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Steam-whirl analysis in a high pressure cylinder of a turbo generator

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Abstract

In this paper steam-whirl instability occurrence and modelling is described. Some weak points in usual modelling and in standard stability criteria are discussed. The motivation of the present study is the case history of a steam turbine that experienced heavy steam-whirl instability though the calculated stability margin was sufficiently high in design conditions. During the electrical load rise of a power plant, the 425 MW steam turbo generator showed an unstable vibrational behaviour as soon as maximum output power was approached.

The combined effect of steam excitation (in bladed rows and in the steam glands) and of low damping in some of the oil film bearings was most likely the main cause of the observed malfunction. A model of the turbo generator has been set up, the steam-whirl exciting force coefficients and the oil film bearing coefficients have been applied and eigenfrequencies and damping factors have been calculated. In order to check the accuracy of this calculation also another method based on energy balance has been used but very similar values have been obtained, confirming the results of the standard stability evaluation approach. The calculation showed that the machine should have been stable, with a sufficient margin of stability, in design conditions. Therefore the steam-whirl excitation models have been analysed for identifying possible weak points which could justify the discrepancies between experimental behaviour and calculated results.

Keywords: Steam-whirl; Instability; Steam glands; Rotor dynamics

1. Introduction

Turbo-machinery like steam turbines and gas compressors may experience unstable vibrations due to interaction between the whirling shaft and the surrounding fluid forces in correspondence of seals where the clearance between rotating and stationary parts are small. These unstable vibrations arise in steam turbines most likely close to the maximum power condition, when pressures and consequent flows are maxima. The vibration amplitudes reach very quickly high unacceptable values, the machines are shut down due to excessive vibration levels and can then be operated only at reduced power levels. The unstable vibrations occur at the shaft first bending natural frequency and can be controlled only by the damping introduced by the oil

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film bearings, or by some additional external damping. When damping is insufficient, instability can be overcome only by modifying the seal geometry. Since the occurrence of this type of instability has limited the development of performances of turbo-machinery, many studies have been devoted in the last 45 years, depending on the development needs of high performance turbo-machinery. Several topics were considered: the analysis of steam-whirl in steam turbines, the instabilities excited by air flow in the axial compressors of gas turbines, the instabilities excited by the fluid in labyrinth seals of high performance centrifugal gas compressors and the instability excited by the liquid in the seals of high performance centrifugal pumps. The need of tight clearances for the development of high efficiency steam turbines has increased the risk of occurrence of steam-whirl, therefore higher accuracy in predicting instability is required.

Despite the fact that there is a consolidated knowledge about these problems, at least in the scientific community, there are still some weak points in modelling the steam-whirl excitation, so that predictions of stability margins sometimes can fail.

The motivation of the present study is exactly a case history of a steam turbine that experienced a heavy steam-whirl instability though the stability margin in design conditions, calculated by the turbine manufacturer according to a consolidated methodology, was sufficiently high. A twin turbine operated in the same conditions had no stability problems at all.

First, the main results of the studies about instability excited by fluid flow over seals or blade rows in turbo-machinery are recalled. Then the recent steam-whirl instability event on the turbine is described, the calculation results of standard modelling and stability margin evaluation are presented. Finally, some possible weak points in standard modelling are identified, which could justify the discrepancies between calculation results and experimental evidence.

2. Steam-whirl modelling

Unstable behaviour occurring in HP steam turbines was first analysed by Thomas [1] in Germany. He realized that the excitation was mainly due to the unequal leakage steam flow over blade rows. Krämer [2] used the model of Thomas for the blade row steam leakage forces, which excite forward whirl, and introduced the model of oil film bearings showing the combined effect on stability. He defined a linearized cross-coupling stiffness coefficient K_{xy} (force in x direction due to shaft displacement in y direction) proportional to the equivalent tangential driving force acting on the stage

$$K_{xy} = k\frac{N}{\Omega D_m l}, \tag{1}$$

where N is the power of the stage, D_m the mean diameter of the blade row, Ω the rotating speed, k a parameter which has to be calculated modelling the leakage flow over the blades (which depends obviously on the clearance) and l the length of the blade.

Reliable values of k are not given, but can be calculated with thermodynamic models of flow over blade rows. Eq. (1) can be explained as follows (see Fig. 1): the offset blade row (assumed offset equal to δ) and the consequent clearance difference makes pressure p_1 greater than pressure p_2, therefore the tangential driving force F_1 is greater than F_2 . The resultant force over the blade row F_y depends on offset δ in x direction and pushes the shaft in y direction, giving rise to the whirling motion. The cross-coupled stiffness is obtained dividing F_y by δ.

Considering the complete blade row, expression (1) is derived, where k/l takes account of the unequal pressure distribution resulting from unequal clearance and leakage over the blade tips. Due to symmetry the same cross-coupling coefficient K_{yx} is obviously obtained in vertical direction.

Values of k can be derived from diagrams in Thomas et al. [3]. These first experimental results, obtained with an offset rotating shaft and not with a whirling shaft, showed that the forces are not linear with the shaft displacement and are much higher for shrouded blades than for single standing blades, revealing the influence of the circumferential flow, which is also sometimes called the ‘‘gas bearing effect’’. This effect, which is present and well known in all hydraulic machinery, is obviously the only one present in the labyrinth seals where no blades are present (such as the balancing drum in HP steam turbines); it is mainly due to the rotation