and damping
Childs and Garcia (1987)	DS	Eccentric rotor	Displ. (frequency)	Mass, damping and stiffness
Kraus et al. (1987)	Ball	Impact	Displ. (frequency)	Mass, damping and stiffness
Ramli et al. (1987)	SQF	Transient	Displ. (time)	Direct damping and stiffness
Stanway et al. (1987)	SQF	PRBS	Displ. (frequency)	Damping
Burrows et al. (1988a)	SQF	Unbalance	Displ. (frequency)	Mass and damping
Childs and Scharrer (1988)	LGS	Uni-directional sine	Displ. (frequency)	Damping, stiffness uncertainty
Ellis et al. (1988)	SQF	Step/sine	Displ. (time)	Direct damping and stiffness
Kang and Jin (1988)	HDJ	Impact	Displ., acc. (time)	Damping and stiffness (0/4/4)

Table 1. Summary of experimental investigations. A chronological summary of papers on the experimental dynamic parameter identification of bearings. (Continued)

References	Bearing type*	Type of excitation	Vibration response measured	Identified dynamic parameters
Roberts et al. (1988)	SQF	Sinusoidal	Displ. (frequency)	Mass, damping and stiffness
Stanway et al. (1988)	SQF	Synchronous	Displ. (frequency)	*n*th-power velocity damping
Childs et al. (1989)	HCS	Sine-sweep	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Ellis et al. (1989)	SQF	Bi-directional sinel	Displ. (time)	Mass, damping and stiffness
Elrod et al. (1989)	HCS	Uni-directional sine	Displ. (frequency)	Damping, stiffness and uncertainty
Hawkins et al. (1989)	AGS	Uni-directional sinel	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Kanemori and Iwatsubo (1989, 1992)	PLS	Eccentric rotor	Displ. (frequency)	Mass, damping and stiffness
Someya (1989)	HDJ	Sinusoidal	Displ. (frequency)	Damping and stiffness
Brockwell et al. (1990)	HDJ	Bi-directional sine	Displ. (frequency)	Damping and stiffness
Burrows et al. (1990)	SQF	SPHS	Displ. (frequency)	Mass, damping and uncertainty
Childs et al. (1990a)	HCS	Uni-directional sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Childs et al. (1990b)	ANS	Eccentric rotor	Displ. (frequency)	Mass, damping and stiffness
Childs et al. (1990c)	DS	Eccentric rotor	Displ. (frequency)	Mass, damping and stiffness
Ellis et al. (1990)	SQF	Random	Displ. (time)	Mass, damping and stiffness
Fritzen and Seibold (1990)	ANS	Impact	Displ. (time)	Mass, damping and stiffness.
Iwatsubo and Sheng (1990)	DS	Eccentric sleeves	Displ. (frequency)	Mass, damping and stiffness
Kim et al. (1990)	TR	Impulse	Acc. (frequency)	Direct stiffness and damping
Kostrzewsky et al. (1990)	HDJ	Bi-directional sine	Displ. (frequency)	Stiffness and uncertainty
Matsumoto et al. (1990)	HDJ	Two-directional sine	Displ. (frequency)	Damping and stiffness
Mohammad and Burdess (1990)	HDJ	Random	Displ. (time)	Damping and stiffness
Muszynska and Bently (1990)	HDJ	Sine sweep	Displ. (frequency)	Damping and effective stiffness
Roberts et al. (1990)	SQF	Step function	Displ. (time)	Mass, damping and uncertainty
Rouch (1990)	SQF	Sinusoidal	Displ. (frequency)	Damping and stiffness
Yanabe et al. (1990)	HDJ	Incremental static/ unbalance/impact	Displacement (frequency)	Damping and stiffness
Chan and White (1991)	TPJ	Impact	Displ. (frequency)	Damping and stiffness
Childs and Ramsey (1991)	AGS	Sine sweep	Displ. (frequency)	Damping, stiffness and uncertainty
Childs et al. (1991)	AGS	Sine sweep	Displ. (frequency)	Damping, stiffness and uncertainty
Imlach et al. (1991)	MB	Incremental static load	Displacement	Direct stiffness
Jung et al. (1991a, 1991b)	SQF	Eccentric sleeve	Displ. (frequency)	Mass and damping
Murphy and Wagner (1991)	HSJ	Eccentric shaft	Displ. (frequency)	Damping, stiffness and uncertainty
Wang and Liou (1991)	Ball	Impact	Displ. (frequency)	Damping and stiffness
Adams et al. (1992)	HSJ	Unbalance	Displ. (frequency)	Mass, damping and stiffness
Brown and Ismail (1992, 1994)	ANS	Multi-frequency	Displ. (time)	Damping and stiffness
Childs and Kleynhans (1992)	ANS	Sine sweep	Displ. (frequency)	Damping, stiffness and uncertainty
Myllerup et al. (1992)	HDJ	Incremental static load	Displacement	Stiffness
Rouvas et al. (1992)	HSJ	Impact	Displ., acc. (frequency)	Mass, damping and stiffness
Sakakida et al. (1992)	HSJ	Sinusoidal	Displ., acc. (frequency)	Damping and stiffness
Zhang et al. (1992a, 1992b)	HDJ	Impact	Displ. (frequency)	Damping, stiffness and uncertainty
Conner and Childs (1993)	BS	Sine-sweep	Displ. (frequency)	Damping, stiffness and uncertainty
Flack et al. (1993)	HDJ	Bi-directional sine l	Displ. (frequency)	Damping, stiffness, uncertainty
Jung and Vance (1993)	SQF	Eccentric sleeve	Displ. (frequency)	Mass and damping
Muszynska et al. (1993)	Seals	Unbalance	Displ. (frequency)	Radial damping and stiffness
Parkins and Horner (1993)	HDJ	Incremental static load	Displacement	Stiffness
Roberts et al. (1993)	SQF	Sinusoidal	Displacement (time)	Non-linear model
Rouvas and Childs (1993)	HSJ	PRBS and Sine	Displ., acc. (frequency)	Mass, damping and stiffness
Childs and Hale (1994)	HSJ	PRBS	Displ., acc. (frequency)	Mass, damping and stiffness
Franchek and Childs (1994)	HYJ	PRBS	Displ., acc. (frequency)	Mass, damping and stiffness
Heshmat and Ku (1994)	HDJ (foil)	Sinusoidal	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Kim and Lee (1994)	Seals	Impact	Displ. (frequency)	Mass, damping and stiffness
Kostrzewsky et al. (1994)	HDJ	Bi-directional sine	Displ. (frequency)	Mass, damping and stiffness

Table 1. Summary of experimental investigations. A chronological summary of papers on the experimental dynamic parameter identification of bearings. (Continued)

References	Bearing type*	Type of excitation	Vibration response measured	Identified dynamic parameters
Ku (1994)	FTB	Sinusoidal	Displ. (frequency)	Damping and stiffness
Ku and Heshmat (1994)	Foil HDJ	Bi-directional sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Tieu and Qiu (1994)	HDJ	Unbalance	Displ. (frequency)	Damping and stiffness
Xu (1994)	HYJ	Sinusoidal	Displ. (frequency)	Damping and stiffness
Zhang and Roberts (1994)	SQF	Sinusoidal	Displ. (frequency)	Non-linear force coefficients
Zhang et al. (1994)	SQF	Sinusoidal	Displacement (time)	Mass, damping and stiffness
Alexander et al. (1995)	AGS	PRBS	Displ., acc. (frequency)	Damping and stiffness
Arumugam et al. (1995)	HDJ	Uni-directional sine	Displ. (frequency)	Damping and stiffness
Chen and Lee (1995)	Ball	Unbalance	Displ. (frequency)	Damping and stiffness
Dmochowski and Brockwell (1995)	HDJ	Bi-directional sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Franchek et al. (1995)	HYJ	PRBS	Displ., acc. (frequency)	Mass, damping and stiffness
Parkins (1995)	HDJ	Bi-directional sine	Velocity (time)	Damping and uncertainty
San Andrés et al. (1995)	HSJ	PRBS	Displ., acc. (frequency)	Mass, damping and stiffness
Santos (1995)	HDJ	Sinusoidal	Displ. (frequency)	Damping and stiffness
Taylor et al. (1995)	HDJ	Sinusoidal	Displ. (frequency)	Damping, stiffness and uncertainty
Tiwari and Vyas (1995, 1997a, 1998)	Ball	Random	Displ., vel. (time)	Non-linear stiffness
Childs and Gansle (1996)	AGS	Uni-directional sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Ismail and Brown (1996)	ANS	Multi-frequency	Displ., vel., acc. (time)	Damping and stiffness
Kostrzewsky et al. (1996)	HDJ	Bi-directional sine	Displ. (frequency)	Damping, stiffness and uncertainty
Lee et al. (1996)	MB	Random	Current, volts, displ. (frequency)	Damping and stiffness Current stiffness
Qiu and Tieu (1996)	HDJ	Comp. sine	Displ. (frequency)	Damping, stiffness and uncertainty
Santos (1996)	HDJ	Sine sweep	Displ. (frequency)	Damping and stiffness
Vance and Li (1996)	DS	Impact	Displ. (time)	Damping
Zhang and Roberts (1996)	SQF	Sinusoidal	Displ. (frequency)	Mass and damping
Arumugam et al. (1997a, 1997b)	HDJ	Uni-directional sine	Displ. (frequency)	Damping, stiffness and uncertainty
Chen and Lee (1997)	Ball	Unbalance	Displ., vel. (time)	Damping and stiffness
Goodwin et al. (1997)	HDJ	PRBS	Displ. (frequency)	Damping and stiffness
Jiang et al. (1997)	HDJ	Impulse	Displ. (frequency)	Damping and stiffness
Marquette et al. (1997)	PLS	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Marsh and Yantek (1997)	RB	Impact	Acc. (frequency)	Direct stiffness
Müller-Karger et al. (1997)	HDJ	Sinusoidal	Displ. (frequency)	Damping, stiffness and uncertainty
Prabhu (1997)	HDJ	Sinusoidal	Displ. (frequency)	Damping
Qiu and Tieu (1997)	HDJ	Impulse	Displ. (frequency)	Damping and stiffness
Reddy et al. (1997)	HDJ	Sinusoidal	Displ. (frequency)	Damping and stiffness
San Andrés and Childs (1997)	HSJ	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Sawicki et al. (1997)	HYJ	Unbalance	Displ. (frequency)	Mass, damping, stiffness and uncertainty
Tiwari and Vyas (1997b)	Ball	Ran/unbalance	Displ., vel. (time)	Non-linear stiffness
Ismail and Brown (1998)	PLS	SPHS	Displ., acc. (time)	Mass, damping, stiffness and uncertainty
Kaneko et al. (1998)	ANS	Eccentric sleeves	Displ. (frequency)	Mass, damping, stiffness and uncertainty
Kostrzewsky et al. (1998)	HDJ	Bi-directional sine	Displ. (frequency)	Damping, stiffness and uncertainty
Mitsuya et al. (1998)	Ball	Impact	Displ. (frequency)	Damping and stiffness
Mosher and Childs (1998)	HYJ	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Nikolakopoulos and Papadopoulos (1998)	HDJ (ER)	Incremental static load	Displacement	Stiffness
Royston and Basdogan (1998)	Ball	Random	Displ. (frequency)	Axial and radial stiffness
Yu and Childs (1998)	GS HCS	Uni-directional sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty

Table 1. Summary of experimental investigations. A chronological summary of papers on the experimental dynamic parameter identification of bearings. (Continued)

References	Bearing type*	Type of excitation	Vibration response measured	Identified dynamic parameters
Childs and Fayolle (1999)	PLS	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Fayolle and Childs (1999)	HYJ	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Ha and Yang (1999)	HDJ	Bi-directional sine	Displ., acc. (frequency)	Damping and stiffness
Howard (1999)	FAB	Incremental static load	Displacement	Damping and stiffness
Kim and Lee (1999)	MB	Magnetic	Control current, displ.	Current and position stiffness
Laurant and Childs (1999)	HYJ	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Li et al. (1999)	GDS	Impact	Displ. (frequency)	Damping and stiffness
Pettinato and Choudhury (1999)	HDJ	Incremental static load/unbalance	Displ. (frequency)	Damping, stiffness and uncertainty
Ransom et al. (1999)	GDS	Impact	Displ. (frequency)	Damping, stiffness and uncertainty
Soto and Childs (1999)	HCS	Swept sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Wygant et al. (1999)	HDJ	Bi-directional sine	Displ. (frequency)	Damping, stiffness and uncertainty
Laos et al. (2000)	GDS	Periodic chirp	Displ., acc. (frequency)	Damping and uncertainty
Li et al. (2000)	GDS	Impact	Displ., acc. (frequency)	Damping and stiffness
Lindsey and Childs (2000)	PLS	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Shamine et al. (2000)	RE	Impact	Acc. (frequency)	Stiffness and damping (tilt)
Tiwari (2000)	Ball	Random	Displ., vel. (time)	Non-linear stiffness
Vance and Ying (2000)	ER	Impact	Displacement (time)	Damping (0/2/0)
Zarzour and Vance (2000)	MD	Impact	Acc. (time)	Damping and stiffness
Howard et al. (2001)	FAB	Impact	Displacement (time)	Damping and stiffness (0/1/1)
Nielsen et al. (2001)	AGS	Swept-sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Pettinato and Flack (2001)	HDJ	Bi-directional sine	Displ. (frequency)	Damping, stiffness and uncertainty
Pettinato et al. (2001)	HDJ	Bi-directional sine	Displ. (frequency)	Damping, stiffness and uncertainty
San Andrés et al. (2001)	SQF	Impact and sine	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Tiwari et al. (2002)	SPR	Unbalance	Displ. (frequency)	Damping and stiffness
Dawson et al. (2002a, 2002b)	AGS	PRBS	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Holt and Childs (2002)	AGS	PRBS	Displ., acc. (frequency)	Damping, stiffness and uncertainty
Laurant and Childs (2002)	HYJ	PRBS	Displ., acc. (frequency)	Mass, damping, stiffness and uncertainty
Vazquez et al. (2002)	SPR	Swept-sine	Acc. (frequency)	Damping and stiffness
Chatterjee and Vyas (2003)	Ball	Sinusoidal	Displ. (frequency)	Damping and non-linear stiffness
Kaneko et al. (2003)	ALS	Eccentric sleeves	Displ. (frequency)	Mass, damping, stiffness and uncertainty
Weatherwax and Childs (2003)	AGS	Eccentric sleeves	Displ., acc. (frequency)	Damping, stiffness and uncertainty

* The following abbreviations are used in the table: AGS, annular gas seal; ANS, annular seal; ALS, annular liquid seal; BS, brush seals; DS, damper seals; ER, electrorheological fluid; FAB, foil air; FTB, foil thrust; GDS, gas damper seal; GJ, gas journal; GS, gas seal; HCS, honeycombed seal; HDJ, hydrodynamic journal; HSJ, hydrostatic journal; HYJ, hybrid journal; LGS, long gas seal; LS, long seal; MB, magnetic; MD, metal mess bearing damper; PLS, plain liquid seal; RB, recirculating ball; RE, rolling element; SPR, springs; SQF, squeeze film; TPJ, tilting pad journal; TR, tapered roller.

available on both rotor dynamics and fluid-film bearings but relatively few attempts had been made to integrate the individual studies of rotor behavior and of dynamic characteristics of bearings in the field of tribology. Several conclusions and recommendations were made by them, most importantly: (a) experimental work in the field of rotor dynamics to study the influence of bearings and supports upon the rotor response was required; (b) additional theoretical studies to consider the influence of thermal and elastic distortion, grooving arrangements, misalignment, cavitation and film reformation were also needed.

An extensive literature is now available on the theoretical calculation of the dynamic characteristics of variety of bearings: rolling element (Palmgren, 1959; Ragulskis et al., 1974; Gargiulo, 1980; Harris, 2001; Changsen, 1991; Gupta 1984); fluid film (Pinkus and Sternlicht, 1961; Smith, 1969; Hamrock, 1994); magnetic bearings (Schweitzer et al., 1994) and seals (Childs, 1993). This reflects the technological importance of the topic. Among the various bearings available, fluid-film bearings, especially hydrodynamic cylindrical-journal bearings, attracted the most interest of practicing engineers and researchers. This is because of their relative simplicity

in the geometry and consequently in the analysis and experimental rig fabrication along with their general representation in mathematical modeling among the all other classes of the bearings. Lund (1980, 1987) reviewed the concept of dynamic coefficients for fluid-film journal bearings.

Lund (1979) gave a review of the theoretical and experimental methods for the determination of fluid-film bearing dynamic coefficients. He indicated the need to include the impedance of the rotor and categorized the experimental methods based on the excitation method used (for example, static load and dynamic force, i.e. harmonic, transient and random). The experimental static locus did not coincide with the theoretical one and it was difficult to analyze the degree to which the result might be affected by measurement tolerances and by uncertainties in establishing test parameters. Lund concluded that there was little point in refining the dynamic analysis until better agreement between theory and measurements had been obtained on the performance of static bearings. Stanway et al. (1979a) gave a brief appraisal of bearing dynamic testing methods. They stressed the need for development of an experimental method that was capable of use with the rotor-bearing system being run under normal operating conditions and placed some confidence bounds on the estimates. Someya (1989) compiled extensively both analytical and experimental results (static and dynamic parameters) for various fluid-film bearing geometries (for example, two-axial groove, two-lobe, four- and five-pad tilting pad).

Other types of bearing were also being studied at this time. Rouch (1990) reviewed briefly the theoretical and experimental developments in squeeze-film bearing dynamic coefficients estimation.

Goodwin (1989, 1991) reviewed the experimental approaches to rotor support impedance measurement. He particularly emphasized the measurement of fluid-film bearing dynamic coefficients and categorized the identification methods by the way in which the loading was applied (static load and dynamic force, i.e. using vibrator, unbalance or transient force). He concluded that measurements made by multi-frequency test signals provide more reliable data, although all measurement methods yield coefficients values which agree with theoretical predictions to within about 20%, and all methods have a significant scatter of results. Childs (1993) gave a comprehensive survey of the rotordynamic experimental data for liquid and gas annular seals and turbines and pump impellers along with their geometry and operating conditions. Swanson and Kirk (1997) presented a survey in tabular form of the experimental data available in the open literature for fixed geometry hydrodynamic journal bearings. They categorized the literature based on bearing type, bore diameter, length to diameter ratio and type of data available (i.e. static parameters such as film pressure, bearing temperature and shaft position; dynamic parameters such as stiffness, damping and added-mass coefficients).

3. Basic Concepts and Assumptions of Bearing Models

For a given bearing and rotational speed, lubrication theory may be used to calculate the reaction forces on the journal from the lubricant film, which are functions of the displacements of the journal from bearing center and of the instantaneous journal center velocities and accelerations. Hence, for

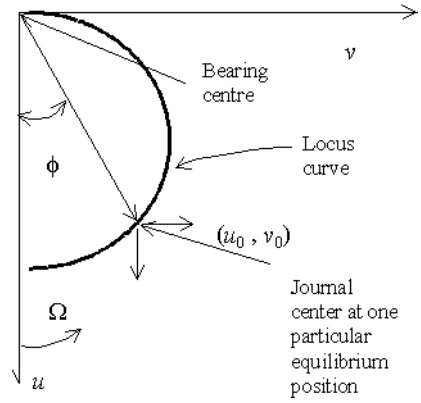

Figure 1. The equilibrium locus of the journal center of a fluid-film bearing.

small amplitude motions, measured from the static equilibrium position (see Figure 1) of the journal (u_0, v_0), a first-order Taylor series expansion yields

$$\Re_x = \Re_{u_0} + k_{xx}x + k_{xy}y + c_{xx}\dot{x} + c_{xy}\dot{y} + m_{xx}\ddot{x} + m_{xy}\ddot{y}$$
$$\Re_y = \Re_{v_0} + k_{yx}x + k_{yy}y + c_{yx}\dot{x} + c_{yy}\dot{y} + m_{yx}\ddot{x} + m_{yy}\ddot{y}$$

where (1)

$$k_{xy} = \left(\frac{\partial \Re_x}{\partial y}\right)_{(u_0,v_0)} \quad c_{xy} = \left(\frac{\partial \Re_x}{\partial \dot{y}}\right)_{(u_0,v_0)} \quad m_{xy} = \left(\frac{\partial \Re_x}{\partial \ddot{y}}\right)_{(u_0,v_0)}$$
$$\Re_x = f_x - m\ddot{x} \qquad \Re_y = f_y - m\ddot{y}$$

and analogously the remaining bearing dynamic coefficients may be defined. In the matrix form of equation (1) all diagonal terms are called *direct* coefficients and off-diagonal terms are called *cross-coupled* terms. The latter terms arise due to the fluid rotation within the bearing. $\Re$ is the reaction force of fluid film on the journal, f is the external excitation force on the journal, m is the journal mass, u_0 and v_0 are the static equilibrium position of the journal from the bearing center, x and y are the displacements of the journal from its static equilibrium position, $\dot{x}$ and $\dot{y}$ are the instantaneous journal center velocities, and $\ddot{x}$ and $\ddot{y}$ are the instantaneous journal center accelerations, in the vertical and horizontal directions, respectively. The "dot" indicates the time derivatives and k_{ij}, c_{ij} and m_{ij} $(i, j = x, y)$ are bearing stiffness, damping, and added-mass (also termed the virtual fluid-film mass or inertia) coefficients, respectively. The indices of the stiffness, damping and added-mass coefficients have the following significance: the first index gives the direction of loading which produces the elastic (damping/inertia) force and the second index gives the direction of the displacement (velocity/acceleration). Because (u_0, v_0) is the equilibrium position, then $\Re_{u_0}$ equals the static load, W. The equilibrium position depends on a unique value of the dimensionless Sommerfeld number $(S = (\mu\Omega RL/W)(R/c_r)^2(L/D)^2)$, where μ is the lubricant viscosity, Ω is the journal rotational speed, D is the bearing bore, R is the journal radius, L is the bearing length, and c_r is the bearing radial clearance. The Sommerfeld number, S, defines the operating conditions (speed, lubricant viscosity, static load, and geometry). The dynamic

coefficients are evaluated for a particular static equilibrium position, which is a function of the Sommerfeld number, S, which means that for a given application, they are functions of rotor speed. Moreover, bearing dynamic coefficients could be external excitation frequency, ω, dependent.

It should be noted that equation (1) is a complete form of linearized fluid-film dynamic equation and it contains twelve stiffness, damping and added-mass coefficients. Consistent with the assumptions inherent in reducing the Navier–Strokes equations to the Reynolds equation, the conventional laminar, thin-film lubrication theory ignores the inertia forces in the fluid film (Pinkus and Sternlicht, 1961; Schlichting, 1960). This is theoretically justified for small values of the Reynolds number (of the order of 1). On the other hand, the assumption of laminar flow ceases to be valid when there is transition to either Taylor vortex flow or to turbulent flow, which for fluid-film cylindrical journal bearing occurs at a Reynolds number of approximately 1000–1500. Thus, there is an intermediate range, say for values of Reynolds number of the order of 100, where added-mass effects may become noticeable (several times the mass of the journal itself) without affecting the assumption of laminar flow. The added-mass coefficients represent the mass of the bearing fluid film (Reinhardt and Lund, 1975) but are significant only in exceptional cases and in most analyses the added-mass of the bearing film is ignored. The stiffness and damping coefficients can be obtained by a finite difference solution of the perturbed Reynolds equation (Lund and Thomsen, 1978).

From lubrication theory (ignoring inertia effects) damping coefficients are symmetric but stiffness coefficients are not. Therefore, principal directions do not exist; although this was assumed by Hagg and Sankey (1956) and Duffin and Johnson (1966–67). Also, in the experimental determination of the coefficients, it is necessary to obtain two independent sets of amplitude–force measurements. Lund (1987) emphasized the experimental measurement of the bearing coefficients and established more uniform agreement with analytical calculations by considering the influence of thermal and elastic deformations and practical problems of manufacturing and operating tolerances of bearing geometry, clearance and lubricant viscosity. Although the load–displacement characteristics of a journal bearing is evidently non-linear, the concept of linear dynamic coefficients is still used for modern rotor dynamic calculations for unbalance response, damped natural frequencies and stability since experience has demonstrated the usefulness of the coefficients. Clearly the applicability of these techniques is dependent on small amplitudes of vibration at the journal (relative to the bearing clearance).

For squeeze-film bearings the governing equation for the fluid-film reaction force has a similar form to equation (1) with negligible stiffness coefficients and no static load:

$$\begin{aligned} \Im_x &= m_{xx}\ddot{x} + m_{xy}\ddot{y} + c_{xx}\ddot{x} + c_{xy}\dot{y} \\ \Im_y &= m_{yx}\ddot{x} + m_{yy}\ddot{y} + c_{yx}\dot{x} + c_{yy}\dot{y}. \end{aligned} \tag{2}$$

Mittwollen et al. (1991) showed theoretically and experimentally that hydrodynamic thrust bearings, which are often treated as an axial support, might affect the lateral vibration of a rotor-bearing system. If no axial force is present, then the resulting reaction moments of a thrust bearing can be written in terms of stiffness and damping coefficients (Jiang and Yu, 1999, 2000)

$$\begin{aligned} \aleph_x &= k_{x\theta}\theta + k_{x\psi}\psi + c_{x\theta}\dot{\theta} + c_{x\psi}\dot{\psi} \\ \aleph_y &= k_{y\theta}\theta + k_{y\psi}\psi + c_{y\theta}\dot{\theta} + c_{y\psi}\dot{\psi} \end{aligned} \tag{3}$$

where $k_{x\theta}$, etc., represent moment dynamic coefficients of the thrust bearing, and θ and ψ are angular displacements (slopes) in the x and y directions, respectively.

All of the bearing models discussed thus far are linearized models. A few researchers have considered non-linear bearing models and these will be described in appropriate places. The present literature survey is aimed at the review of experimental methods for the determination of the parameters of the bearings in rotor-bearing systems, and will hopefully be useful to both to practicing engineers and to researchers in this field. For the practicing engineer guidance for simple experimental determination of these parameters with associated uncertainty is offered, whilst researchers may appreciate the diverse methods available and the discussion of their limitations so as to develop improved methods.

4. Methods Using Incremental Static Load

Mitchell et al. (1965–66) obtained the four stiffness coefficients of fluid-film bearings experimentally by incrementally loading the bearing and measuring the change in position. They used the influence coefficient approach to obtain the following simple relationships

$$\begin{aligned} k_{xx} &= (\alpha_{yy}/\gamma) \quad & k_{xy} &= -(\alpha_{xy}/\gamma) \\ k_{yx} &= -(\alpha_{yx}/\gamma) \quad & k_{yy} &= (\alpha_{xx}/\gamma) \end{aligned} \tag{4}$$

where

$$\begin{aligned} \gamma &= \alpha_{xx}\alpha_{yy} - \alpha_{xy}\alpha_{yx} \\ \alpha_{xx} &= (x_1/\Delta F_x) \qquad & \alpha_{yx} &= (y_1/\Delta F_x) \\ \alpha_{xy} &= (x_2/\Delta F_y) \qquad & \alpha_{yy} &= (y_2/\Delta F_y) \end{aligned} \tag{5}$$

and x_1 and y_1 are displacements in vertical and horizontal directions respectively, of the journal center from its static equilibrium position on the application of a static incremental load ΔF_x in the vertical direction and x_2 and y_2 are displacements corresponding to a static incremental load ΔF_y in the horizontal direction. The indices of the influence coefficients, α, have the following significance: the first index gives the direction of displacement and the second index gives the direction of the force (i.e. opposite to the bearing dynamic coefficients). Since the estimation of stiffness requires the establishment of a relationship between the force and the corresponding displacement, this method can be applied to any type of bearing. Using this approach, Benckert and Wachter (1980) obtained seal stiffness coefficients by statically displacing a rotor with seals relative to its stator. The circumferential pressure distribution was measured and integrated and the resulting reaction force was calculated. If there is no facility to apply a horizontal load, it still possible to obtain approximate values for the stiffness coefficients of fluid-film bearings, in horizontal directions, as Morrison (1962) showed that

$$\begin{aligned} \alpha_{yy} &= y/\Delta F_y \approx \varepsilon\cos\phi/W = u_0/W \\ \alpha_{xy} &= x/\Delta F_y \approx -\varepsilon\sin\phi/W = -v_0/W \end{aligned} \tag{6}$$

where u_0 and v_0 are the journal static equilibrium position coordinates (see Figure 1). Parkins and Horner (1993) and Nikolakopoulos and Papadopoulos (1998) also used this approach to identify the four stiffness coefficients for tilting-pad journal bearings and electrorheological (ER) fluid-film journal bearings, respectively.

Myllerup et al. (1992) made use of the experimental static eccentricity locus (plot of eccentricity ratio, ε, versus the attitude angle, ϕ) and of the load–eccentricity ratio ($W/\mu\Omega$ versus ε) of the journal for fluid-film bearings. By using the locus differential method the following exact relationship for the stiffness coefficients can be obtained

$$\begin{aligned}
(c_r k_{xx}/W) &= (\mu\Omega/W)\frac{\partial}{\partial\varepsilon}\left(\frac{W}{\mu\Omega}\right)\cos\phi \\
(c_r k_{xy}/W) &= (\mu\Omega/W)\frac{\partial}{\partial\varepsilon}\left(\frac{W}{\mu\Omega}\right)\sin\phi \\
(c_r k_{yy}/W) &= (1/\varepsilon)\cos\phi - \left(\frac{\partial\phi}{\partial\varepsilon}\right)\sin\phi \\
(c_r k_{yx}/W) &= (1/\varepsilon)\sin\phi + \left(\frac{\partial\phi}{\partial\varepsilon}\right)\cos\phi
\end{aligned} \tag{7}$$

where c_r is the journal radial clearance, μ is the lubricant viscosity, Ω is the journal angular speed and W is the static vertical load per bearing. With this method it is difficult to ensure satisfactory numerical accuracy, especially at small eccentricity ratios, because the stiffness coefficients are calculated by differentiating the static equilibrium locus. This method does not yield damping characteristics of the bearing and it is restricted to an axisymmetric geometry, i.e. plain cylindrical bearing.

With the assumption of a short-width journal bearing, both the stiffness and damping coefficients can be obtained from the experimental static eccentricity locus using the closed-form relationships (Smith, 1969; with minor typological corrections) or in terms of the eccentricity ratio, ε, and the attitude angle, ϕ as given by Hamrock (1994). It is well known that this theory is a rather crude approximation to real bearings but the mathematical simplicity ensures its use in many models. Some clear evaluations of its weaknesses would be extremely useful.

In summary, results using the static load approach show great sensitivity to measurement errors. It is perhaps a little unfair to make comparisons as the errors in a number of other approaches are, as will be seen below, difficult to quantify.

5. Methods Using Dynamic Load

Most of the developments in the identification of dynamic bearing parameters that have taken place in the last 45 years have been accomplished using dynamic load methods, one of the advantages being that they may be readily implemented on a real machine. In all but a few cases to date, the rotor has been assumed to be rigid. The excitation can be applied either to the journal or to the bearing housing (preferably floating on the journal) depending on practical constraints. The basic form of the equations of motion for a rotor-bearing system will remain the same irrespective of the types of excitation force used to excite the system.

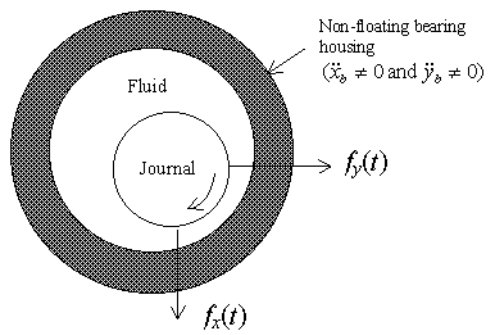

Figure 2. A non-floating bearing housing and a rotating journal floating on the fluid.

5.1. Forces Applied to Journal

For the case when the excitation is applied to the journal (Figure 2), the fluid-film dynamic equation, for the rigid rotor case, can be written as

$$\begin{bmatrix} m_{xx} & m_{xy} \\ m_{yx} & m_{yy} \end{bmatrix}\begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix}\begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} = \begin{Bmatrix} f_x - m(\ddot{x} + \ddot{x}_B) \\ f_y - m(\ddot{y} + \ddot{y}_B) \end{Bmatrix} \tag{8}$$

where m is the mass of the journal, x and y represent the motion of the journal center from its equilibrium position relative to the bearing center, and x_B and y_B are the components of the absolute displacement of the bearing center in vertical and horizontal directions, respectively. It is assumed here that the origin of the coordinate system is at the static equilibrium position, so that gravity does not appear explicitly in the equation of motion.

There will be one equation of this form for each of the bearings and the terms x_B, y_B represent the motion of the supporting structure. For the case of a rigid rotor with bearings on a rigid support, equation (8) expands to the form

$$\mathbf{M}_B\ddot{\mathbf{q}} + \mathbf{C}_B\dot{\mathbf{q}} + \mathbf{K}_B\mathbf{q} = \mathbf{f} - \mathbf{M}_R\ddot{\mathbf{q}} \tag{9}$$

where each of these matrices is now 4 × 4 and $\mathbf{q}$ is the generalized translation coordinate. The subscripts R and B refer to the rotor and bearings, respectively. If the rotor is flexible, this equation becomes

$$\mathbf{M}_B\ddot{\mathbf{q}} + \mathbf{C}_B\dot{\mathbf{q}} + \mathbf{K}_B\mathbf{q} = \mathbf{f} - \mathbf{M}_R\ddot{\mathbf{q}} - \mathbf{C}_R\dot{\mathbf{q}} - \mathbf{K}_R\mathbf{q}. \tag{10}$$

Collecting terms gives

$$(\mathbf{M}_B + \mathbf{M}_R)\ddot{\mathbf{q}} + (\mathbf{C}_B + \mathbf{C}_R)\dot{\mathbf{q}} + (\mathbf{K}_B + \mathbf{K}_R)\mathbf{q} = \mathbf{f}. \tag{11}$$

If the usual eight-parameter bearing model is used, then $\mathbf{M}_B = 0$. Of course, the matrices will now, in general, be larger than 4 × 4. This form was used by Arumugam et al. (1995) and Chen and Lee (1995, 1997) to extract K_B and C_B in terms of the known and measurable quantities such as the rotor model, forcing and corresponding response.

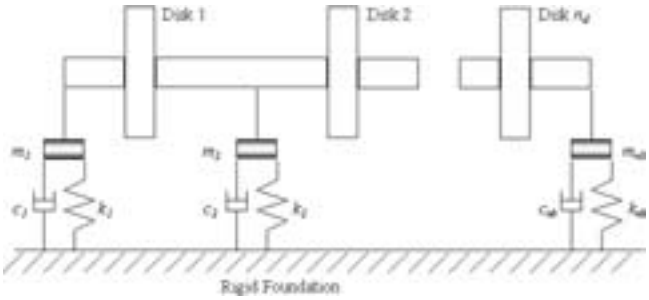

Figure 3. Schematic diagram of the flexible rotor-bearings system.

5.2. Forces Applied to Rotor

It is convenient at this point to form overall system mass, damping and stiffness matrices by adding the separate contributions of the bearings and rotor in equation (11). The modified form of this equation (11) is now applied to study the sinusoidal response of a rotor at speed Ω, then the response is of the form $\mathbf{q} = \mathbf{Q}e^{j\Omega t}$. The governing equation of motion is given by

$$[-\mathbf{M}\Omega^2 + j\Omega\mathbf{C} + \mathbf{K}]\mathbf{Q} = \mathbf{F}_u = [\mathbf{Z}(\Omega)]\mathbf{Q}. \qquad (12)$$

Using $[\mathbf{Z}(\Omega)]$ to denote the dynamic stiffness matrix and $\mathbf{F}_u$ as the unbalance force, Ω is the rotational frequency of the rotor. This equation may be conveniently partitioned between the internal rotor nodes and the bearing nodes. On the assumption that there is no unbalance excitation at the bearings, equation (12) becomes

$$\begin{bmatrix} \mathbf{Z}_{R,ii} & \mathbf{Z}_{R,ib} \\ \mathbf{Z}_{R,bi} & (\mathbf{Z}_{R,bb} + \mathbf{Z}_{B,bb}) \end{bmatrix} \begin{Bmatrix} \mathbf{Q}_{R,i} \\ \mathbf{Q}_{R,b} \end{Bmatrix} = \begin{Bmatrix} \mathbf{F}_u \\ 0 \end{Bmatrix} \qquad (13)$$

where the second subscripts, i and b, correspond to internal and connection degrees of freedom (DOFs), respectively. The DOFs of the rotor at the bearing locations are called connection DOFs, $\mathbf{Q}_{R,b}$, and the DOFs of the rotor other than at the bearing locations are called internal DOFs, $\mathbf{Q}_{R,i}$. It is assumed here that balance planes are present only at the rotor internal DOFs. This is an important case as it covers most operational machinery.

Tiwari et al. (2002) used equation (13) to extract the bearing parameters, $\mathbf{Z}_{B,bb}$, in terms of the known and measurable quantities. The basic form of the estimation equation is the same for both the rigid and flexible rotor cases. For the most general case, the bearing dynamic coefficients depend upon both the frequency of external excitation, ω, and the rotational frequency of the rotor, Ω. Effectively, there are four equations for the eight unknown bearing parameters (for example, for hydrodynamic bearings effective stiffness and damping coefficients and for squeeze-film bearings added-mass and damping coefficients). To estimate all eight dynamic coefficients, the response corresponding to at least two independent sets of forces (only in the magnitude and/or phase but not in the frequency, since bearing dynamic parameters could be external excitation frequency dependent) are required. Even for a multi-DOF rotor-bearing system (for example, the one shown diagrammatically in Figure 3), the response at each bearing location corresponding to at least two independent sets of force is required to find eight dynamic coefficients of all the bearings in the system (Tiwari et al., 2002). For the case when bearing dynamic coefficients are rotor rotational frequency dependent but not external excitation frequency dependent, all twelve dynamic coefficients (i.e. added-mass, damping and stiffness) can be estimated, in principle, by using force–response measurements at the minimum of three excitation frequencies.

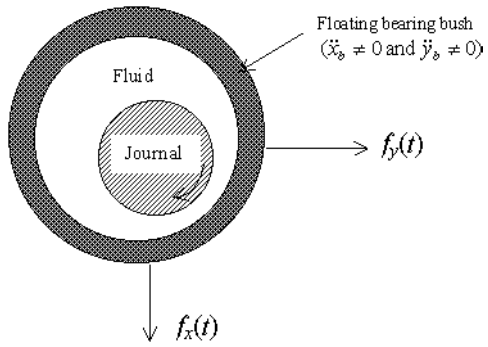

Figure 4. A fixed rotating shaft and a non-rotating bearing floating on the fluid.

6. Methods Using an Excited

In practical situations the application of a calibrated force to the journal can only rarely be applied. Glienicke (1966–67) adopted the technique of exciting the floating bearing bush (housing) sinusoidally in two mutually perpendicular directions (Figure 4) and in each case measuring the amplitude and phase of the resulting motions. Using these measurements, the stiffness and damping coefficients were calculated from the frequency-domain equations. Unfortunately, the author did not describe the procedure in detail.

Morton (1971) devised a measurement procedure for the estimation of the dynamic bearing characteristics by using the receptance coefficient method. He used very low forcing frequencies, ω, (10 and 15 Hz) to excite the lightweight floating bearing bush. By assuming the inertia force due the fluid film and bearing housing masses to be negligible and for sinusoidal motion, equation (8) may be written as

$$\begin{bmatrix} z_{xx} & z_{xy} \\ z_{yx} & z_{yy} \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} = \begin{Bmatrix} F_x \\ F_y \end{Bmatrix}$$

with (14)

$$z = k + j\omega\, c$$

where X and Y are complex displacements and F_x and F_y are complex forces in the vertical and horizontal directions, respectively. In equation (14) k now represents the effective bearing stiffness coefficient, since while estimating the bearing dynamic stiffness, z, the fluid-film added-mass and journal mass effects will contribute to the real part of the dynamic stiffness and the effective stiffness is estimated.

Parkins (1979, 1981, 1995) used a slightly different method by applying oscillatory vertical and horizontal forces to the journal and whose relative phase and magnitude may be

independently adjusted so that $x = \dot{x} = 0$ at all time t. Then equation (14) gives

$$\begin{aligned} k_{xy} &= f_x/y \quad \text{and } k_{yy} = f_y/y \quad \text{for } \dot{y} = 0, y \neq 0 \\ c_{xy} &= f_x/\dot{y} \quad \text{and } c_{yy} = f_y/\dot{y} \quad \text{for } \dot{y} \neq 0, y = 0. \end{aligned} \tag{15}$$

Similar relationships may be developed for $y = \dot{y} = 0$.

This approach eliminates the inversion of a dynamic stiffness matrix with no measurement of phase between journal displacement (or velocity) and applied force. It was concluded that for higher eccentricity ratios (greater than 0.78) the non-linearity in the measured coefficients non-linearly was found to be significant. Parkins (1981) expressed the bearing dynamic coefficients in terms of a first-order expansion as

$$k_{xx} = k_{xx_0} + \alpha_{xx}x, \quad c_{xx} = c_{xx_0} + \beta_{xx}y. \tag{16}$$

He proposed the *selected orbit technique* where the coefficients were obtained from special orbits with straight lines at the crossover point. Subsequently, in 1995 he extended the method so that a figure eight shaped orbit was required and the crossover point was utilized for estimation of bearing parameters. The orbit was obtained by the application of mutually perpendicular forces, whose relative magnitude and phase could be adjusted and whose exciting frequencies were Ω and 2Ω. Jing et al. (1998) described an on-line procedure for measuring the four damping coefficients of fluid-film journal bearings from imposed dynamic orbits of a figure of eight. A microcomputer was used to control the dynamic forces, to detect the figure of eight shaped orbit, to find the crossover point and, finally, to compute the damping coefficients. Brockwell et al. (1990) and Dmochowski and Brockwell (1995) measured the dynamic coefficients of a tilting pad bearing. The authors used a flexible rotor and sinusoidal loads were applied with phasing such that straight-line displacement responses were attained, a procedure similar to that of Parkins (1979, 1981). The uncertainty analysis of the derived bearing dynamic coefficients due to the possible error in the measurements was presented in detail. Experimental and theoretical dynamic coefficients were in moderate agreement.

Someya (1976), Hisa et al. (1980) and Sakakida et al. (1992) identified the dynamic coefficients of large-scale journal bearing by using simultaneous sinusoidal excitations on the bearing at two different frequencies and measuring the corresponding displacement responses. This is the called two-directional compound sinusoidal excitation method and all eight bearing dynamic coefficients could be obtained from a single test. When the journal is vibrating about the equilibrium position in a bearing, the dynamic component of the reaction force of the fluid film can be expressed by equation (8). If the excitation force and dynamic displacement are measured at two different excitation frequencies under the same static state of equilibrium then equation (8), ignoring the fluid-film added-mass effects, can be solved for the eight unknown coefficients as

$$\begin{bmatrix} X_1 & Y_1 & j\omega_1 X_1 & j\omega_1 Y_1 \\ X_2 & Y_2 & j\omega_2 X_2 & j\omega_2 Y_2 \end{bmatrix} \begin{Bmatrix} k_{xx} \\ k_{xy} \\ c_{xx} \\ c_{xy} \end{Bmatrix} = \begin{Bmatrix} F_{x_1} - m_B\omega_1^2 X_{B_1} \\ F_{x_2} - m_B\omega_2^2 X_{B_2} \end{Bmatrix}$$

$$\begin{bmatrix} X_1 & Y_1 & j\omega_1 X_1 & j\omega_1 Y_1 \\ X_2 & Y_2 & j\omega_2 X_2 & j\omega_2 Y_2 \end{bmatrix} \begin{Bmatrix} k_{yx} \\ k_{yy} \\ c_{yx} \\ c_{yy} \end{Bmatrix} = \begin{Bmatrix} F_{y_1} - m_B\omega_1^2 X_{B_1} \\ F_{y_2} - m_B\omega_2^2 X_{B_2} \end{Bmatrix} \tag{17}$$

where ω is the external excitation frequency and the subscripts 1 and 2 represent the measurements corresponding to two different excitation frequencies. Since equation (17) corresponds to eight real equations, on substituting the measured values of the complex quantities F_x, F_y, X, Y, X_B and Y_B, the bearing dynamic coefficients can be obtained.

For the case when excitation is applied to the journal, from equation (8), the estimation equation would be similar to equation (17) except the bearing housing mass and accelerations will be replaced by the journal mass and accelerations respectively (here it is assumed that when the bearing housing is fixed the added-mass effects are negligible). Diana et al. (1980) used an assigned force excitation system with hydraulic inertial exciters. The advantage of this was to change the excitation force frequencies independent of angular velocity of the journal. Equation (17) was used to estimate the stiffness and damping coefficients by a least-squares method and using the force and displacement measurements at more than two different excitation frequencies. They used measurements of pressures to obtain the dynamic forces transmitted by the fluid films.

Roberts et al. (1988) obtained the dynamic coefficients of the squeeze-film bearing using sinusoidal excitations and the corresponding frequency responses. They did not consider the cross-coupled terms and direct added-mass, damping and stiffness coefficients were obtained using a frequency fit, quadratic in the real part and linear in the imaginary part, of the obtained complex dynamic coefficients. Ellis et al. (1990) extended the state variable filter (SVF) method (Gawthrop, 1987) to the identification of twelve dynamic coefficients of squeeze film bearings. In this paper they used electromagnetic exciters in the vertical and horizontal directions to perturb the journal. With this arrangement they could achieve a wide variety of journal orbit motions within the clearance annulus between the journal and housing. Zhang et al. (1994) identified added-mass, damping and stiffness coefficients of a short cylindrical squeeze-film bearing executing circular centered orbits. Circular centered orbits were achieved using digitally generated external force signals and the test rig was same as used by Ellis et al. (1990). They used the fluid force model similar to equation (2). The identification of coefficients was performed by paired tests at close orbit frequencies and solving pairs of linear simultaneous equations. Roberts et al. (1993) used harmonic analysis to construct a simple parametric model for the non-linear fluid induced forces of squeeze-film bearings. The various parameters in this model were estimated from experimental data by using the SVF identification method. All the dominant terms in this model were shown to be related to the non-linear added-mass and damping coefficients in the theoretical hydrodynamic model developed by Zhang et al. (1993). Zhang and Roberts (1994) used the theoretical hydrodynamic model directly as a basis for constructing a fully parametric non-linear model by using the SVF identification method. Experimental results were presented for the radial motion. Subse-

quently, Zhang and Roberts (1996) used a frequency-domain method for estimating the unknown parameters in a non-linear squeeze-film force model. The differentiation of the data, which were used in SVF method, was completely avoided by applying window functions directly to various terms in the equations of motion. Experimental results were presented for both radial motions and circular centered orbits.

Matsumoto et al. (1990) identified stiffness and damping parameters of journal bearings with offset-halves. Two bi-directional exciters were used to excite the bearing simultaneously. Measurements were taken of the amplitude and phase and used to estimate the eight bearing coefficients simultaneously. A considerable amount of scatter was found in the estimated parameters. Rouch (1990) obtained direct stiffness and damping coefficients of squeeze-film bearings directly from frequency response function (FRFs) by using the dynamic magnification factor and resonance frequency. He presented results over a wide range of excitation frequencies and lubricant viscosities.

Flack et al. (1993) devised a hydrodynamic journal bearing test rig with dynamic measurement capabilities. To complement dynamic data, the operating eccentricity was measured as well as the details of the film operation, namely, the circumferential pressure distributions and film thickness profiles. The dynamic responses were generated using steady-state harmonic excitation to the bearing housing by a two-axis shaker system. An equation of the form (17) was used to estimate the bearing dynamic coefficients. The nominal values of the stiffness and damping measurements at a given test condition fell within the uncertainty bands, and usually within 3%. Typical uncertainties were less than 10% for stiffness and less than 30% for damping. Kostrzewsky et al. (1990, 1994, 1996, 1998) and Taylor et al. (1995) presented the dynamic coefficients of a plain two-axial groove and three-lobed journal bearing using the same test rig facility and the same estimation method as that of Flack et al. (1993). For a plain two-axial groove journal bearing the typical average uncertainty was found to be 12% for the dynamic coefficients. For the speed, load and temperature ranges tested, the dynamic coefficient results for each speed agreed within the uncertainty of the data. Subsequently, the effect of pad pivot friction (Wygant et al., 1999), the effect of load orientation (Pettinato and Flack, 2001) and effects of external excitation frequency and orbit magnitude (Pettinato et al., 2001) on the dynamic coefficients of the tilting pad journal bearing were investigated using a similar approach to that of Flack et al. (1993).

Heshmat and Ku (1994) and Ku (1994) identified damping and stiffness coefficients of a self-acting compliant foil journal and thrust bearings, respectively. For foil thrust bearings (non-rotating) the direct stiffness coefficient was obtained by the static load method and the direct damping coefficient (equivalent viscous damping; Thomson, 1981) was obtained by using the area of the measured dynamic load–defection curve (hysteresis loops) with respect to the static equilibrium positions, for a sinusoidal dynamic force. For foil journal bearings, the excitation was applied to the non-rotating journal at a single frequency by two exciters, one at a time. For each test condition, bearing static load, shaker input force, external excitation frequency, bearing housing reaction forces, journal accelerations, bearing housing displacements, and relative displacements between the journal and housing were measured. Equation (8) was used as the governing equation of the system with fixed bearings and no added-mass coefficients. Childs and Hale (1994) devised a rigid rotor facility to identify the dynamic coefficients of high-speed hydrostatic bearings, with shakers to provide sinusoidal excitation in two perpendicular directions to the stator (bearing housing). The coefficients were estimated from the FRFs by performing a least-squares fit using the identification procedure developed by Rouvas and Childs (1993). With the assumption that the bearing dynamic coefficients are independent of forcing frequency, the stiffness forces are independent of forcing frequency, the damping force is a function of forcing frequency and the inertia force is a function of the square of forcing frequency. By using a quadratic polynomial fit to the real part of the FRF and a linear fit to the imaginary part, the added-mass, damping and stiffness terms were identified. Using a similar approach, Marquette et al. (1997) identified eccentricity dependent bearing dynamic coefficients of annular seals and Kurtin et al. (1993), Franchek and Childs (1994), Franchek et al. (1995), Fayolle and Childs (1999) and Laurant and Childs (1999) presented rotordynamic coefficients of different types of hybrid bearings. Xu (1994) developed a bearing test rig for static and dynamic tests of fluid-film journal bearings (hydrostatic and hybrid). Micro pressure and displacement transducers contained in the shaft measured the distribution of fluid-film pressure and thickness and two electromagnetic exciters were located under and behind the test bearing to excite the test bearing in order to obtain its eight dynamic stiffness and damping coefficients. A misalignment jig was employed to align or misalign the test bearing with respect to the axis of the shaft. A considerable amount of deviation occurred in the estimated dynamic coefficients especially at high eccentricity ratios.

Arumugam et al. (1995) identified eight oil film coefficients for cylindrical and tilting pad bearings using FRFs from experiments and the finite element method. The rotor mass was excited by a uni-directional sine sweep perturbation force using a non-contact electromagnetic exciter and the displacement responses were measured at the bearing to estimate the experimental FRFs. From the equations of motion of the rotor-bearing system (equation (11)) the standard linear regression equation was developed. They showed that the number of FRFs needed for the estimation was only two per bearing for tilting pad bearings and four per bearing for cylindrical bearings. They observed scatter in the estimated bearing dynamic coefficients and the reason they gave for this was the excitation of both the forward and reverse modes simultaneously since the applied perturbation did not discriminate on direction of the applied force. Reddy et al. (1997) identified dynamic coefficients of four-pad tilting pad journal bearings using a similar approach. Subsequently, Arumugam et al. (1997a, 1997b) identified the stiffness and damping coefficients of cylindrical and three lobe journal bearings at different levels of bearing misalignment (twisting misalignment) using a similar approach. Tieu and Qiu (1996) and Qiu and Tieu (1996) obtained dynamic coefficients of two different freely alignable bearings. They assumed that the bearing dynamic coefficients are independent of the external excitation frequency. To avoid the effect of different test conditions on the estimated coefficients, two harmonic forces

at different frequencies were applied on the rigid rotor simultaneously and all the data were acquired during one test run.

Ha and Yang (1999) investigated the effects of external excitation frequency on the stiffness and damping coefficients of a five-pad tilting pad journal bearing. They used two orthogonally mounted hydraulic exciters to independently control the frequency of excitation. An estimation expression similar to equation (17) was used to estimate the bearing dynamic parameters. They concluded that the variation in external excitation frequency had little effect on both the stiffness and damping coefficients of the bearing.

6.1. Approaches for Increased Information Retrieval

Burrows and his co-workers developed several schemes in the time and frequency domains to identify the bearing dynamic coefficients. Burrows and Stanway (1977) used a pseudo-random binary sequence (PRBS) excitation to identify eight dynamic coefficients of a journal bearing. From a state-space representation of the system (without added-mass and fixed bearing housings), a multiple regression estimator was developed in the discrete time domain using a finite difference method. In addition, they used the covariance matrix associated with the estimator to generate confidence bounds for the estimates. However, this method suffered from biased and scattered estimates due to problems with the least-squares estimator, especially in the presence of noise. The same approach was applied to estimate the direct damping coefficients of a squeeze-film bearing by sampled observation of laboratory experiments (Stanway et al., 1979b), where the motion of the journal in the vertical and horizontal directions was assumed to be uncoupled.

By using the concept of structural identifiability (Bellman and Astrom, 1970), Stanway and Burrows (1978) showed that the necessary and sufficient conditions to estimate the eight fluid-film dynamic coefficients were (i) a single input function applied either vertically or horizontally, and capable of exciting all the natural frequencies and (ii) the measurement of the horizontal and vertical displacement responses resulting from the applied force. Stanway et al. (1979a) used a PRBS excitation to generate the FRF and extracted the direct damping coefficients of squeeze-film bearings by using the dynamic magnification (amplification) factor, Q (i.e. the ratio of the system gain at resonance to the gain at zero frequency; Thomson, 1981). The standard relationship between the dynamic magnification factor and the damping ratio, ζ, is given as $\zeta = 1/(2Q)$. Alternatively, Burrows et al. (1981) proposed the use of Schroeder-phased harmonic signals (SPHS; Schroeder, 1970) to identify the dynamic characteristics of the squeeze-film bearings and compared their results with the PRBS method. The basic attraction of SPHS is that they are synthesized from harmonic components and thus an arbitrarily specified signal spectrum may be generated with any desired frequency resolution. This flexibility in selecting the signal characteristics is desired to suppress the excitation of certain frequencies in a complex system. Using a digital computer, the SPHS can be generated with a flat spectrum and having sharp cut-off. Burrows and Stanway (1980) showed that time-domain estimation using a multi-frequency test signal and a least-squares estimator is conceptually superior to the single-frequency approaches, although in practice measurement noise and transformation from continuous to discrete time in the numerical process might cause large errors in the coefficient estimates. Using the concept of structural identifiability, they showed that the minimum test requirements to estimate a unique numerical value for each of the twelve bearing dynamic coefficients were as follows: (i) the oil film must be excited, first in the x direction (vertical) and then in the y direction (horizontal) using a multiple-frequency test signal which excites all the system modes; (ii) the displacement responses in the x and y directions must be measured; (iii) the corresponding force vectors must also be monitored. Moreover, they showed that when the excitation was applied only in a single direction then a maximum of ten unique bearing dynamic coefficients could be obtained from the experiments.

Dogan et al. (1980) also used PRBS excitation for the identification of the eight journal bearing dynamic coefficients. The bearing dynamic coefficients were obtained in the frequency domain by fitting the theoretical response function to the experimental results. In contrast, Burrows and Sahinkaya (1982a) used a similar excitation and obtained bearing coefficients directly from the test data in the frequency domain without recourse to curve-fitting techniques. They used the fast Fourier transform (FFT; Cooley and Tukey, 1965) of the responses and the procedure is similar to the time-domain technique described by Burrows and Stanway (1977). However, several of the sources of error in the earlier approach were avoided, as it was apparent from the comparison of the results with the previous results. They also assessed the relative merits of time-domain and frequency-domain methods and showed that frequency-domain techniques were less susceptible to noise. Burrows and Sahinkaya (1982b) extended the frequency-domain approach for the multi-DOF rotor-bearing system by using a lumped mass approach (i.e. rigid masses connected by massless elastic sections; Nikolajsen and Holmes, 1980).

Burrows et al. (1984) proposed an adaptive squeeze-film bearing, which was capable of switching between two levels of damping by means of adjusting the oil supply pressure to the squeeze-film bearings. They used a frequency-domain method to estimate the bearing damping coefficients. Burrows and Turkay (1982) analyzed the sensitivity of the amplitude and phase response characteristics with respect to the estimated coefficients that vary with the external excitation frequency. They concluded that a good fit between experimental and estimated frequency response characteristics at single excitation frequencies did not guarantee the quality of all of the estimated bearing coefficients. The variances of the bearing dynamic coefficients could be minimized by the design of the experimental procedure as well as by the data analysis. The choice of the suitable test signal was discussed by Burrows et al. (1981). Sahinkaya et al. (1984) discussed the use of the D-optimality concept (Kiefer and Wolfowitz, 1959) to optimize the measurement axes orientation to improve the quality of the estimates (i.e. minimize the variance of the bearing dynamic coefficients). Sahinkaya and Burrows (1984a) applied a Kalman filter together with a sequential least-squares estimator in the time domain to reduce the effect of noise on the estimates of four linearized damping coefficients of squeeze-film bearings. They used SPHS to excite the system and the corresponding displacements were measured. Mohammad and Burdess (1990) also used an extended Kalman filter approach to identify journal

bearing stiffness and damping coefficients. They found considerable scatter in the identified coefficients.

Lee and co-workers used the complex modal testing method extensively for the identification of rotor-bearing systems (Lee, 1991; Lee and Joh, 1993; Lee et al., 1996; Lee and Lee, 1997). Introducing the complex displacement, $p(t) = x(t) + jy(t)$, and force, $g(t) = f_x(t) + jf_y(t)$, the directional frequency response function (dFRF) between the complex input and complex output for an anisotropic rotor-bearing system (equation (11)), is defined as

$$P(j\omega) = H_{gp}(j\omega)G(j\omega) + H_{\bar{g}p}(j\omega)\overline{G}(j\omega) \tag{18}$$

where $P(j\omega)$, $G(j\omega)$ and $\overline{G}(j\omega)$ are the Fourier transforms of the complex signals $p(t)$, $g(t)$ and $\bar{g}(t)$, respectively, and the bar indicates the complex conjugate. In the above equation, $H_{gp}(j\omega)$ and $H_{gp}(j\omega)$ are referred to as the normal and reverse dFRFs, respectively. For an isotropic rotor system, the reverse dFRF vanishes. Unless $g(t)$ and $g(t)$ are fully coherent, the dFRF estimates of the complex input and output models in the frequency domain are

$$\begin{aligned} H_{gp}(j\omega) &= \frac{S_{gp}(j\omega)}{S_{gg}(j\omega)} \frac{1 - \dfrac{S_{\bar{g}p}(j\omega)S_{g\bar{g}}(j\omega)}{S_{gp}(j\omega)S_{\bar{g}\bar{g}}(j\omega)}}{1 - \gamma^2_{g\bar{g}}(j\omega)} \\ H_{\bar{g}p}(j\omega) &= \frac{S_{\bar{g}p}(j\omega)}{S_{\bar{g}\bar{g}}(j\omega)} \frac{1 - \dfrac{S_{gp}(j\omega)S_{\bar{g}g}(j\omega)}{S_{\bar{g}p}(j\omega)S_{gg}(j\omega)}}{1 - \gamma^2_{g\bar{g}}(j\omega)} \end{aligned} \tag{19}$$

where $S_{ik}(j\omega)$, $i, k = p, g, \bar{g}$, are the two-sided directional auto- and cross-spectral density functions (dPSDs and dCSDs) between p, g and $\bar{g}$, respectively, and $\gamma^2_{g\bar{g}}(j\omega)$ is the directional coherence function (dCOH) between g and $\bar{g}$, defined as

$$\gamma^2_{g\bar{g}}(j\omega) = \frac{|S_{g\bar{g}}(j\omega)|^2}{S_{gg}(j\omega)S_{\bar{g}\bar{g}}(j\omega)} \tag{20}$$

Lee et al. (1996) estimated the experimental dFRFs from rotor-bearing test rig responses corresponding to two independent random excitations. They used the measured dFRFs directly to identify the model of a magnetic bearing by defining a non-linear objective function and solving the developed non-linear regression problem by the Hooke–Jeeves pattern search method (Reklaitis et al., 1983). Complex modal testing allowed a physical insight into the backward and forward modes, and enabled the separation of those modes in the frequency domain. It also preserved the directivity information of the modes and thus effective identification was possible. In general, frequency-domain methods have proved superior in terms of data content.

Goodwin et al. (1997) used a test bearing, which was floating on a rigidly mounted rotating shaft due to hydrodynamic action. It was excited by a multi-frequency force signal (PRBS). Measurement of the resulting relative movement between the bearing and the journal enabled the fluid-film coefficients to be estimated. The main aim of the work was to investigate the effect on the dynamic characteristics of the opening and closing of an accumulator valve which was connected between a recess in the bearing surface to a standard commercial gas bag accumulator. The purpose of this arrangement was to reduce the dynamic stiffness of the bearing so as to improve the machine response. The bearing dynamic coefficients reduced by 15% and the stiffness coefficients alone reduced by 50%. Santos (1995, 1996) performed a similar investigation to adjust the bearing dynamic coefficients. By using a frequency sweep test, the bearing dynamic coefficients were estimated for tilting pad journal bearings. Considerable discrepancies were found between the theoretical and experimental coefficients especially for extremely high chamber pressure. Prabhu (1997) obtained, experimentally, the effect of misalignment on direct damping coefficients for cylindrical and three-lobed journal bearings. The damping coefficients were estimated from the FRF of the bearing setup using the half-power points (dynamic magnification factor method). They concluded that damping was much higher for the cylindrical bearings as compared to the three-lobed bearing, and misalignment affected the former much more than the latter. The damping coefficient increased with the misalignment.

7. Methods Using Unbalance Mass

From a practical point of view, the simplest method of excitation is to use an unbalance force because no sophisticated equipment is required for the excitation, and the rotational speed dependency of the bearing dynamic characteristics can be identified relatively easily. The disadvantage is, of course, that information is limited to the synchronous response. Nevertheless, since this is the predominant requirement, the application of forces due to unbalance is extremely useful. Hagg and Sankey (1956, 1958) were among the first to experimentally measure the oil-film elasticity and damping for the case of a full journal bearing with unbalance force only. They used the experimental measurement technique of Stone and Underwood (1947) in which the vibration amplitude and phase of the journal motion relative to the bearing housing were measured using the vibration diagram. They considered only the direct stiffness and damping coefficients along the principal directions (i.e. major and minor axes of the journal elliptical orbit).

The stiffness and damping coefficients may be obtained on the basis of the measured unbalance response whirl orbit. Since the cross-coupled coefficients are ignored, the results represent some form of effective rotor-bearing coefficients and not the true film coefficients. Duffin and Johnson (1966–67) identified bearing dynamic coefficients of large journal bearings using a similar approach to that of Hagg and Sankey. They proposed an iterative procedure to calculate all eight coefficients from the latter method. The method was similar to that of Hagg and Sankey (1956) except that cross-coupled stiffness and damping coefficients were also incorporated in the equations of motion.

Four equations can be written relating the measured values of displacement amplitude and phase X, Y, φ_x and φ_y, together with the known value of the unbalance, F, and four stiffness coefficients (obtained from static locus curve method; Mitchell et al., 1965–66) to obtain the four unknown damping coefficients. This allowed the solution of two sets of simultaneous equations having two equations in each set. The accuracy of the results was greater than the method (Glienecke, 1966–67) in which two sets of four simultaneous equations were used to obtain the stiffness and damping coefficients. The benefit was accentuated by the ill conditioning present in these latter equations, in which an error in

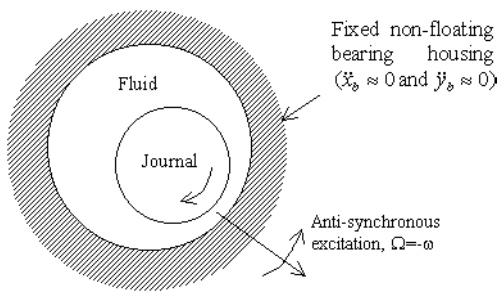

Figure 5. Anti-synchronous excitation by an auxiliary unbalance unit.

phase of 1° leads to a coefficient error of 5%. In this case, the coefficient error was reduced to about 2%. Yanabe et al. (1990) used a similar approach for the identification of dynamic coefficients of journal bearings; however, they measured the unknown unbalance (residual) forces by using force transducers between the bearing and housing.

Bannister (1972, 1976) used a rigid rotor with an applied non-synchronous rotating load applied to the bearing. In addition, the excitation force was capable of being applied to the journal in the same direction and also in the reverse direction to the journal rotation (see Figure 5), without influencing the speed or stopping the bearing. An additional feature was that the misaligning couple could be determined for varying amounts of misalignment. Elliptical motion between the bearing and shaft was measured. The eight linear coefficients and 28 non-linear coefficients were measured for a thin wall bush bearing.

Stanway (1983, 1984) used synchronous excitations due to unbalance to excite the rotor and the resulting synchronous unbalance responses under normal operating conditions were used for the identification of the four damping coefficients of a squeeze film. It was assumed that the four stiffness coefficients and unbalance information were known. From the state-space representation of the dynamic equation (8), with no added-mass effect and fixed bearings, a non-linear filtering technique introduced by Detchmendy and Sridhar (1966) was used to estimate damping coefficients from time-series records of the displacement responses corresponding to unbalance forcing. Numerical simulations were performed to validate the approach from the point of view of easy implementation to *in situ* testing of turbomachinery. However, they did not refine the method to estimate all the dynamic coefficients of the fluid-film journal bearings. Subsequently, Stanway et al. (1987) applied a time-domain method to estimate the damping coefficients of squeeze-film bearings in an experimental test rig. In a subsequent paper, Stanway et al. (1988) extended the time-domain method for non-linear identification of a squeeze-film bearing. They assumed that the damping forces were proportional to the *n*th power of the appropriate components of bearing-ring velocity and this led to the non-linear bearing model containing four coefficients and four exponents. The bearing model was simplified to contain six parameters to be estimated by taking the cross-damping terms as reciprocal. The unbalance synchronous forcing was achieved by using two exciters simultaneously with same frequency and with a 90° phase difference. Stanway et al. (1989) described an approach to identify the damping law of an ER fluid in vibration. The damping law was obtained by employing a non-linear, sequential filter to estimate the parameters associated with an *n*th-power velocity model of the damping mechanism. The feasibility of the approach was demonstrated in an experimental setup to identify the damping law of ER damper.

Sahinkaya and Burrows (1984b) presented a time-domain approach, which was related to their frequency-domain approach (Burrows and Sahinkaya, 1982a), for the estimation of fluid-film journal bearing dynamic coefficients from the rotor synchronous response. The method was illustrated by the estimation of the damping coefficients of a squeeze-film bearing and requires knowledge of the out-of-balance or the addition of a known trial mass. Moreover, this approach was ineffective when used with a flexible rotor. Burrows and Sahinkaya (1985) and Burrows et al. (1988b) presented the assessment of the time-domain (discrete time least-squares estimator and modified extended Kalman filter) and frequency-domain techniques based on the synchronous unbalance response. The oil-film force model developed by Burrows et al. (1986) was used for the simulation. They concluded that the frequency-domain method was able to produce better estimates for damping coefficients even if the displacement signals were contaminated by up to 10% white noise. They stressed the need to develop new methods to estimate bearing dynamic coefficients from operational data without introducing a test signal (i.e. without a calibrated external excitation to the system). Burrows et al. (1988a) modified the time-domain estimation algorithm, developed by Sahinkaya and Burrows (1984b) to estimate four damping coefficients, to estimate two direct damping and two direct added-mass coefficients from the synchronous response associated with the inherent shaft unbalance. They showed that added-mass coefficients might be significant and theoretical predictions were inadequate. Burrows et al. (1990) discussed the choice of a suitable model structure for better physical interpretation of experimentally derived bearing dynamic coefficients and highlighted the dangers of misinterpretation of the estimated parameters that could arise through the use of reduced-order models. They showed that different model structures could provide adequate characterization of the measured input/output relationship (i.e. identified models of different structure could predict observed responses).

Muszynska (1986,1998) and Muszynska and Bently (1990) advocated two perturbation techniques (calibrated displacement or force) for non-synchronous single-mode testing used for the identification of a fluid force model in rotor-bearing systems. The advantage of these methods was the rotational character of the input function (either displacement or force) in the forward or reverse direction (the same or opposite to rotor rotation). This allowed for independent identification of forward and backward modes of the rotating machine. By implementing a circular periodic frequency-swept force technique, the fluid force model was identified, experimentally, as a function of perturbation frequency. They used this identified fluid force model to predict the rotor stability and stability margin. Muszynska et al. (1993) outlined the use of a non-synchronous frequency-sweep (forward and backward rotating force perturbation) method for the identification of the dynamic stiffness characteristics of the low-order lateral modes in rotor-bearing-

seal systems. The application of non-synchronous perturbation (independent of the rotational speed of the rotor) could be applied by using an aerodynamic driven free spinner, auxiliary-driven unbalanced shafts connected to the machine rotor end through pivoting bearings and unbalanced or eccentric disks mounted on rotors through rolling element bearings and driven by an auxiliary motor. Two synchronized unidirectional electromagnetic exciters, placed perpendicular to each other, could also be used for excitation of the forward or backward rotating frequencies (Joh and Lee, 1993). However, such a perturbation applied to a rotating machine requires additional devices, which in the case of large industrial machines would be impractical.

Lee and Hong (1989) identified the bearing dynamic coefficients by using unbalance response measurements of rigid rotor systems supported by two anisotropic bearings. They identified four damping and four stiffness coefficients, utilizing unbalance response measurements from four sensors at two locations for two different trial unbalance conditions. They expressed the unbalance response as two synchronous vibrations, forward and backward; i.e.

$$\mathbf{q}(t) = \mathbf{Q}_f \, \mathrm{e}^{j\Omega t} + \mathbf{Q}_b \, \mathrm{e}^{-j\Omega t} \tag{21}$$

where $\mathbf{Q}_f$ and $\mathbf{Q}_b$ are the forward and backward whirl response vectors, respectively. Then the unbalance response can be written as

$$\begin{bmatrix} \mathbf{Z}_{ff} & \mathbf{Z}_{fb} \\ \mathbf{Z}_{bf} & \mathbf{Z}_{bb} \end{bmatrix} \begin{Bmatrix} \mathbf{Q}_f \\ \mathbf{Q}_b \end{Bmatrix} = \begin{Bmatrix} \mathbf{F}_u \\ 0 \end{Bmatrix} \tag{22}$$

where $\mathbf{Z}$ is the dynamic stiffness matrix, $\mathbf{F}_u$ is the unbalance force vector, and subscripts f and b refer to the forward and backward whirl, respectively. Inverting equation (22) gives

$$\begin{Bmatrix} \mathbf{Q}_f \\ \mathbf{Q}_b \end{Bmatrix} = \begin{bmatrix} \mathbf{R}_{ff} & \mathbf{R}_{fb} \\ \mathbf{R}_{bf} & \mathbf{R}_{bb} \end{bmatrix} \begin{Bmatrix} \mathbf{F}_u \\ 0 \end{Bmatrix} = \begin{bmatrix} R_{ff} \\ R_{bf} \end{bmatrix} \mathbf{F}_u \tag{23}$$

where $\mathbf{R}$ is the dynamic flexibility matrix. Information of the backward whirl response is lost when bearing exhibits the isotropic condition. Hence, isotropic bearings suffer from a lack of information necessary to identify uniquely all the bearing dynamic parameters; as was evident in the works of Stanway (1983) and Sahinkaya and Burrows (1984b). Tiwari et al. (2002) tackled the problem of isotropic bearings by using two methods: first, by using regularization (Tikhonov and Arsenin, 1977) with the condition of isotropic bearings and, secondly, by using the unbalance responses corresponding to rotation of the shaft in both directions (i.e. clockwise and anticlockwise). The second method ensures the information from the forward and backward whirl in the responses. For fluid-film bearings where the dynamic characteristics change for the reverse direction of rotation of the rotor an external auxiliary means of excitation described by Muszynska and Bently (1990) can be used. Hong and Lee (1992) and Hong et al. (1999) extended the method for flexible rotor-bearing system by using a model updating technique (Friswell and Mottershead, 1995).

Murphy and Wagner (1991) presented a method for the extraction of stiffness and damping coefficients for hydrostatic bearings using a synchronously orbiting intentionally eccentric journal as the sole source of excitation. The relative whirl orbits across the fluid film were made to be elliptic with asymmetric stiffness in the test bearing's supporting structure. This elliptic nature was utilized in the data reduction process. Their study considered the bearing coefficients to be skew-symmetric. Adams et al. (1992) and Sawicki et al. (1997) obtained twelve dynamic coefficients (stiffness, damping and added-mass) of hydrostatic and hybrid journal bearings, respectively, by using experimentally measured responses corresponding to at least three discrete orbital frequencies, for a given operating condition. They assumed that the bearing dynamic coefficients are independent of frequency of excitation. The estimation equation was similar to equation (17) except the rotor mass was ignored and fluid-film added-mass coefficients were considered. Dual piezoelectric/strain gage load/displacement measuring systems were used to give a confidence in the measurements. The difference between these two sets of dynamic force measurements was typically less than 2%. The test spindle (double-spool-shaft) had provision for a circular orbit motion of adjustable magnitude with independent control over spin speed, orbit frequency and whirl direction. The bearing dynamic coefficients were obtained by using a least-squares linear regression fit at all frequency data points over the tested frequency range. They observed that relatively small changes in the bearing dynamic coefficients occurred for backward versus forward whirling.

Tieu and Qiu (1994) estimated the linearized fluid-film parameters of two journal bearings from the out-of-balance responses where the rigid shaft was excited by a known unbalance force (synchronous excitation). The measurements were taken in the time domain and the FFT in conjunction with resampling were used to simplify the data acquisition process and minimize the effects of noise.

Recently, Tiwari et al. (2002) have developed a bearing parameter identification algorithm for multi-DOF rotor-bearing systems treating the shaft as flexible and having bearings with speed-dependent parameters. From the dynamic stiffness equation of a rotor-bearing system, a general algorithm was derived to extract bearing parameters. From a minimum of two run-downs with different unbalance configurations, speed-dependent bearing dynamic parameters were identified. A numerical simulation illustrated the algorithm and checked the robustness against measurement noise. For nearly isotropic bearings when the shaft orbit becomes nearly circular at the bearings, combinations of regularization and generalized Singular Value Decomposition techniques were used to solve the resulting ill-posed problem. For circular orbits, it was demonstrated that by measuring noisy bearing responses with the rotation of the rotor both in the clockwise and anticlockwise directions, the bearing estimation problem becomes well conditioned. The regularization algorithm was tested using an experimental rotor-bearing rig.

Since in practice the run-down/up synchronous responses from large machinery are readily available, the application of the methods described in this section to the run-down/up synchronous responses of such machines is the great hope for the future.

8. Methods Using an Impact Hammer

Until the early 1970s, the usual method to obtain the dynamic characteristics of systems was to use sinusoidal excitation. Downham and Woods (1971) proposed a technique using a pendulum hammer to apply an impulsive force to a machine structure. Although they were interested in vibration monitoring rather than the determination of bearing coefficients, their work is of interest because impulse testing was thought to be capable of exciting all the modes of a linear system. Due to the wide application of the FFT algorithm and the introduction of the hardware and software signal processor, the testing of dynamic characteristics by means of transient excitation is now common. Morton (1975a, 1975b) developed an estimation procedure for transient excitation by applying step function forcing to the rotor. With the help of a calibrated link of known breaking load, the sudden removal of the load on the rotor in the form of a step function (broad-band excitation in the frequency domain) was used to excite the system. The Fourier transform was used to calculate the FRFs in the frequency domain. He assumed the bearing dynamic parameters to be independent of the frequency of excitation. The analytical FRFs, which depend on the bearing dynamic coefficients, were fitted to the measured FRFs. He also included the influence of shaft deformation and shaft internal damping into the estimation of dynamic coefficients of bearings. The resulting estimated coefficients exhibited considerable experimental scatter. The cross-coupling terms were particularly poorly defined and Morton attributed this to ill conditioning of the receptance matrix but he did not pursue the point. The main limitation of his technique lies in the procedure required to determine the bearing dynamic coefficients from the resulting step response. The method has several other problems including the application of a true step function, internal shaft damping and shaft unbalance and the resulting dynamics. Chang and Zheng (1985) used a similar step-function transient excitation approach to identify the bearing coefficients and they used an exponential window to reduce the truncation error in the FFT due to a finite length forcing step function. They used equations of motion with negligible added-mass and fixed bearing housings and applied step-function forces in the horizontal and vertical directions one at a time, leading to two independent excitations. The estimation equations were obtained from the Fourier transform of the equations of motion for independent step-function forces. The experimentally measured force and response in the time domain were used to calculate the magnitude and phase angles of the four FRFs using the FFT.

Nordmann (1975) and Nordmann and Schöllhorn (1980) identified the stiffness and damping coefficients of journal bearings by modal testing (Ewins, 1984) by means of the impact method. An impact hammer excited a rigid rotor, running in journal bearings. Two independent impacts (i.e. first in the vertical direction and then in the horizontal direction) were applied to the rotor and the corresponding responses were measured. Input signals (forces) and output signals (displacements of the rotor) were transformed into the frequency domain and the four complex FRFs were calculated. They assumed the bearing dynamic parameters to be independent of the frequency of excitation. The analytical FRFs, which depend on the bearing dynamic coefficients, were fitted to the measured FRFs. Stiffness and damping coefficients were the results of an iterative fitting process.

Using a similar frequency-domain identification method, Nordmann and Massmann (1984) and Kim and Lee (1994) identified stiffness, damping and added-mass coefficients for annular and anti-swirl seals. Measurements were carried out on a test rig, and the system's response to impact forces was used to calculate complex FRFs (Nordmann, 1982). The analytical model, depending upon the seal parameters, was fitted to the measured data in order to find the dynamic coefficients. They used the least-squares method in conjunction with the instrumental variable method (Friswell and Mottershead, 1995) to fit the FRFs to estimate the bearing dynamic coefficients. They found considerable scatter in the identified parameters. Using the same test rig and experimentation, Fritzen and Seibold (1990) identified seal dynamic coefficients by using an extended Kalman filter in the time domain. They compared their results with different time-domain and frequency-domain identification methods (for example, least-squares, instrumental variable and quasi-linearization methods; Eyckhoff, 1974). However, in some cases poor convergence and even divergence was reported. Zhang et al. (1988) used a similar impact method, but with a different fitting procedure to reduce the computation time and the uncertainty due to phase measurement. They quantified the influence of measurement noise, the phase-measuring error and the instrumentation reading drift on the estimation of bearing dynamic coefficients.

Ramli et al. (1987) measured the dynamic coefficients for a squeeze-film damper using a technique similar to that of Morton (1971). A non-rotating shaft was used and thus no cross-coupling coefficients could be identified. Poor agreement was shown between experimental and predicted coefficients. Roberts et al. (1986) and Ellis et al. (1987) described a method to estimate the direct damping and added-mass coefficients of a squeeze-film bearing. They used an invariant imbedding algorithm to process the free-decay experimental displacement records. Ellis et al. (1988) demonstrated that the SVF of parametric identification could be used to estimate squeeze-film dynamic coefficients from experimental data. For the case of a centrally supported squeeze-film bearing executing low amplitude unidirectional, uncoupled motion, the system can be considered as a single-DOF system, similar to Roberts et al. (1986) and Ellis et al. (1987). It was shown that this method could be used effectively to process either transient (Ellis et al., 1988) or forced data (Ellis et al., 1989). Using this technique they estimated four direct added-mass and damping coefficients from experimental transient data. Roberts et al. (1990) extended the SVF method of parameter estimation to identify the eight fluid added-mass and damping coefficients of a squeeze-film bearing from transient time-domain experimental data.

Zhang et al. (1992a) obtained eight bearing dynamic coefficients by fitting the measured FRFs to those calculated theoretically. They also analyzed the influence of noise and measurement errors on the estimates, quantitatively, in order to improve the accuracy of estimated bearing dynamic coefficients. They tested their algorithm by simulation using a half-sinusoid impulse excitation and with a different level of noise added to the resulting response. To reduce the uncertainty due to noise in the response they averaged the frequency responses. To reduce the effect of phase-measurement errors,

they defined an error function using just the amplitude components of the FRFs. This non-linear objective function was then used to estimate the bearing parameters by an iterative procedure. They also demonstrated that when an impact test was used it was necessary to remove the unbalance response from the signal, especially at higher speeds of operation, and they concluded this to be the reason for the scatter in the results by impact excitation, as compared to the discrete frequency harmonic excitation. The companion paper (Zhang et al., 1992b) gave the details of an experimental investigation of the estimation method. A mechanical pendulum hammer was used to excite the rotor, which ensured that the lines of action of the excitation forces pass through the rotor center and that the excitation direction was consistent. Rouvas et al. (1992) identified bearing parameters using power spectral density methods (Rouvas and Childs, 1993). Using an impact test for two load cases, on a hydrostatic bearing setup, the complex FRFs were obtained as a function of excitation frequencies and were used to identify bearing dynamic parameters. They made use of the signal processing techniques of modal analysis for suppression of signal noise effects.

Marsh and Yantek (1997) devised an experimental setup to identify the bearing stiffness by applying known excitation forces (e.g. measured impact hammer blows) and measuring the resulting responses by accelerometers. They estimated the bearing stiffness of rolling element bearings (consisting of four recirculating ball bearing elements) of a precision machine tool using the FRFs. The tests were conducted on a specially designed test fixture (for the non-rotating bearing case). They stressed experimental issues such as the precise location of the input and output measurements, sensor calibration, and the number of measurements.

Qiu and Tieu (1997) obtained the dynamic coefficients of two journal bearings from impulse responses. They considered a rigid shaft mounted on the two journal bearings. The estimation procedure for bearing coefficients was similar to their previous work (Tieu and Qiu, 1994) but used transfer functions (ratios of the response to the force in the frequency domain) instead of the force and response separately. Since an impact force was used to excite the rotor, for the same speed of rotation more force–response information was available at different frequencies. They suggested using transfer functions at many frequencies near the shaft speed to increase the accuracy. They then compared the oil-whirl frequencies in the experimental setup with the values predicted using the estimated bearing coefficients and with linear stability theory. Jiang et al. (1997) identified dynamic characteristics of large journal bearings by using electromagnetic impulse excitation. They used a non-linear error function optimization algorithm to estimate the bearing dynamic coefficients. Considerable scatter was found (both in magnitude and sign) in most of the dynamic coefficient values except the direct stiffness terms.

Since impact tests have to be conducted for each rotor speed at which bearing dynamic parameters are desired, this method is time-consuming. In general, the governing equations for a bearing include coupling between the two perpendicular directions and this limits the amount of information that can be extracted from a single impulse test (Burrows and Stanway, 1977). Errors in the estimates will be greater for the case when bearing dynamic coefficients are functions of external excitation frequency as distinct from functions of rotor rotational frequency. Impulse testing may lead to underestimation of input forces when applied to a rotating shaft as a result of the generation of friction-related tangential force components (Tonnesen and Lund, 1988; Muszynska et al., 1993) and, further, is prone to poor signal-to-noise ratios because of the high crest factor.

9. Methods Using Unknown Excitation

In industrial machinery, the application of a calibrated force is difficult to apply. The presence of inherent forces in the system, due to residual unbalance, misalignment, rubbing between the rotor and stator, aerodynamic forces, oil whirl, oil whip and instability, render the assessment of the forcing impossible. Adams and Rashidi (1985) used bearing stiffness coefficients measured using the static loading method and measured orbital motion at an adjustable threshold speed to extract bearing damping coefficients by inverting the associated eigenproblem. The approach stems from the physical requirement for an exact internal energy balance between positive and negative damping influences at an instability threshold. This approach does not require the measurement of dynamic forces. The approach was illustrated by simulation.

Lee and Shih (1996) presented an estimation procedure for finding rotor parameters including bearing dynamic coefficients, shaft unbalance distribution and disk eccentricity in flexible rotors based on the transfer matrix method. The normal equations were formulated by using the relations between measured response data and the known system parameters. Then parameter estimation was performed using the least squares method by assuming that the bearing dynamic coefficients were constant at close spin speeds. The procedure was demonstrated by numerical simulations.

Tiwari and Vyas (1995, 1997a) offered a means of estimating the bearing non-linear stiffness without explicit force measurements, based on the analysis of random response signals measured at the bearing caps, provided that the system can be assumed to be perfectly balanced. The rotor-bearing system was modeled as a single-DOF model through the Fokker–Planck equation and the vibrations, resulting from random imperfections of the bearing surfaces and assembly, were processed through a curve-fitting algorithm to obtain the necessary bearing stiffness parameters. The method does not require an estimate of the excitation forces and works directly on the measured response signals of the system. The algorithm was illustrated on a laboratory rotor-bearing test rig and the results were compared with those obtained through an existing analytical model. Subsequently, the method was extended to the flexible rotor-bearing multi-DOF systems (Tiwari and Vyas, 1997a, 1998) and for the case of combined random and sinusoidal (residual unbalance) excitations (Tiwari and Vyas, 1997b). Based on a similar approach, Tiwari (2000) extended the method to the identification of the form of non-linearity (i.e. softening or hardening type) in the restoring (stiffness) force as well as its parameter estimation for the element bearings.

Odiari and Ewins (1992) used Volterra and Wiener based techniques for the identification of non-linear dynamic parameters of rotor-bearing systems. It was shown by simulation studies that the frequency-based approach produces

more accurate results than their time-domain equivalents. Also, the computation time is less for frequency-domain methods. Khan and Vyas (1999) extended the non-parametric Volterra kernel identification procedure to non-linear bearing stiffness estimation of a single-DOF rotor-bearing system. Subsequently, Khan and Vyas (2001a, 2001b) extended the identification procedure to a more general multi-DOF rotor-bearing system and the procedure was illustrated using a numerical simulation. Chatterjee and Vyas (2003) used the Volterra series and method of harmonic probing to experimentally estimate the non-linear bearing stiffness parameters of a rotor-bearing system supported by rolling element bearings.

10. Accuracy of the Estimated Bearing Coefficients

In many tests the measured and calculated response of rotors and machinery has been compared. Because the calculations are based on theoretical values for the bearing coefficients, they serve as an indirect check on the theory. There are indications, however, that the unbalance response of a rotor is often not very sensitive to deviations in the values of the bearing coefficients (Lund, 1979; Burrows and Turkay, 1982). Only a few works have been reported that evaluate the uncertainty of the measured bearing dynamic coefficients of the bearings, but now the trend is towards providing uncertainty and/or comparison with theoretical models. Bearing dynamic coefficients are not measured directly but are derived from calculations using a linear model of the test rig and measured response data. Woodcock and Holmes (1969–70) were among the first to discuss uncertainty due to numerical calculation (inversion of a matrix) due to error in the measurement of the phase angle. Hisa et al. (1980) considered uncertainty due to the measurement of excitation force, displacement and phase on the estimation of bearing dynamic coefficients. They concluded that the most important uncertainty in the measurements was the phase angle. Childs and Scharrer (1988) used a method described by Holman and Gajda (1984) to obtain uncertainty in the dynamic coefficients. Based on the available uncertainty in the force, external excitation frequency and displacement measurements, the resulting uncertainty for the stiffness and damping coefficients were obtained. They found that the measured cross-coupled damping results were rarely greater than the uncertainty. Kostrzewsky and Flack (1990a) developed different algorithms for obtaining and reducing experimental data. They used an average magnitude and phase method to relate uncertainties in the measurements of orbit data to the accuracy of the derived bearing dynamic coefficients. An uncertainty analysis, including the matrix inversion, was performed for the coefficients by applying uncertainties to the force and the response orbits and solving for the bearing dynamic parameter values. Kostrzewsky and Flack (1990b) presented several case studies on the predicted uncertainties of derived bearing coefficients. Using a test rig, a study was conducted on the effect of the shape, size and orientation of the imposed dynamic orbits on the coefficient uncertainties. Highly elliptic orbits with large angular differences between the major axes yielded low uncertainties. In general, results indicated that, because of the large number of measured quantities and limiting assumptions of some of the algorithms, considerable care is needed in choosing the most suitable algorithm to minimize the uncertainties. Roberts et al. (1990) and Murphy and Wagner (1991) performed an error analysis on the bearing dynamic coefficients. The uncertainties were obtained as a function of the ellipticity ratio (defined as the ratio of the major and minor axes of the elliptical displacement orbit). They found that the small change in the data becomes greatly magnified during the solution when the ellipticity ratio approaches one (i.e. the displacement orbit approaches a circle). Müller-Karger et al. (1997) obtained uncertainties in the dynamic coefficient identification due to the fluid-film non-linearity by using a simulation of the Reynolds equation for a three-lobe bearing. They demonstrated that an uncertainty in the coefficients of up to 20% was found as compared to the linearized coefficients from a small perturbation solution of the Reynolds equation which are commonly used in rotating machine design and analysis. This change in the uncertainty was due to fluid-film non-linearity. The theoretically obtained dynamic coefficients were compared with the results from the test rig and the procedure of Taylor et al. (1995).

Qiu and Tieu (1996) discussed various experimental uncertainties (for example, due to transducers, manufacturing tolerances, force application methods, level of vibration amplitudes and data processing) and the uncertainty in the estimated bearing coefficients were analyzed using a perturbation technique. The typical uncertainty of the estimated bearing dynamic coefficients was 32%. Arumugam et al. (1997a) obtained the uncertainty of the bearing dynamic coefficients using the method as discussed by Kostrzewsky and Flack (1990a). They considered uncertainty due to the measurements of displacement, force, phase and speed. They found that the uncertainties in the stiffness and damping coefficients were less than 7% and 10%, respectively. Sawicki et al. (1997) performed the uncertainty analysis based on the accuracy of the calibrated load cells, the resolution accuracy of the data acquisition system, the nominal sensitivity of the displacement probes and the accuracy of the speed readout. Kaneko et al. (1998, 2003) integrated the measured pressure distribution in the direction along the static load line and compared with the known static load. They found a relative difference of the order of less than 5%. They obtained the seal dynamic coefficients uncertainty in accordance with the ANSI/ASME Standard on Measurement Uncertainty (1986) and found an uncertainty of 14% in the stiffness coefficients, 18% in the direct damping, 25% in the cross-coupled damping and 35% in the added-mass coefficients. Fayolle and Childs (1999) obtained systematic uncertainties due to the accuracy of the measuring instrumentation and conform to the ANSI/ASME Standard on Measurement Uncertainty (1986) referenced by Coleman and Steele (1989). They also obtained deviations that arise from the numerical process used to identify the rotordynamic coefficients (i.e. curve fits of the FRFs). They found that the uncertainty was usually acceptable in comparison to the measured values, except for the added-mass coefficients. Ransom et al. (1999) estimated the uncertainty of the bearing dynamic coefficients based on precision uncertainty of the sensors, the uncertainty resulting from the analog to digital conversion and the variance introduced by the identification procedure.

There is a need to always incorporate the evaluation of the accuracy of the measured dynamic coefficients of the

bearings, both from experimental measurements and from theoretical calculations/modeling (for example, the uncertainty from the inversion of a matrix, non-linearity of the bearing dynamic parameters, etc.).

11. General Remarks and Future Directions

This paper has given a review of the identification procedures applied to bearing parameter estimation. The main emphasis has been to summarize the existing experimental techniques for acquiring measurement data from the rotor-bearing test rigs, theoretical procedures to extract the relevant bearing parameters and to estimate associated uncertainties. It is felt that both the experimental and theoretical aspects are important since most of the state-of-the-art procedures require matrix inversion and, if the experimental data acquired are not sufficiently independent, then ill conditioning may occur. The following are the outcomes of the present review.

11.1. General Remarks

(a) It is well established that current theoretical rotor models are sufficiently accurate to represent flexible rotor-bearing systems for the bearing dynamic parameter identification.
(b) In general, the coefficients derived from static load testing are extremely sensitive to measurement errors and the sensitivity of dynamically derived coefficients is more difficult to assess.
(c) Frequency-domain methods are preferable in terms of the quantity of data to be handled/stored. The signal-to-noise ratio is found to be better for frequency-domain methods.
(d) Unbalance response methods require measurements for both the forward and backward whirling of the rotor to ensure the availability of all of the modal information in the measurements. Provision of an external excitation force that is anti-synchronous (i.e. the excitation frequency, ω, equals the minus of the rotor speed, Ω) is sufficient.
(e) In almost all of the available papers, interactions between the bearings in rotor-bearing systems have not been taken into account.
(f) While using impact tests (multi-frequency tests) it is necessary to remove the unbalance response from the signal, especially at higher speeds of operation.
(g) In multi-frequency tests (either by using an exciter or impact hammer) it is assumed that the bearing dynamic coefficients are independent of the frequency of excitation.
(h) The error due to noise can be minimized by averaging the frequency response.
(i) There is a need to minimize the time required to capture the measured data for the *in situ* determination of the parameters, in order to reduce problems due to parameter drift.

11.2. Future Directions

(a) There is still a need for experimental work in the field of rotor dynamics to study the influence of bearings and supports upon the rotor response, in particular for full-scale rotor systems. New experiments should be devised and more effective use of the available data needs to be made, especially with the inherent practical constraints for measurements and development of new identification techniques.
(b) The validation of dynamic coefficients using data derived from actual machines in the actual operating environment is required.
(c) Synchronous unbalance response, which can easily be obtained from the run-down/up of large turbomachinery, should be exploited more for the identification of bearing dynamic parameters along with the estimation of unbalance.
(d) There is a need to develop new mathematical models of bearings (sometimes referred to as the problem of bearing identification) based on experiments to represent the behavior of rotor-bearing systems.
(e) There is a need to develop new effective non-linear bearing models and their identification (Garibaldi and Tomlinson, 1988; Stanway et al., 1988; Burrows et al., 1990; Tiwari, 2000; Chatterjee and Vyas, 2003) to predict non-linear phenomena (for example, jump and flutter phenomena). There needs to be further investigation of the application of advanced engineering methods, such as the Fokker–Planck equation (Dimentberg, 1988) and Volterra and Wiener based techniques (Schetzen, 1980), for the experimental identification of non-linear dynamic parameters of rotor-bearing systems.
(f) For maximum usefulness to analysts, the identified dynamic parameters of bearings should be documented in tabular form and equation coefficients for curve fits given along with the documentation of the operating conditions.
(g) There is a need for standardization of the data given in publications in the field of bearing dynamic parameter identification.
(h) From the present state-of-the-art methods of identification of dynamic bearing parameters, a comprehensive collection of data, similar that by Someya (1989), is expected soon by joint efforts of researchers in the field of bearing identification.
(i) There is a need for raw/processed measured data to be made available and exchanged among the researchers in the field.

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