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Influence of Rotor Unbalance Increasing on its Autobalancing Stability

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Abstract

The paper studies the patterns of change in the range of rotor autobalancing stability resulting from the static unbalance operational increases. It considers the rotor, which performs the plane motion. It is stated that during the process of rotor unbalance growth there is extreme increase and then decrease in the lower boundary of autobalancing stability range. At the same time there are the smallest stability margin and the highest level of residual vibration of rotating machine. The analytical formula used for the determination of the highest possible in-use autobalancing stability boundary is given. The qualitative difference in the behavior of both two-bodies and multi-bodies autobalancers is shown.

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Keywords: rotor; vibration; autobalancer; unbalance; autobalancing stability; stability boundary.

1. Introduction

Autobalancing devices (ABD) of passive type are used to reduce the vibration of rotating machines (see, for example, [1,2] and others). One of their advantages is the ability to automatically eliminate the rotor unbalance, which changes in the process of exploitation. According to the simplified engineering ABD theory for single-disc rotor performing a plane motion, the autobalancing motion mode is stable at all rotor speeds exceeding the critical speed [3,4]. Moreover in the stability range the ideal autobalancing mode is realized. At this mode the compensating bodies – spheres or pendulums – are in fixed positions relative to the disc and the rotor transverse oscillations are

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eliminated. From this it follows that autobalancing efficiency and its stability boundary don't depend on the value of the current rotor unbalance (within the ABD capacity).

In actual fact, at the autobalancing mode there is always some non-zero level of residual rotor vibration, which essentially depends on the current unbalance [5,6]. Also the real autobalancing stability boundary exceeds markedly the critical rotor speed [7,8]. As is known, there is a steady increase in the rotor unbalance during exploitation. For example, according to the research work [9,10] the rotor unbalance may increase in 5-30 times. That results in an increase of the residual rotor vibration and the lower boundary of autobalancing stability.

While autobalancer designing it is practically important to choose such its parameters which provide a sufficiently low level of the machine vibration for a specified period of exploitation [11,12]. An indirect indicator of vibration of rotor with ABD can be a margin of autobalancing stability as per rotational speed. In this regard, it seems topical to identify the relationship between autobalancing stability boundary and actual rotor unbalance. This problem is partially considered in the work [13]. However, this research was carried out only by numerical calculation method and there were no analytical formulas to determine the stability parameters.

The paper purpose is the analysis of influence of operational rotor unbalance increasing on its autobalancing stability and obtaining analytical formulas for the limit values of stability parameters.

2. The physical model. The dimensionless parameters

Let's consider the single-disc rotor which is based on two isotope supports. Statically unbalanced rotor disc is in the middle between the supports and performs plane motion. Autobalancer with compensating bodies in the shape of balls or pendulums is located in the plane of disk (Fig. 1). Direct contact between compensating bodies is missing.

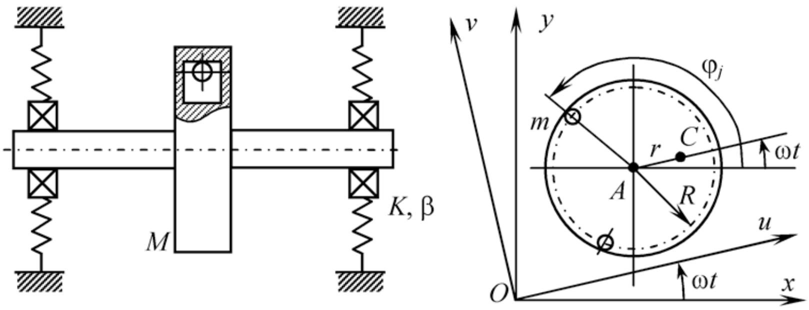

Fig. 1. Mechanical system "rotor – autobalancer".

This mechanical system is characterized by following physical parameters:
ω – angular speed of rotor rotation, radian /s;
M, r – mass of disc (kg) and rotor eccentricity (m);
$S = M\,r$ – rotor unbalance, kg·m;
K – stiffness of shaft and its supports, referred to the center of disc, N/m;
β – the coefficient of the rotor external viscous damping, s^{-1};
$p = \sqrt{K/(M+nm)}$ – the critical speed of rotor rotation, radian/s;
x, y – the actual coordinates of disc geometric center, m;
m, n, R – mass of the compensating body (kg), their number and radius of bodies movement circle in ABD (m);
β_0 – the coefficient of internal viscous resistance to the bodies motion in ABD, s^{-1};
α_j – constant angular positions of bodies relative to disc in the autobalancing mode, j=1,2...n, radian;
ψ_j – small angular deviation of the j-th body relative to α_j, radian;
$\phi_j = \omega t + \alpha_j$ – the actual angular coordinate of the j-th body relative to axis x, radian.

The analysis of the system dynamics can be reduced to the study of the equations that depend on the following dimensionless parameters [5,14]: