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Expert condition monitoring on hydrostatic self-levitating bearings

Ramon Ferreiro Garcia [a,*], José Luis Calvo Rolle [b], Manuel Romero Gomez [c], Alberto DeMiguel Catoira [c]

[a] Ind. Eng. Dept., University of A Coruna, ETSNM, Paseo de Ronda 51, 15011 A Coruna, Spain
[b] Ind. Eng. Dept., University of A Coruna, EUP, Avda. 19 de Febreiro s/n, Ferrol, A Coruna, Spain
[c] Energy and Propulsion Dept., University of A Coruna, ETSNM, Paseo de Ronda 51, 15011 A Coruna, Spain

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ABSTRACT

Neural network based functional approximation techniques associated with rule based techniques are applied on the condition monitoring task of rotating machines equipped with hydrostatic self levitating bearings. Based on fluid online measured characteristic data, including pressures and temperature, the inherent hydraulic pumping system and the self levitating shaft is monitored and diagnosed applying vibration analysis carried out using virtual measurements. Required signals are achieved by conversion of measured data (fluid temperatures and pressures) into virtual data (vibration magnitudes) by means of neural network functional approximation techniques. Previous to the condition monitoring task (vibration analysis), a supervision task of the system behaviour is carried out in order to validate the information being processed. It is concluded that the vibration analysis based on the analysis of the dynamic behaviour of oil pressure (non accelerometer based signals) subjected to disturbances such as changes in oil operating conditions including viscosity, is successfully feasible.

1. Introduction to the hydrostatic bearings environment

1.1. The environment background

Computational fluid dynamics depends on fluid viscosity which is a function of the fluid temperature and pressure, a scenario in which an important group of analytic model based diagnostic tasks suffers from the required accuracy. Since viscosity can be assumed essentially as fluid friction, and consequently, like friction between moving solids, viscosity transforms kinetic energy of macroscopic motion into heat energy, while the temperature dependence of viscosity at isobaric conditions is well established (the viscosity increases with decreasing temperature), current discussion on the dependence of viscosity on external pressure (at isothermal conditions) shows a spectrum of controversial statements.

Assuming the number of degrees of freedom of the system is equal to 2 for analysis simplicity, and choosing temperature and pressure as the independent variables determining the properties of the system, in such cases, the dynamic viscosity μ can be considered as a function of pressure and temperature according to

$$\mu = \mu(p, T) \quad (1)$$

It is assumed, also, that the analysis is restricted to cases where the thermal expansion coefficient of the liquid is positive, (this property is fulfilled at atmospheric pressure for most but not all liquids), so that the following relations must be fulfilled

$$\left(\frac{\partial \mu}{\partial T}\right)_p < 0, \quad \left(\frac{\partial \mu}{\partial p}\right)_T > 0 \quad (2)$$

These relations imply that the viscosity must decrease with increasing temperature (for isobaric processes), and must increase with increasing pressure (at isothermal conditions). Moreover, considering the viscosity as a function of pressure and temperature, i.e., $\mu = \mu(p,T)$, the following identity can be written:

$$\left(\frac{\partial \mu}{\partial T}\right)_p \left(\frac{\partial T}{\partial p}\right)_\mu \left(\frac{\partial p}{\partial \mu}\right)_T = -1 \quad (3)$$

Eq. (3) follows from purely analytical considerations and does not involve originally any experimental physics. Taking into account the viscosity dependencies given by Eq. (2), it is concluded that the inequality

$$\left(\frac{\partial p}{\partial T}\right)_\mu > 0 \quad (4)$$

must be fulfilled. This equation described as a purely mathematical relation has a quite definite physical meaning. As mentioned earlier, the viscosity of liquids of constant composition can be considered as a function of two state variables, pressure and temperature, i.e., $\mu = \mu(p,T)$. However, if viscosity is considered as constant, then this

* Corresponding author. Tel.: +34 981 167000x4205.
E-mail addresses: ferreiro@udc.es (R.F. Garcia), jlcalvo@udc.es (J.L.C. Rolle), m.romero.gomez@udc.es (M.R. Gomez), alberto.demiguel@udc.es (A.DeMiguel Catoira).

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Nomenclature

p	fluid pressure (bar)	p_i	pressures (Pa)
P_1	supply pressure at pump discharge (bar)	z_i	elevation levels (m)
P_2	supply pressure before restriction (bar)	v_i	fluid velocity (m/s)
P_3	supply pressure after restriction (recess pressure) (bar)	ρ	density (kg/m^3)
P_4	supply pressure after recess (ambient pressure $\cong 0$) (bar)	g	acceleration gravity (m/s^2)
Q_R	mass flow rate at restrictor (kg/s)	ω	angular velocity (pump rotational speed) (rad/s)
Q_O	mass flow rate at film lands (kg/s)	D	pumps impeller diameter (m)
$\dot{m}$	pump mass flow rate at film lands (kg/s)	P_w	estimated hydraulic power (W)
Q	pump volumetric flow rate (m^3/s)	P_{wm}	measured hydraulic pump power (W)
T	fluid (oil) temperature (°C)	η	net pump efficiency
μ	fluid viscosity	Fex	external disturbance force (N)
κ_T	isothermal compressibility coefficient	ANN	artificial neural network
α_T	isobaric thermal expansion coefficient	NN	neural network
V	total fluid volume (m^3)	FFT	fast Fourier transform
H	pump actual head rise (m)		

relation gives dependence between pressure and temperature (at constant viscosity). So Eq. (3) means that in order for the viscosity to remain constant, an increase of temperature leads to effects which can be compensated by an increase of pressure. Furthermore, according to (Schmelzer, Zanotto, & Fokin, 2005) one can easily formulate a method to quantitatively estimate the pressure dependence of viscosity (in isothermal conditions) provided the temperature dependence of viscosity (at constant pressure) and some other purely thermodynamic characteristics of the liquid are known. Indeed, variations in viscosity can be connected with variations of free volume and suppose that the free volume is uniquely connected with the total volume of the system.

In Schmelzer et al. (2005) there are research results of experimental studies where the viscosity must increase in oils, as a rule, with increasing pressure. However, the opposite was experimentally observed for some materials such as glass-forming silicate liquids. The question in this way arises on how this behaviour can be explained analytically satisfying an accuracy criteria. An attempt to analytically approximate such function was carried out, concluding that, in most cases of interest, an increase of viscosity with increasing pressure must be expected, although exceptions are possible. Alternative theoretical approaches connect the decrease of viscosity with structural changes of the respective systems under pressure, which are not described appropriately by free volume concepts. Since for these complex systems, a decrease of the viscosity with pressure is observed as a rule, at least, for sufficiently high pressures, one has to check, first, whether a decrease of viscosity with increasing pressure exists, in contradiction with free volume theories and, second, how to incorporate such additional structural behaviour into a model independently of the particular mechanism of structural change considered. Due to the above mentioned reasons, and in order to achieve a manageable solution to the referred controversy on the effects of pressure and temperature on viscosity, it is highly interesting to look for an alternative model on the basis of experimental techniques in order to accurately represent the particular fluid dynamic's behaviour.

The paper is organised so that in Section 2 the studied test rig is described, in Section 3 an experimental model is developed on the basis of functional approximation techniques, the monitoring and supervision problem including an hybrid (model based-rule based) diagnostic strategy is described in Section 4, and, in Section 5, a discussion of the results and concluding remarks completes the paper.

2. Description of the test rig structure

2.1. Introduction

A scheme of the self-levitating bearing test rig, consisting of a generic hydrostatic bearing supporting a rotating shaft is presented in Fig. 1. A closed loop oil pump transfers pressurised oil to the hydrostatic bearing system, so that rotor levitation is not subjected to wear rate even at low or zero speed. As consequence of the pressurised oil film, it has the advantage that it can maintain complete separation of the sliding surfaces by means of high fluid pressure during the starting and stopping of shaft rotation. In order to keep the oil pressurised, hydrostatic bearings require the mentioned external hydraulic system to pump and circulate the oil which causes some temperature dependent energy losses involved in the circulation of the fluid (Ghosh & Majumdar, 1980; O'Donoghue, Rowe, & Hooke, 1969; Rowe, O'Donoghue, & Cameron, 1970).

2.2. The generic pumping system analytical model

The use of analytical models achieved by first principles conduct us to the use of analytical redundancy and hence to the data acquisition system validation. For this reason, a model based fluid transfer system is defined on the basis of first principles such as the energy equation, the affinity laws, and the power according the following generic expressions.

Energy equation:

$$H = \frac{p_1 - p_4}{\rho \cdot g} + z_2 - z_1 + \frac{v_2^2 - v_1^2}{2g} \tag{5}$$

Eq. (5) express that net head is equal to the static pressure difference plus the potential energy plus the kinetic energy.

Affinity laws:

$$\frac{Q_1}{Q_2} = \frac{\omega_1}{\omega_2}; \quad \frac{H_1}{H_2} = \frac{\omega_1^2}{\omega_2^2}; \quad \frac{Pw_1}{Pw_2} = \frac{\rho_1}{\rho_2}\frac{\omega_1^3}{\omega_2^3}, \tag{6}$$

$$\frac{Q_1}{Q_2} = \frac{\omega_1}{\omega_2}\left(\frac{D_1}{D_2}\right)^3; \quad \frac{H_1}{H_2} = \frac{\omega_1^2}{\omega_2^2}\left(\frac{D_1}{D_2}\right)^2; \quad \frac{Pw_1}{Pw_2} = \frac{\rho_1}{\rho_2}\frac{\omega_1^3}{\omega_2^3}\left(\frac{D_1}{D_2}\right)^5 \tag{7}$$

Transfer power:

Conventionally, when the differences in elevations and velocities are small, they can be considered negligible yielding