Controlling Non-linear Axial Vibrations of a Turbine Rotor

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1 Introduction

The rotor of a 3-stage axial turbine designed for an ORC plant, supported by two axial/radial lubricated roller bearings, has some axial clearance in between the outer bearing ring and the shoulder in the casing to allow for thermal expansion. The rotor can move freely within the clearance before the bearings on both sides are able to apply a restoring force. . The axial thrust on the shaft is balanced in design conditions, but in off design conditions some unsteadiness in the operating fluid flow could generate thrust fluctuations and consequently axial shaft vibrations. In order to reduce the severity of this possible source of vibration, noise and fatigue stresses, some damping should be introduced in the system. CFD calculations have allowed to characterize the damper, which is highly non- linear. The damper has been introduced in the model of the machine and its performance has been analysed by comparing the behaviour of the damped rotor to the un-damped rotor, at the different exciting frequencies. The comparison has been performed necessarily in the time domain due to the presence of two non- linearity: unsymmetrical damper and non-linear elastic restoring force. The results of the comparison have shown the efficiency of the damper especially in conditions close to resonance.

2 Rotor description

Fig. 1 shows a typical picture of an axial turbine with an overhung bladed wheel, axially connected to the condenser. The rotor is com-posed by a shaft supported by lubricated roller bearings carrying the overhung bladed wheel. Details of the shaft are shown in Fig. 2

Fig. 1. Sketch of the axial turbine

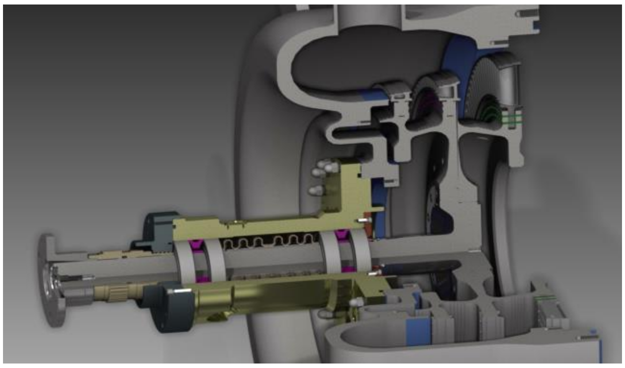

Fig.2. Detail of the shaft with bearings and power output flange

Each one of the two bearings is composed by a couple of angular contact ball bearings, mounted face to face, suitable for combined axial/radial loads. Two shoulders, one on the right hand side in the casing, the other in the opposite side obtained in the cover bolted to the casing, provide the axial stops. A clearance of c= 0.5 mm is left between the two stops, in order to allow for thermal expansion of the shaft. A preload on the spring placed between the inner rings of the two couples of bearings (shown in Fig. 2) is applied during assembling of the machine.

The trend of the restoring force F as function of the displacement d is shown in Fig. 3.

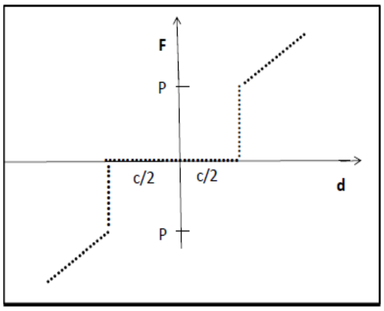

Fig.3. Restoring force F versus axial displacement d

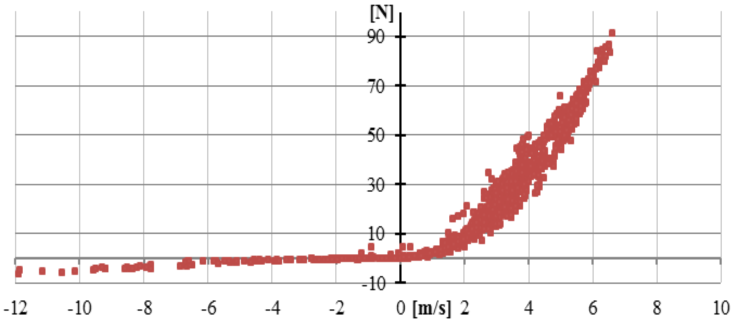

Fig. 4 Damping force as function of vibration velocity

In order to reduce possible undesired axial oscillations a unilateral squeeze film damper has been designed by means of CFD simulation. The squeeze film damper is basically composed by an annular chamber behind the left bearing ring, filled with oil. Oil flow is restricted by an auto-adaptive orifice. The device has been characterized for defining its model (Fig.4 shows the resulting damping force), and the model has been introduced in the mechanical model of the system. The resulting system behaviour has been compared to the behaviour without damper as shown in Fig. 5, for 3 different exciting frequencies.

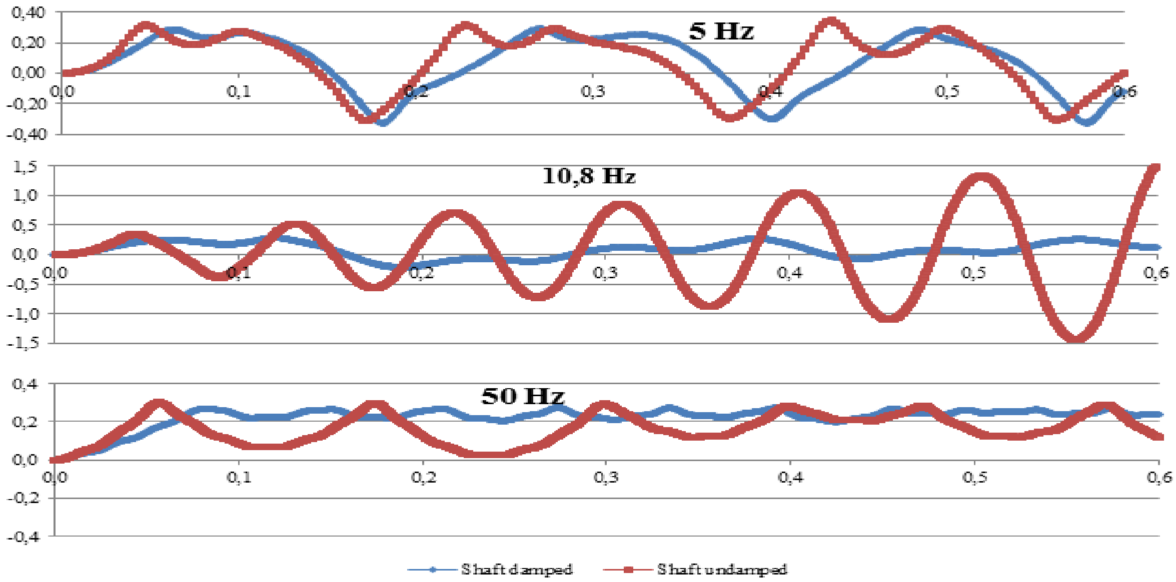

Fig. 5. Damped versus un-damped rotor vibrations at different exciting frequencies

The efficiency of the damper is proved in resonant conditions and at higher frequencies. At lower frequencies the unilateral damper reduces slightly the shock in one direction only.

The damper has been introduced in the machine, but excitation of axial vibrations for testing the device was not possible.