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STUDY OF A BALL-BURNISHING VIBRATION-ASSISTED TOOL

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ABSTRACT
This work first of all, talks about the characterization of vibrating plates of a ball-burnishing tool assisted by a vibration. This vibration helps to make easier the development of this finishing process, because it helps more easily deform the workpiece material. Optimum dimensions for the plates of the tool are obtained, as well as the mathematical model that characterizes operation.
Keywords: ball-burnishing, vibration, acousto-plasticity

1. INTRODUCTION
A good surface finish in one piece with a complex geometry is a difficult problem to treat. At present, many traditional processes are on study, to improve and adapt them to new difficulties found on industry. Furthermore, parallel searches provide novel solutions for lengthening the life of many components that undergo daily, for example, to high rates of wear or the effect of cyclical forces. The ball burnishing process could help to solve some of these needs.
Ball-burnishing, is defined as a technological operation consisting in plastically deform a surface irregularities, by the action of the force exerted by a ball [1]. This process could be performed with a conventional tool [2], or assisted by a vibration, as it is presented in this study. Assisting this process with a vibration, a greater movement of dislocations of material workpiece is achieved. This allows us to work harder materials with less force, and even improve the results of the process [3].
This paper aims to study the behavior of a vibration-assisted ball-burnishing tool. To do that first, the characterization via a physical model of the vibrating plates of the tool is done, because they are the components which transmit the vibrations to the workpiece. That is why they are the essential elements for the tool operation. Subsequently the model will be experimentally verified.
This process is not yet established in the industry nowadays. Its implementation could be of great importance, because it allows solve significant demands, such as obtaining parts with good mechanical properties and hard surfaces, using lower strength. Furthermore, the dislocations movement helps to introduce higher values of compressive residual stresses in the material deeper layers of the burnished parts. This would ensure an increase in fatigue life of many commercial parts like axles, dies, molds, etc. This last advantage would solve one of the major concerns of the today's mechanical manufacturing industry.
The use of vibration as a method of assisting the conventional manufacturing processes, is widely used in modern industry and it has been referenced by several authors [4, 5], as a method to improve the surface quality of the workpieces.

2. DEVELOPMENT STUDY

2.1. Vibration generator

As already mentioned, the vibrations are of wide application in mechanical manufacturing processes. The rationale for its use is the large number of dynamic forces generated due to the acceleration and deceleration at high frequencies, which are beneficial to the work surface.

In current industry vibration generator most widely used are the piezoelectric transducers, however, in the case of this study, the vibrations are generated by an electromagnetic transducer which is used to convert alternating current into a variable magnetic field. Those field produces an attractive cyclical force on the metal plates of a defined thickness, deforming them and causing them vibrating at a frequency determined by the magnetic field (figure 1).

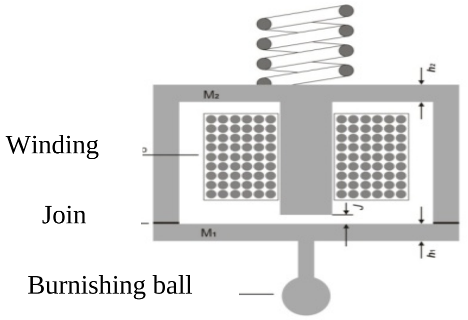


Figure 1. Functional diagram of the vibration system.

M_1- Plate attached to the burnishing ball
M_2- Plate attached to the spring
h_1- M_1 plate thickness
h_2- M_2 plate thickness
J- Gap

Starting from these premises, from the standpoint of design, it is important to monitor certain parameters of which will depend the correct functioning of the tool and which are those specified in the scheme of Figure 1. These elements of special interest are:

- The thicknesses of the plates M_1 y M_2,
- The interrelationship that is established between them,
- The gap *J*; defined as the distance between the coil core and the center of the plate M_1. Conceptually this distance must be greater than the maximum deflection suffered M_1 plate during operation of the tool.

Therefore, this paper seeks the maximum deflection of the plate M_1 from which to estimate the optimal value of the gap, there is a significant decrease in magnetic field strength, which is the one that manages to convey the vibration plates, and in turn to ball burnishing.

From here, it is useful to study the behavior of the deflection, when both plates coupled work (M_1 y M_2). Moreover, this deflection value will also depend on the thickness of the two plates. This is the reason that throughout the study evaluated various thicknesses to determine which one is right.

To characterize this tool makes the following assumptions:

1. The amplitude and frequency of the vibration generated directly dependent on the magnetic attraction force generated by the coil.
2. The deformation of the plates are in an elastic regime, therefore, the material properties such as Young's modulus and Poisson's ratio, are considered constant.
3. The tool should work in a resonant mode whose frequency must be estimated.

2.2. Modeling vibration generator system as a coupled plates

A plate resists transverse loads exclusively through flexion. The behavior that may have a flexure plate depends primarily on the relationship between the characteristic length *(D)* and thickness *(h)*. Plates with a reason $10<D/h<80$, are called thin plates and are characterized by having no flexural rigidity. Said thin plates in turn can be divided into two groups, depending on the reason, maximum deflection of the plate (w) over the thickness (h) [6]. Taking into account these classifications, the plate considered in this study is a thin plate rigid, because: the plate diameter D is 53mm, the plate thickness h is less than 5mm and the maximum deflection of the plate w is lower to 0,08 mm. With these values the aspect ratio D/h and the ratio w/h is reduced to:

$$\frac{D}{h} > 10 \quad \text{y} \quad \frac{w}{h} < 0{,}2 \qquad \text{...(1)}$$

Assuming that the deformations in solids to consider are infinitesimal, the relationship between the components of the stress and components of the strain are dependent on the material composing the solid. In the case of elastic and isotropic solid, the constitutive equations take the form of generalized Hooke's law [6].Therefore, assuming that the deformations in solids to consider are infinitesimal, the relationship between the components of the stress and the components of the strain, depend on the material composing the solid. In the case of an elastic and isotropic solid, the constitutive equations take the form of generalized Hooke's law [6]. The assumption of isotropic material is suitable to define the behavior of the steel C-45K (according to standard EN 10083-2) with will be manufactured the vibrating plates.

Deduction of equations considering all theories is obtained differential equations 2 and 3, which predict the deflection w of each of the circular plates, depending on the radius r and time t. These equations are the basis of the model to develop.

$$\frac{\partial^4 w}{\partial r^4}+\frac{2}{r}\frac{\partial^3 w}{\partial r^3}-\frac{1}{r^2}\frac{\partial^2 w}{\partial r^2}+\frac{1}{r^3}\frac{\partial w}{\partial r}-\frac{P}{K}=\frac{\rho}{K}\frac{\partial^2 w}{\partial t^2} \quad \text{...(2)}$$

$$\frac{\partial^4 w}{\partial r^4}-\frac{2}{r}\frac{\partial^3 w}{\partial r^3}+\frac{1}{r^2}\frac{\partial^2 w}{\partial r^2}-\frac{1}{r^3}\frac{\partial w}{\partial r}+\frac{P}{K}=\frac{\rho}{K}\frac{\partial^2 w}{\partial t^2} \quad \text{...(3)}$$

Where P is the vertical load distribution, and is defined by Equation 10. The value of the forcing function is constant.

$$P = P_0 \, sen \, 2\pi f \quad \text{...(4)}$$

Being P_0 is the initial value of the power system load, which is supposed to load and f, driving frequency of the coil. Furthermore, the flexural modulus of rigidity plate K, is as shown in equation 11. The boundary conditions are in (6)

$$K=\frac{Eh^3}{12(1-v^2)} \quad \text{...(5)}$$

$$w\left(r=\frac{D}{2}\right)=0 \qquad \frac{\partial w}{\partial r}\left(r=\frac{D}{2}\right)=0 \quad \text{...(6)}$$

2.3. Solution for the stationary problem

The problem under study is truly dynamic, forces involved change as you go through the process over time. For this case we can find the maximum deflection that can reach the tool plates for each of the conditions stated in the study, derived from the analysis of the static behavior. From here on, it is necessary to analyze the resulting vibration of the vibration coupling of the two plates. The analysis is performed for the following cases: M_1=1mm and M_2=1mm, M_1=1mm and M_2=2mm, and M_1=2mm and M_2=1mm.

It starts by considering that there is a forced vibration is introduced by a vibration generator in the tool. Is considered Rayleigh method. For this case we can get to estimate the resonance frequencies (first vibration mode). For this we have again used the software developed.

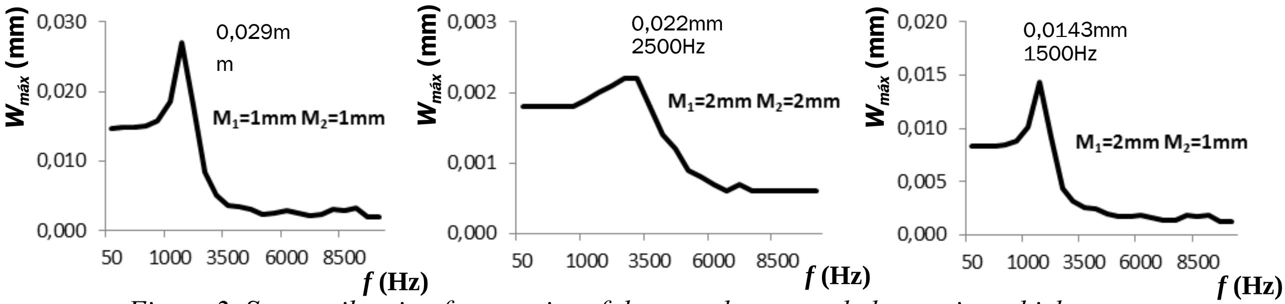
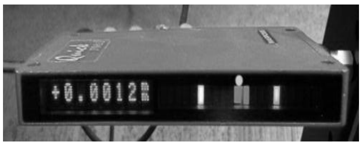


Figure 2. Sweep vibration frequencies of the two plates coupled to various thicknesses.

In figure 2 one can observe the resulting deflection values obtained when coupling the two vibration plates, for the first mode of vibration. This graphs shows the behavior of the relative deflection is greater in the case in which the two plates have a thickness of 1mm, as might be expected. When the plates have a thickness of 2mm, there is a smaller deflection compared to the first case. The behavior of the plates was analyzed for a frequency sweep between 50Hz and 10000Hz.

3. EXPERIMENTAL VERIFICATION OF THE DEFLECTION OF THE PLATE

To check the actual relative deflection, it has taken the plate M1, when the tool is subjected to the influence of the driver of the sinusoid. We used a digital electronic display unit displacement reading, manufactured by the MARPOSS. The sensor is placed in the central point of the tool shank, which is the maximum displacement area, to take the appropriate reading at the moment in which M_1 is vibrating at 2600Hz (figure 3). This frequency is taken from the results of simulation of the deflection of M_1=2mm.



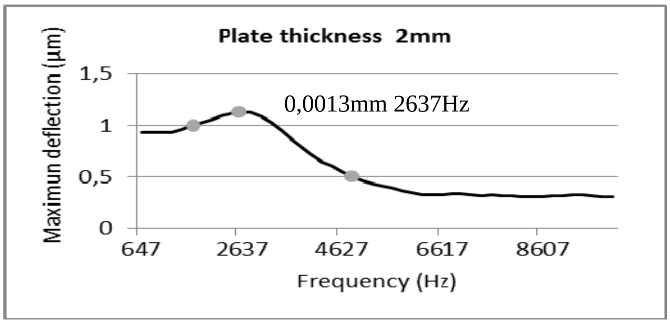


Figure 3. Sweeping the excitation frequency of the plate M1 sinusoidal, contrasted with the experimentally measured values to 2600Hz

It has been determined experimentally that the plate designed for the tool, under the terms of the theoretical study has a maximum relative deflection of 0,0012mm to 2600Hz, as seen in Figure 5. This value agrees with that shown in the graph, where the maximum deflection (0,0013mm) is obtained 2637Hz.

In order to obtain various experimental values, besides the deflection mode resonance measurements were taken at various frequencies. It was possible to obtain a reading of 0,0010mm to 2200Hz, and another from 0,0005mm to 4800Hz. Both results corroborate the behavior of the curve obtained in the simulation. From the results of manufacturing a prototype tool deciding the thickness of two plates (M_1 and M_2), is 2mm.

4. CONCLUSIONS

Has been successfully developed to characterize the plates involved in the design of a tool to be used in the ball-burnishing process assisted by a vibration.

There have been reliable numerical solutions to predict the values of deflection plates and suffering from them to optimize the different basic design parameters: the thickness of the plates, the join and the working frequency of the vibration generator.

It has manufacturing and set up a prototype tool and the first experimental results show improvements compared to conventional burnishing process on surface roughness parameters evaluated.

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