# HIGH CYCLIC FATIGUE

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## 1. Introduction

A thermal or/and structural cyclic loading, which acts on one mechanical component, can lead towards mechanical failure of the whole machine. By a start-up and a shut-down of the turbine (see Figure 1), its mechanical parts are subjected to loadings, which undergo a continuous change from zero to the nominal value and again to zero, like a rotational speed which varies between standstill and its nominal velocity, as well as temperature and pressure that alter between the ambient (or below zero) and their elevated magnitudes corresponding to the designed service condition. These loadings have static nature and are characterised by the mean stress $\sigma_m$ (see Figure 1c), which can induce locally plastic deformations within the region of the stress concentration caused by different geometrical notches of the designed component. These cyclic static (mean) stresses above the yield limit of the material lead towards the low cyclic fatigue (LCF) determined by the number of start-ups and shut-downs. After the crack initiation, a remaining life of the cracked part is determined by using the method predicting the crack propagation in the material.

Under the nominal service condition the rotating blade is stimulated to vibration by partial arc admissions, non-symmetrical circumferential flow distribution, nozzle impulses caused by the stator vanes or flow pulsation caused by a random combustion process or an acoustic resonance. Usually the blade is excited by non-uniform circumferential distribution of the flow pressure in the turbine or compressor channel. By entering into and moving out of zones of different pressure, the rotating blade is excited to periodical vibration. This type of the excitation is called as a rotational harmonic excitation (engine order). Even for the blade designed to be free of resonance under the base load, during every start-up and shut-down, the blade has to pass the engine orders, which can be determined in the Campbell diagram. Figure 1a shows the blade vibration caused by an excitation, whereby Figure 1b illustrated the superposition of two resonances of the blade, whose possible resonance mode shapes are given in Figure 1d. This type of cyclic loadings causes high cyclic fatigue (HCF). If the blade is not resonance proof in its design process, a HCF fatigue life of $10^7$ cycles can be reached in minutes or hours in one LCF cyclic as it is presented in Figure 1. In praxis, the entire HCF consists the life up to the crack initiation because the initiated crack propagates usually very fast in the material in relation to the crack behaviour for the LCF.

A major goal in the development process of rotating turbomachinery turbine blades is to prevent them from high cyclic fatigue (HCF) failures, especially for the blades operating with variable rotational speed. To avoid failures of airfoils due to either flutter or resonance, freestanding blades often are connected circumferentially by different types of coupling elements. Vibrations of long blades with integrally machined shrouds or winglets (snubber, midspan shroud) can be reduced by *2 - 3* times with respect to the resonance response of the freestanding airfoil. In Figure 2 it is illustrated that resonance amplitudes of mode shape $f_1$ for the *0*-th nodal diameter, which refers to the steady-state response of the freestanding blade, is reduced by the shroud coupling to amplitude A for nodal diameters above *8*, for which the blades are effectively coupled by the shroud.

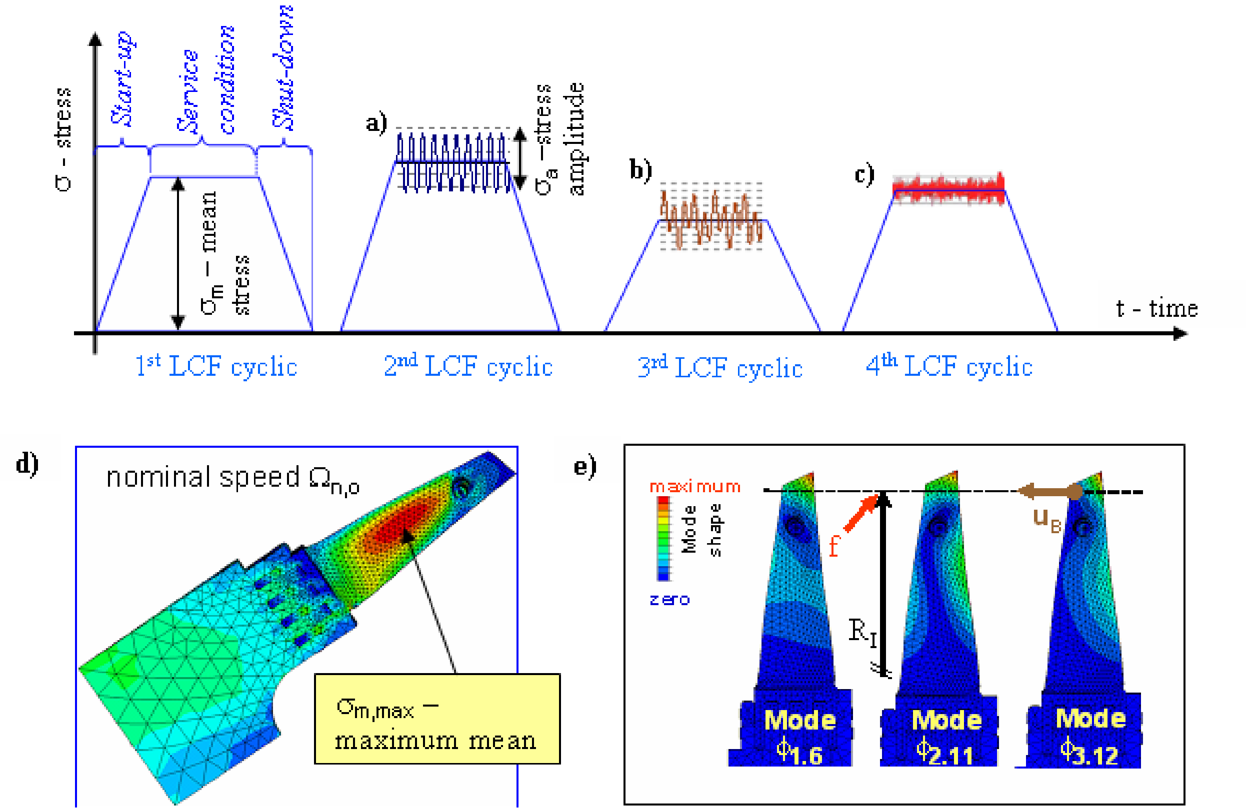


**Figure 1** **Operation schedule including start-ups and shut-downs of the turbine as its LCF loading with indicated vibrations under the service condition as HCF loading, where**
**a) vibrations due to one resonance excitation**
**b) vibrations as a superposition of two resonance excitations**
**c) stochastic vibration**
**d) mean centrifugal stresses in the steam turbine blade (Szwedowicz et al, 2006) and**
**e) its *3* lowest mode shapes excitable for engine orders *k* of *6*, *11* and *12***

In the design process, the reliable prediction of the resonance blade conditions and its dynamic stresses is difficult engineering task due to uncertainties with the assessment of damping magnitudes and excitation forces acting on vibrating blades. Within a HCF design, the maximal dynamic blade stresses are usually obtained from the correlation between the numerical modal (free vibration) results and dynamic strain measurements. These vibratory strains are transferred from the gauges either by slip rings or telemetry during engine tests with the blade prototype. In the late 90s, the Tip Timing technique becomes a very popular experimental approach, which allows measuring of the tip oscillations of all rotating freestanding blades in the turbine or compressor stage. This measurement delivers engineers valuable data about the individual dynamic behavior of each airfoil, which differs due to mistuning effects. In reality the bladed disc is a system of *N* blades, whose geometry slightly differs from each other due to manufacturing