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INFLUENCE OF GEOMETRIC DEFECTS IN BEARING OUTER RACE ON VIBRATION GENERATION: AN ORIENTED STUDY FOR MANUFACTURING TOLERANCES SPECIFICATION AND ALLOCATION

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Abstract: *Deviations of real to ideal designed geometry of manufactured bearing components cause vibrations, and design tolerances must assure that geometry defects will not result in such undesirable effects. In this paper, a study about a ball bearing with angular contact is presented. Based on the relationship between irregularities in bearing outer race and generated vibrations, a relationship between different form parameters to be controlled during manufacturing is proposed. In this way, selection of the most appropriate control and manufacturing process for the part can be achieved, minimising cost and ensuring function.*
Key words: *Functional GD&T, analysis for function, analysis for manufacturing, bearing vibration, raceway waviness*

1. INTRODUCTION

To find out how geometric defects affect mechanism functionality it is very important to establish the types of defects to be controlled and their permissible values. The analysis must ensure part function, permitting maximum variability in manufacturing and thereby reducing costs.

This paper analyses the influence of waviness in bearing raceway outer ring on vibration generation, finding a relationship that allows direct control of the functional requirement "vibration" during manufacturing stage, rather than making indirect control through geometric tolerances with no distinction between magnitude and type of waviness. This analysis suggests that only waves of certain orders generate significant vibrations, thus being this one the geometric parameter to control, though usually done in an indirect way by specifications of roundness.

2. PROBLEM STATEMENT

A functional requirement to be considered during bearing design is the limitation of generated vibrations, in order to reduce noise and to avoid resonance effects by natural frequencies coupling of other mechanism components. Hence, control of clearances and geometric defects of contact surfaces is important. Usually, tight dimensional and geometric tolerances are used to ensure a better functionality. However, such restrictive tolerances often entail an indirect control that hides the real problem origin, though solving it. In order to reduce error magnitude, better machines and more controlled manufacturing processes will be required.

Vibrations have three main characteristics: frequency, wave length and wave amplitude. The objective of this work is to determine the necessary geometric functional constraints (GFC) in the outer race of a bearing with angular contact to fulfil the functional requirement of vibrations absence. With this purpose, dependence of generated vibrations magnitude and its frequency with eventual irregularities present in the bearing outer race are analysed. The result will allow establishing functional relationships or equations (FEq) between the functional requirement (FR) “vibration” (frequency and amplitude) and the geometric feature “profile defects” of the analysed element (Figure 1). The case of defects in the inner race is analysed in previous work (Serrano & Sancho, 2000).

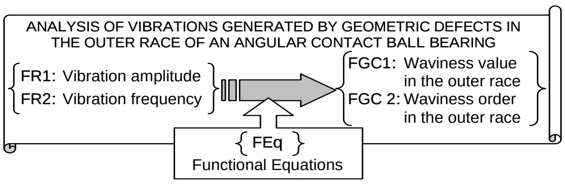

Fig. 1. Initial statement of problem

3. ANALYSIS OF RESULTING VIBRATIONS

In bearing components, waviness due to imperfect manufacture is an imperfection whose wavelength is much bigger that contact area width between balls and raceways (Hertzian contact width). Thus, it can be assumed that contact local deformation does not influence on waviness profile.

Waviness effect on resulting vibrations has been studied by many authors (Yhland, 1967; Harris & Kotzalas, 2007). In the work by Aktürk (Aktürk, 1999), vibrations produced by waviness in the inner and outer raceways and in the balls are studied, modelling the shaft-bearing set as a mass-spring system, where the shaft acts as a mass and the raceway and balls as massless non-linear springs. In this way, the system undergoes non-linear vibrations under dynamic conditions.

The relationship between the *i-th* ball (δ_i) and the Hertz contact force (W_i) is obtained according to $W_i=K\cdot\delta_i^{3/2}$, where K is the stiffness coefficient for the same material properties of the two contacting bodies $K=(K_i^{-2/3}+K_o^{-2/3})^{-2/3}$, where K_i and K_o are inner and outer raceways to ball contact stiffness respectively (Harris & Kotzalas, 2007; Aktürk, 1993).

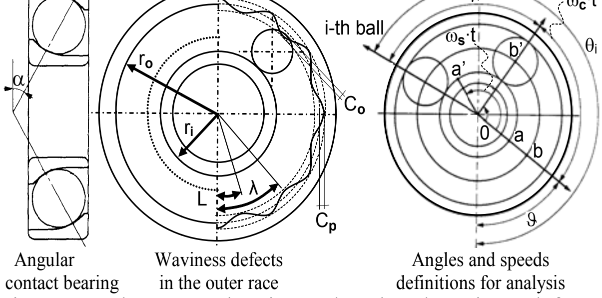

Fig. 2. Angular contact bearing and analysed waviness defects

Figure 2 illustrates the analysed ball bearing and a view of the defects in the outer raceway. It can also be seen that a sinusoidal waviness of amplitude C_p is considered and also a constant interference due to a preload of amplitude C_o. Assuming that the inner race moves at the speed of the shaft ω_i and the ball centre at the speed of the cage ω_c, the height of the waviness to consider as the interference between the *i-th* ball and the outer race can be expressed as a function of time $C_i=C_o+C_p\cdot sin[N\cdot(\vartheta+(\omega_c-\omega_i)\cdot t+\gamma i)]$, where ϑ is the angle between the ball number 0 and the reference axis, N is the number of waviness on the circumference, and γ is the angle between consecutive balls.

The analysis has been carried out for the system described in Table 1, assuming in the model that the shaft is perfectly rigid and uniform, and it is supported by two preloaded angular ball bearing (15º contact angle).

Inner ring bore diameter:	40 mm	Number of balls:	8 u.
Inner ring diameter:	46 mm	Unloaded contact angle:	15 º
Outer ring diameter:	62 mm	Pitch diameter of ball set:	54 mm
Inner ring groove radius:	4,1mm	Mass of the shaft:	550 N
Outer ring groove radius:	4,6mm	Preload each ball:	10 N
Ball diameter (d_b):	8 mm	Shaft rotating speed (rpm):	5000

Tab. 1. Data of the analysed system

Applying the movement equations to this system and solving them for this particular case, using the iterative Runge-Kutta method, results are showed in Figure 3, and it can be concluded that:

1) For vibrations with greater amplitude, the frequency depends on the waviness order according to the relations: $k=q\cdot m\pm p$ (waviness order) and $q\cdot m\cdot\omega_c$ (frequency for vibrations caused), where m is the number of bearing balls, and p and q are integers ≥1 and ≥0 respectively. However, vibrations of smaller amplitude appear also at other frequencies.
2) The most severe vibrations appear when Ball Passage Frequency (BPF) matches natural frequency of the system.
3) The most severe vibrations appear for a waviness order $k=i\cdot m\pm 1$, in the radial vibration case, and $k=i\cdot m$ in the axial vibration case. For some order waviness, the amplitudes of vibrations are negligible.

Previous results evidence that the greatest vibration amplitudes appear for those order waviness included between a preceding and following multiple of the balls number.

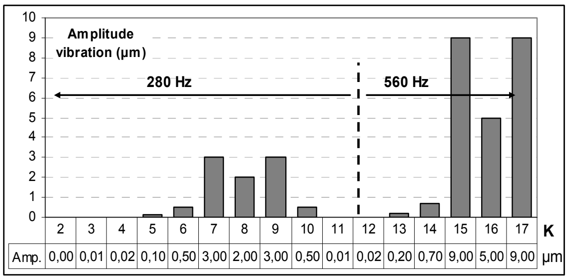

Fig. 3. Results of amplitude and frequency of vibrations

4. OBTAINING GEOMETRICAL CONSTRAINTS

From previously analysed results, functional geometrical constraints (FGC) and their relationships to functional requirements (FR), that is, functional equations (FEq), can be established. First, maximum possible information influencing geometrical characteristics is extracted and then how these, in turn, influence FR. It can be seen that:

1) Resulting vibration frequencies depend on the rotating speed and on the relationship between the balls number and the waviness order. Thus, a concordance relationship can be established.
2) Vibrations severity depends on the waviness order and, to a smaller extent, on the waviness amplitude.
3) It is important to avoid those waviness orders which are a preceding and following multiple of a balls number. For other combinations, vibrations are negligible. E.g., in a 6-8 balls bearing, common low order defects that lead to elliptical, triangular or square forms, produce very small amplitude vibrations.

It can be noticed that vibrations frequency and amplitude depend on the waviness magnitude and order. Therefore, the FGCs are: FGC1 (roundness error) and FGC2 (waviness order), existing three functional equations which relate their value with deformation and vibrations.

- FGC1: Roundness tolerance of the outer raceway. Its value is limited to allow a uniform running and to avoid balls blocking or excessive races and balls deformations, and the incomplete contact between race balls and races. The condition is $FGC1 \leq \#$, where # is a specific value.
- FGC2: Waviness types to avoid in the outer raceway. Waviness (periodical) that produces great amplitude vibrations should be avoided.
- FEq1: Relates the deformation of the races and of the balls contact area when a force is applied. It is the hertzian contact mentioned in section 3.
- FEq2: Relates the resulting vibration amplitude and the waviness order. The most severe vibrations appear for certain waviness orders. Therefore, it limits configurations which generate great amplitudes. The *FGC2* leads to $k \notin [\, i\cdot m-1,\, i\cdot m+1\,]$, where $i=1, 2, 3, \ldots$
- FEq3: Relates resulting vibration frequencies with the rotating speed and waviness orders. It establishes what frequencies are produced and they can be used, if necessary, to limit or avoid a specific frequency spectrum. The *FGC2* leads to $f=q\cdot m\cdot w_c/2\cdot\pi$.

5. CONCLUSIONS

From the analysis carried out, dependence between generated vibrations and type and magnitude of raceway imperfections has been obtained. Accordingly, geometric parameters to be controlled to limit undesirable vibrations have been deduced, namely: roundness error and waviness order. Typically, just a roundness tolerance is considered, thus being very restrictive to avoid, indirectly, all originated vibrations. However, these tolerances could be wider if waviness order would also be controlled to avoid really harmful frequencies.

These geometric parameters make it possible to take actions directly in the origin of incorrect function. Transference of the maximum functional information to the manufacturing stage allows process optimization with assurance of required function. In this way, the two maxims of Functional Geometric Dimensioning and Tolerancing (FGD&T) would be satisfied: every part meeting FGD&T can be used, and parts that can be used will not be rejected for not meeting FGD&T.

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