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# Experimental study of the start-up and dynamic characteristics of a double effect foil thrust bearing

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## Résumé :

*Cette étude porte sur la détermination expérimentale des caractéristiques complètes de fonctionnement d'un ensemble butée et contre-butée à feuilles. Cet ensemble est soumis successivement à des charges statiques et dynamiques. Sous charge statique sont obtenues les caractéristiques de démarrage, à savoir le couple maximum, le couple fluide et la vitesse de décollage en différents points de fonctionnement. Sous charge dynamique, appliquée par un pot vibrant électromagnétique, les raideurs et les amortissements sont étudiés. Une méthode d'identification des coefficients dynamiques est développée et validée.*

## Abstract:

*This paper deals with the experimental analysis of the start-up and dynamical characteristics of a double effect foil thrust bearing. For the start-up characteristics the maximum torque, fluid film torque and lift-off speed are measured at various operating conditions. The dynamic coefficients are obtained at various static loads after exciting the double effect foil thrust bearing with an electromagnetic shaker. An identification method of the coefficients is developed and validated.*

**Key words:** Foil thrust bearing, aerodynamic bearing, dynamic coefficients, start-up characteristics, vibration study.

## 1. Introduction

Today more than ever, energy efficiency is a major issue faced by the transportation industry. One common way of consuming less is to reduce the transported mass. To meet this criterion, diameters of turbomachines are diminishing and rotation speeds are increasing, in order to always have more power available. In these high speed turbopumps, classical solutions as ball bearings find their limitations and are replaced by aerodynamic components, such as foil journal bearings and foil thrust bearings (FTB). FTB are an evolution of rigid aerodynamic thrust bearings which lacks damping due to the very low viscosity of air. FTB are composed of a rigid plate on top of which is fixed a compliant foil structure, described in the next section of the paper. In this structure, Coulomb friction appears and dissipates energy, thus providing damping. Due to the high non-linearity of this phenomenon and to the complexity of the foil structure, numerical predictions lack of accuracy. Experimental analysis remains compulsory to the determination of the behavior of FTB, as well as to provide reference material for further numerical computing. A test rig was therefore designed and enables the determination of the start-up and the dynamic characteristics of various FTB. In this paper, a double effect foil thrust bearing (DEFTB) is studied. This choice represents a widely used architecture in turbomachinery. For the start-up measurements static loads and rotation speed are the only parameters taken into account. For a chosen operating point the dynamic coefficients are identified after the DEFTB was excited

thanks to an electromagnetic shaker. The dynamic coefficients (stiffness and damping) are not only dependent of the operation conditions (rotation speed and static load): they vary with the excitation frequency. It was found that the dynamic stiffness increases as excitation frequency and load increase, and that the equivalent viscous damping increases with the increase of the load but diminishes with the rise of the excitation frequency.

## 2. Experimental set-up

The test rig dedicated to the study of the FTB has two different configurations: one dedicated to start-up test and one to dynamical test. Both configurations are used in this paper and they are presented in FIG. 1. The start-up configuration was presented in [1] where a simple FTB was tested. Yet some modifications were made in order to adapt the DEFTB. This rig can apply rotation speeds between 10 and 40 *krpm* under static loads comprised between 0 and 70 *N* in both start-up and dynamic configurations. The test rig is composed of two vertical rotors. The upper part is non-rotating and supports the DEFTB and the sensors. It is guided by a long aerostatic journal bearing providing a high radial stiffness and no axial stiffness. The lower part is the rotating shaft, guided by three angular ceramic ball bearings and is entrained by a 2.3 *kW* spindle.

Details of the mounting of the DEFTB are depicted on FIG. 2. During start-up tests, three equally spaced non-contact displacement probes are attached to the holding bell of the DEFTB and the start-up torque is measured via a static force sensor bolted on the side of the holding bell. This sensor is pre-tensioned by four springs. The load is applied through a stinger and weights - not represented - fixed to the top of the upper rotor. During dynamical tests, three accelerometers are added to the displacement probes; the side static force sensor and its tensioning springs are removed.

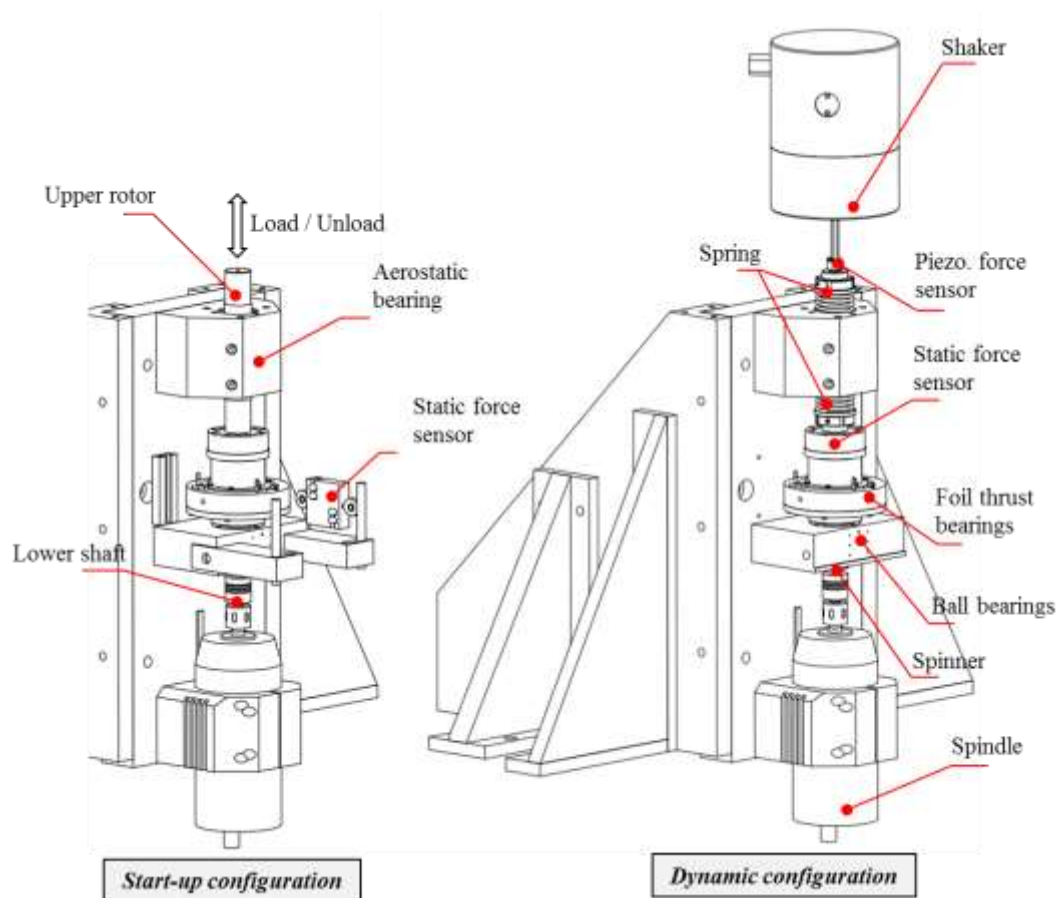


FIG. 1 - Perspective view of the test rig in dynamic and start-up configuration

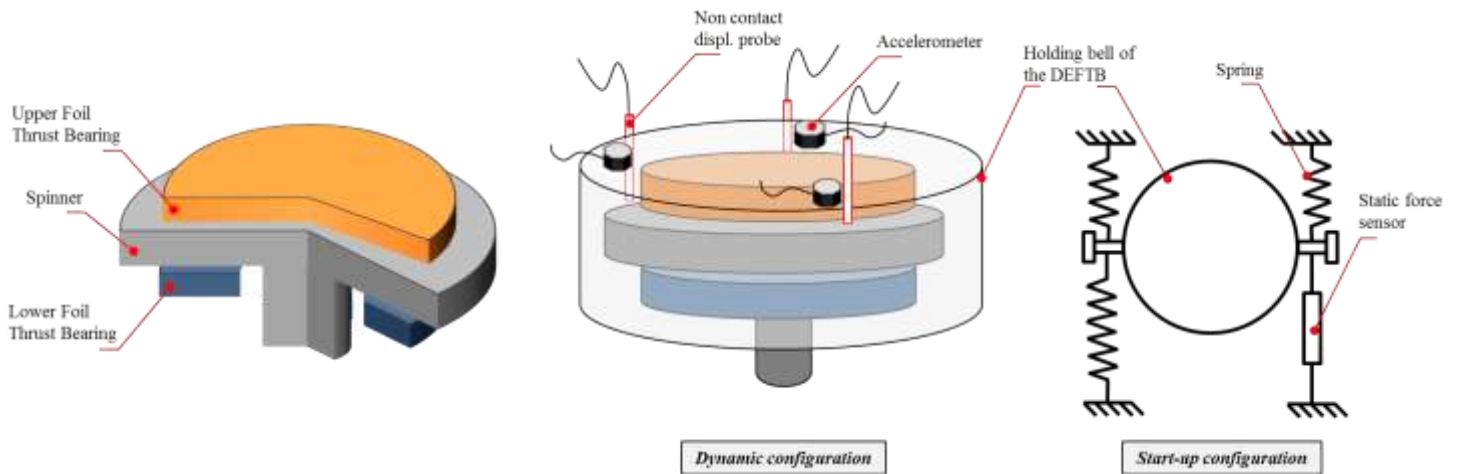


FIG. 2 - Mounting of the double effect foil thrust bearing, instrumentation for the dynamic and start-up experiments



FIG. 3 - Pictures and details of the compliant foil structure

The static load is applied by the spring under the long aerostatic bearing. It is constrained by a precision locknut. The dynamic load is applied by the electromagnetic shaker, bolted at the top of the upper rotor via a piezoelectric force sensor.

### 3. Start-up measurements

The spindle speed follows a trapezoidal command pattern: constant acceleration, constant speed and constant deceleration. The constant rotation speed and the load characterize the test, while accelerations and decelerations are kept constant for each experiment. In order to overcome the lack of power of the spindle - which was unable to overcome the friction torque of the DEFTB for large loads - the start-up was made for the unloaded DEFTB. The static load was applied once the DEFTB was fully airborne and the start-up torque was measured during the stop of the spindle under the effect of the friction torque. The assumption that the characteristics of the start-up and the shut-down are similar is justified in [1].

The results of the test for start-up torque, fluid-film torque and lift-off speeds are represented on FIG. 4. It can be seen on FIG. 4 that the speed has no influence over the start-up torque. The same conclusion can be stated from the fluid-film torque, where rotation speed seems to have little influence compared to the load. Nevertheless these graphs enable the determination of a maximum and minimum power loss of the DEFTB. The quite large standard deviations on the values of the lift-off speed denote that this parameter is more difficult to determine accurately.

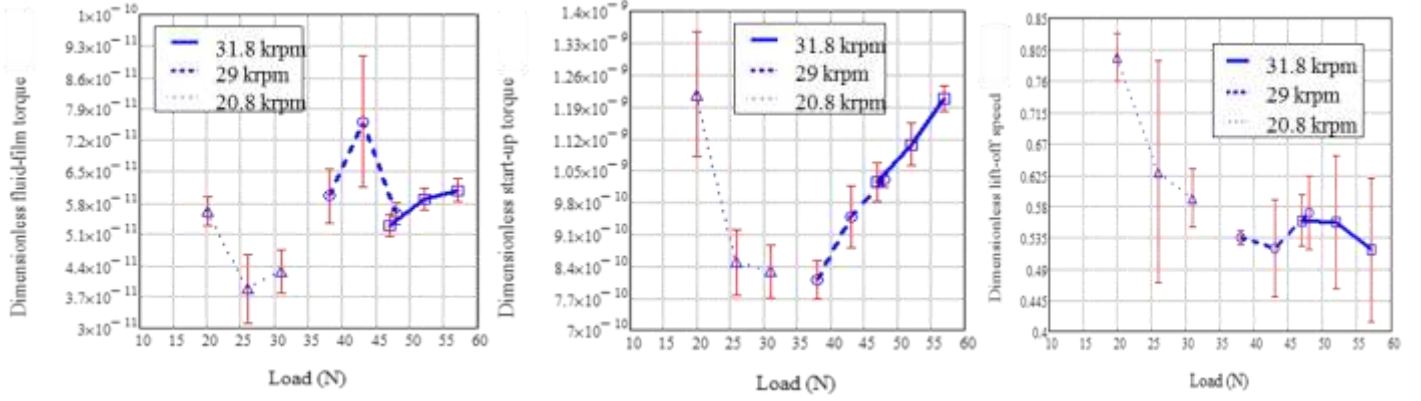


FIG. 4 - Dimensionless start-up torque and dimensionless fluid film torque for 20.8, 29 and 31.8 krpm under various loads.

#### 4. Dynamic measurements

It is necessary to apply as many frequencies as possible to the DEFTB. They can be applied either separately or as multi-frequency excitations. This second solution represents a large gain in time but is limited by the power of the shaker [2]. The force is a superposition of sinusoidal waves of different frequencies but of the same amplitudes. The command for  $N$  waves with frequencies between  $f_1 = n_1/T_{max}$  and  $f_2 = n_2/T_{max}$  is:

$$u = \sum_{n=n_1}^{n_2} a_n \cos\left(\frac{2\pi n t}{T_{max}} + \varphi_n\right) \quad (1)$$

With  $a_n = \sqrt{2/N}$  and  $\varphi_n = 2\pi \sum_{i=1}^n \frac{i}{N} + \varphi_0$ . With the limitation of the power of the shaker, a 5 Hz pitch will be used for frequencies between 150 ... 750 Hz, split in smaller intervals, to ensure a convenient dynamic displacement of the DEFTB. The results presented are averages of 100 records of 1 sec sampled at 8192 Hz.

The excitation force is applied on the top of the upper rotor. The different parts of the upper and lower shafts act as rigid bodies. The upper shaft represented on FIG. 1 is linked to the support via the two springs represented. The mass of this system is  $m_1$ . It is linked to the second mass  $m_2$  composed of the holding bell and the DEFTB via the static force sensor of stiffness  $k_c$ . At the bottom, the spinner supported by ball bearings is another rigid body of mass  $m_3$  and the stiffness of the bearings  $k_r$  add another degree of freedom. It can be seen that the dynamic properties of the DEFTB contained in the 3 degrees of freedom model (3DOF) depicted on FIG. 5. Hence it is compulsory to accurately identify the characteristics of components of the test rig (mass and stiffness) prior to obtaining the dynamic coefficients of the DEFTB. This determination was made prior to this study, and the different masses and stiffness are accurately known. The impedance of the DEFTB remains the only unknown. The equations of motion of the 3DOF system are:

$$\begin{cases} m_1 \ddot{x}_1 + k_r x_1 + k_c (x_1 - x_2) = f \cos(\omega t) \\ m_2 \ddot{x}_2 + k_c (x_2 - x_1) + k_b (x_2 - x_3) + c_b (\dot{x}_2 - \dot{x}_3) = 0 \\ m_3 \ddot{x}_3 + k_b (x_3 - x_2) + c_b (\dot{x}_3 - \dot{x}_2) + k_{rl} x_3 = 0 \end{cases} \quad (2)$$

An accelerometer is positioned on  $m_1$ .

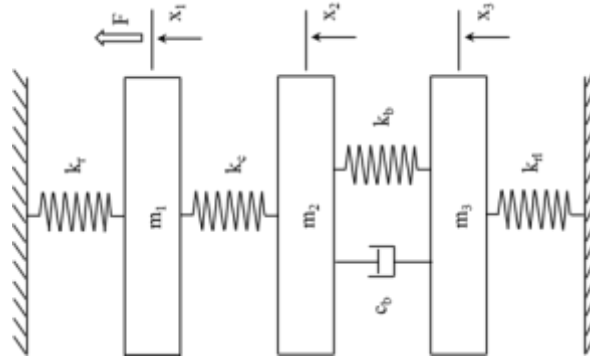


FIG. 5 - The 3DOF dynamic model of the test rig

Writing  $X_1 = -A_1/\omega^2$  and  $X_2 = -A_2/\omega^2$ , the second equation of (4) leads to the impedance of the DEFTB.

$$Z_b = -\frac{m_2 A_2}{X_{23}} + \frac{k_c(A_2 - A_1)}{\omega^2 X_{23}} \quad (3)$$

The results at various operating points are depicted in FIG. 6. It can be seen that both the load and the excitation frequency have an influence on the dynamic stiffness and equivalent viscous damping of the DEFTB. The main energy dissipation phenomenon is the dry friction occurring in the compliant foil structure, and the use of the structural damping loss factor enables to quantify it [3]. The air film has no influence over the dynamical characteristics [4]. The expression of the loss factor is:

$$\eta = \frac{c}{k} \omega \quad (4)$$

The variation of the loss coefficient versus the excitation frequency is drawn on FIG. 6. It can be seen that it decreases linearly with the excitation frequency. At the opposite, it increases with the increase of the static load. The maximum value of the loss factor, 0.9 corresponds to a 120 N load at 150 Hz excitation frequency.

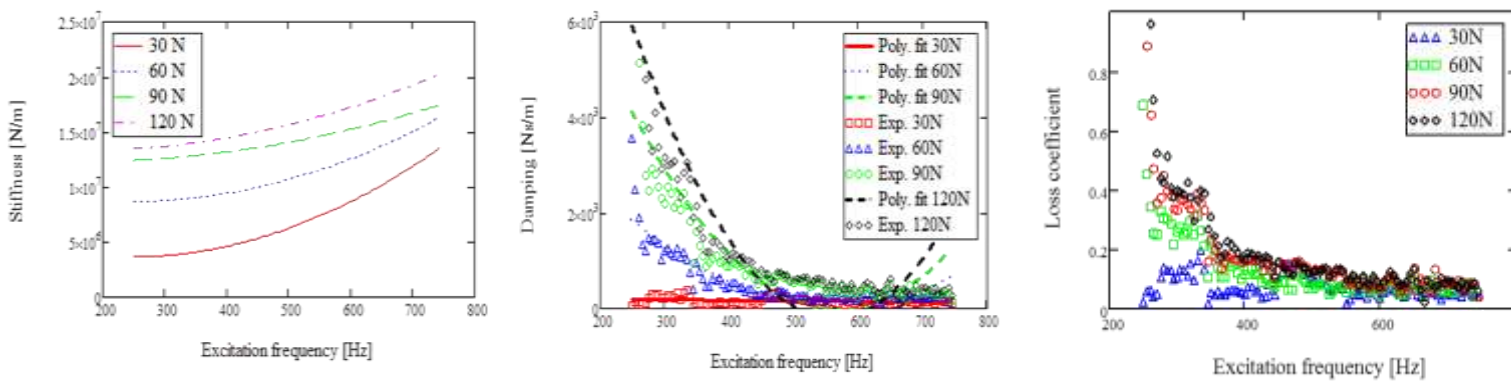


FIG. 6 - Dynamic stiffness, equivalent viscous damping and loss factor of the DEFTB for various loads and excitation frequencies

These values of loss coefficients are in agreement with what was recently found about foil journal bearings [5].

## Summary and conclusions

The start-up and dynamic characteristics of a double effect foil thrust bearing were investigated in this paper. The test rig and its two configurations were presented. Were studied the start-up torque, fluid film torque and lift-off speed at 9 operating points (3 different speeds and 3 different static loads). The dynamic stiffness and the equivalent viscous damping were measured for four different loads at excitation frequencies between 200 Hz and 800 Hz. They both demonstrated an increase with the excitation frequency. The air film was found to have no influence on the dynamic coefficients and on the energy dissipation of the DEFTB. The effect of dry friction on the energy dissipation of the component was estimated using the loss factor. The values of the loss factor compared favorably with the literature [5].

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