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Análise da influência de descontinuidades em elementos mecânicos no desbalanceamento

Analysis of the influence of mechanical element discontinuities on unbalancing

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Resumo
O estudo do desbalanceamento em rotores é essencial para a segurança e confiabilidade no projeto de máquinas e outras estruturas mecânicas que possuem partes rotativas. Ao compreender o comportamento dinâmico e os efeitos de peças desbalanceadas, os engenheiros podem corrigir esse problema e estabelecer um programa de manutenção para proteger os sistemas mecânicos das vibrações excessivas e prevenir, desta forma, das falhas catastróficas, principalmente devido à fadiga. O balanceamento em um e dois planos é o procedimento mais comum para corrigir massas desequilibradas em peças rotativas. Neste trabalho, pretende-se avançar na compreensão dos efeitos da distribuição de massa em peças de seção irregular. Em certas situações, há limitação local para proceder às correções, isto é, acrescentar uma massa de correção, por exemplo, devida à irregularidade e as restrições do elemento de máquina. Para validar a solução proposta, foi utilizado uma amostra de um virabrequim e de um comando de válvulas, onde o desbalanceamento é um problema comum, face às descontinuidades geométricas presentes, que são próprias desse tipo de peça. O resultado alcançado demonstrou a validade dos sistemas aplicados que permitiu o balanceamento das amostras dentro dos níveis recomendados pela norma.
Palavras-chave: Vibração. Rotores. Equilíbrio. Indústria.

Abstract

The study of rotor unbalancing is essential for safety and reliability in the design of machinery and other mechanical structures with rotating parts. By understanding the dynamic behavior and effects of unbalanced components, engineers can address this issue and establish a maintenance program to safeguard mechanical systems from excessive vibrations and prevent catastrophic failures, primarily due to fatigue. Balancing in one and two planes is the most common procedure to correct unbalanced masses in rotating parts. In this work, the aim is to advance the understanding of the effects of mass distribution in irregular-section parts and develop a tool to calculate and improve balancing. In certain situations, there are local limitations to corrections, such as adding correction mass due to irregularities and machine element constraints. To validate the proposed solution, a sample of a crankshaft and a camshaft actuation mechanism was used, where unbalancing is a common issue due to the inherent geometric discontinuities in these types of components. The achieved result demonstrated the validity of the application, allowing the balancing of the samples within the levels recommended by the standard.

Keywords: Vibration. Rotors. Balancing. Industry.

1. Introduction

Vibration is a common phenomenon in mechanical structures, mainly in rotating machinery, and one of the possible sources is due to the presence of unbalanced masses in these elements. The intensive trepidation produced by unbalanced forces can cause several problems, leading to the failure of the mechanical system. This imbalance in mechanical pieces occurs when the distribution of mass along of the body and around the axis is not homogenous. such results have already been discussed in the literature (Budynas and Nisbth, 2016; Coelho, 2018).

The unbalance in a shaft with a discontinuity is explained by the theory of rotating masses, where an eccentric mass rotating around an axis generates a centrifugal force proportional to this unbalanced mass, the distance from its orbit of rotation to the axis (eccentricity), and the square angular velocity as discussed in (Coelho, 2013). This kind of force causes alternative stress which in certain conditions can cause the shaft to fail due to fatigue phenomenon as discussed in (Wowk, 1998).

Unbalance in a crankshaft can lead to many problems, resulting in high levels of vibrations which can cause discomfort to vehicle occupants, as well as premature wear of bearings and other components that make the engine. Vibration can also affect engine performance, causing loss of power, inefficient fuel consumption, and increased emissions. To correct the unbalance on a crankshaft, apply balancing techniques. Balancing can be static or dynamic. No static balancing, but additional parts are added or removed from the crankshaft to achieve a balance. already out of balance dynamic, the shaft is rotated at high speed, and vibration sensors are used to measure the resulting vibrations. With Based on these measurements, the balance weights are adjusted until the vibrations are minimized. Then, the unbalance can negatively affect on engine performance and durability. This work intends to advance in the comprehension of the effects of mass distribution and to build a tool that it allows to calculate and make an improvement in balancing. The proposed application will help users to select the set of the plans to spread the correct masses along properly of the body rotor. Sometimes, it is not feasible to proceed with corrections on an exact local, because of building restrictions. To validate the technique, they were used as specimens a crankshaft shown in the Figure 1 and a camshaft, shown in the Figure 2 where imbalance is a characteristic problem due to the irregular geometry shape of these elements.

Figure 1 – This is a crankshaft of the four-stroke engine used as a specimen in the balancing rig.

Figure 2 – This is a camshaft of an engine used as a specimen in the balancing rig.

2. Unbalance analysis

It is extremely important for the proper functioning of the crank and connecting rod system in an internal combustion engine that the crankshaft is working properly. To understand unbalanced analysis, it is necessary to consider the unbalanced forces resulting from the variation in gas pressure and the unbalanced forces resulting from the inertia of the moving parts.

2.1 Unbalanced forces resulting, from varying gas pressure

The crankshaft plays a key role in converting linear motion into rotational motion, transmitting energy to propel the vehicle or mechanical system where it is installed. During its operating process, a crankshaft receives different loads due to the pressure of the gases, this pressure is not constant. Equation 1 considers unbalanced forces resulting from changing gas pressure. Figure 3 shows the system to be analyzed (Rao, 2018).

$$M_t = \left(\frac{F}{cos(\phi)}\right) cos(\theta), \tag{1}$$

where M_t it is the torque that tends to turn the crankshaft. It is necessary to consider a torque

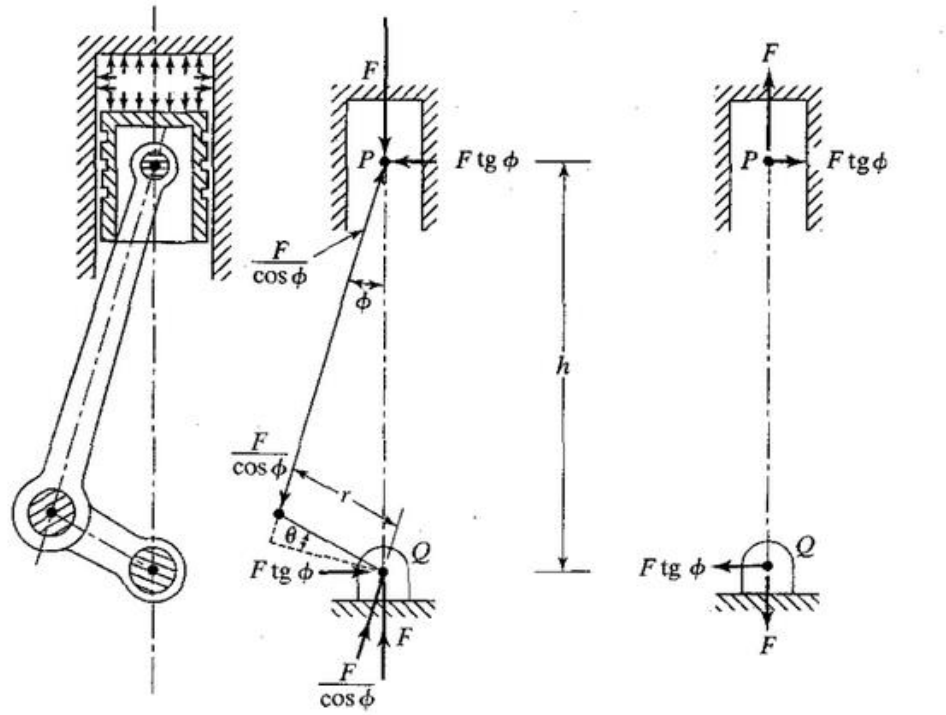

Figure 3 – Forces in a slider Crank Mechanism (Rao, 2018).

$M_Q = Fhtg\phi$ in the motor structure, which h can be determined by the geometry of the system presented in equation 2.

$$h = \frac{h\, cos\, \theta}{sin(\phi)} \tag{2}$$

Thus, the resulting torque will be described by equation 3.

$$M_Q = \frac{F_r cos\theta}{cos(\phi)} \tag{3}$$

2.2 Unbalanced forces resulting from inertia of moving parts

To analyze the unbalanced forces resulting from the inertia of the moving parts, it is necessary to classify three points: acceleration in the piston, acceleration of the crank pin and the forces of inertia.

2.2.1 Unbalanced forces resulting from inertia of moving parts

Figure 4 shows the schematic draw for a crankshaft r, the connecting rod l and the piston of an alternative engine. Consider a counterclockwise rotation with an angular velocity w. The displacement of piston P can be expressed in the equation 8.

$$\ddot{x_p} = rw^2(cos\,(wt) + \frac{r}{l} cos\,(2wt)). \tag{4}$$

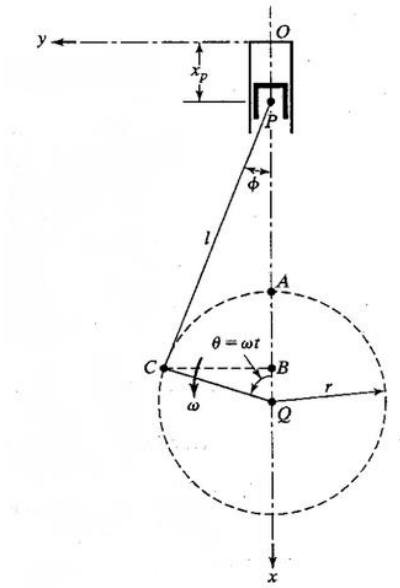

Figure 4 – Schematic draw for model the connecting rod and piston crank movements (Rao, 2018)

2.2.2 Crank pin acceleration

The acceleration of the crank pin, which can also be referred to as "radial acceleration of the crankshaft," is of utmost importance for the dynamics of internal combustion engines. The acceleration of the crank pin describes the rate of change of the crank pin's velocity over time. During the operation of the mechanical system in which it is embedded, the acceleration of the crank pin is influenced by the interaction between the pistons, connecting rods, and crankshaft. This acceleration is the result of the reciprocating forces generated by the combustion of gases in the cylinder and the motion of the pistons. These results were previously discussed in Silva (2013). The vertical and horizontal displacements of crank pin C with respect to the coordinate axes xy shown in the Figure 3 are given by:

$x_c = OA + AB = l + r(1 - cos\ (wt))$ and (5)
$y_c = CB = r\ sin\ (wt)$. (6)

The acceleration of the crank pin will be obtained by differentiating the equations 5 and 6 respect to time:

$\dot{x}_c = rw\ sin\ (wt)$, (7)

$\dot{y}_c = rw\ cos\ (wt)$, (8)
$\ddot{x}_c = rw^2\ cos\ (wt)$ and (9)
$\ddot{y}_c = rw^2\ sin\ (wt)$. (10)

2.2.3 Inertia force

Is it possible to calculate the unbalanced forces in a crankshaft using the following equations

$F_x = (m_p\ddot{x}_p + m_c\ddot{x}_c)$ and (11)

substituting the equation 4 and 9 in 11, for the accelerations P and C, it results in

$$F_x = (m_p + m_c) r w^2 \cos(wt) + m_p \left(\frac{r^2 w^2}{l}\right) \cos(2wt) \tag{12}$$

The vertical component is defined by

$$F_y = m_p \ddot{y}_p + m_c \ddot{y}_c, \tag{13}$$

where $\ddot{y}$ it is given by the equation 10, therefore

$$F_y = -m_c r w^2 \sin(wt). \tag{14}$$

2.3 Unbalance

An unbalance analysis is always required for such components. An unbalanced crankshaft can cause discomfort to vehicle occupants, mechanical system problems, and even component failures. For the crankshaft, an unbalanced analysis was carried out, where it is possible to identify a certain degree of deviation from the axis of inertia. We will see later about fatigue failure, this failure is initiated by a crack. This crack can be generated by the imbalance present in the crankshaft shaft. The unbalanced axis may experience greater stress on a certain part of your body, unevenly. This generates small stress concentrators and thus cracks. A Teknikao brand balancer, model NK750, was used for unbalance analysis. For this unbalanced analysis, a balancing class of G 250 was applied for crankshafts of 4-cylinder fast diesel engines, considering ISO 1940 classes. The crankshaft weighs 13.86 kg. All these parameters were entered into the calibration software Figure 5 shows the result of the analysis (Wowk, 2018)

Figure 5 – Unbalance analysis. The red color in the left circle represents an unbalanced condition, otherwise, in the right circle represents green color indicates balanced condition. Source: TEKNIKAO operational screen.

3. Fatigue Analysis

As we have seen earlier, a working crankshaft is subjected to alternating multi-axial loads, whether it's compression due to internal combustion engine explosions or its own working mode. Additionally, considering the unbalanced masses as our focus, they also generate repeated or fluctuating loads, which leads us to include a section on fatigue failure analysis in this work. It has long been known that elements subjected to alternating or fluctuating loads can fail due to fatigue.

Fatigue failure was first observed around 1800 when the axles of a railway wagon started to fail after a short period of service due to the varying loads from the train wheels. Fatigue failure is initiated by a crack and propagated due to cyclic loads. Figure 6 presents this process where A is the initiation of load propagation, B represents the waves, and C is the failure. In other words, fatigue starts with a nucleation point in the stress concentration region and spreads out around the cross-section forming a surface like well-known beach marks. One of the most popular models based on yield criteria is the von Mises model (Pereira, 2018). Equation 15 is according to the von Mises theory,

$$\sigma_1 \leq f_{-1}. \tag{15}$$

Where σ_1 is the maximum principal stress and f_{-1} is the fatigue strength under fully reversed normal loading. For the maximum shear theory, we have

$$\tau_{máximo} = \frac{\sigma_1 - \sigma_3}{2} \leq t_{-1}. \tag{16}$$

where σ_3 is the minimum principal stress and t_{-1} is the fatigue strength limit under fully reversed torsional loading. Finally, the von Mises theory can be written as

$$\sigma_{eq} = \sigma_{misses} \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \leq f_{-1}}. \tag{17}$$

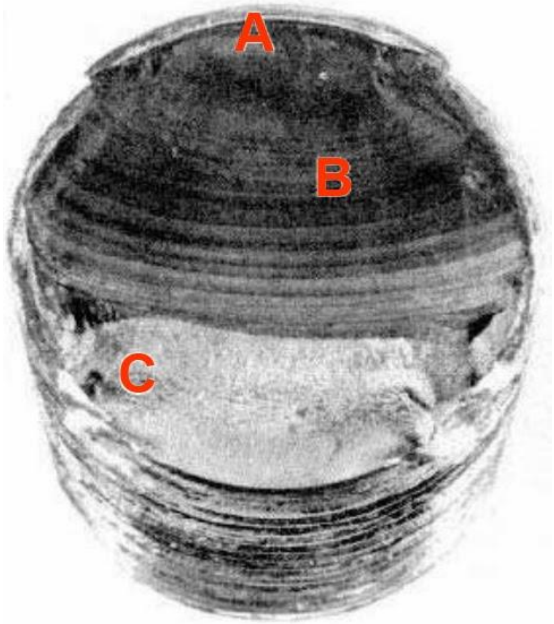

Figure 6– Fatigue failure. The failure started at the thread root at A, propagated across most of the cross-section shown by the beach marks at B, before final fast fracture at C. Source: Budynas and Nisbeth, 2016.

Here, σ_{misses} is the Von Mises stress and σ_2 is the intermediate principal stress. It is worth noting that the expressions 15, 16 and 17 are valid only for proportional loading, and furthermore, they can be complemented with a Goodman or Gerber failure criteria to account for the effects of mean stress presence (Pereira, 2018). Through a numerical analysis in the ANSYS software (Figure 7), it was clearly observed the application of the von Mises theory for fatigue failure. It was noticed that in the regions below, the applied load is lower. In the lateral supports, a higher overload was observed. It is worth noting that this analysis is focused only on the central part of the crankshaft as we are considering a region with higher loads during the working process.

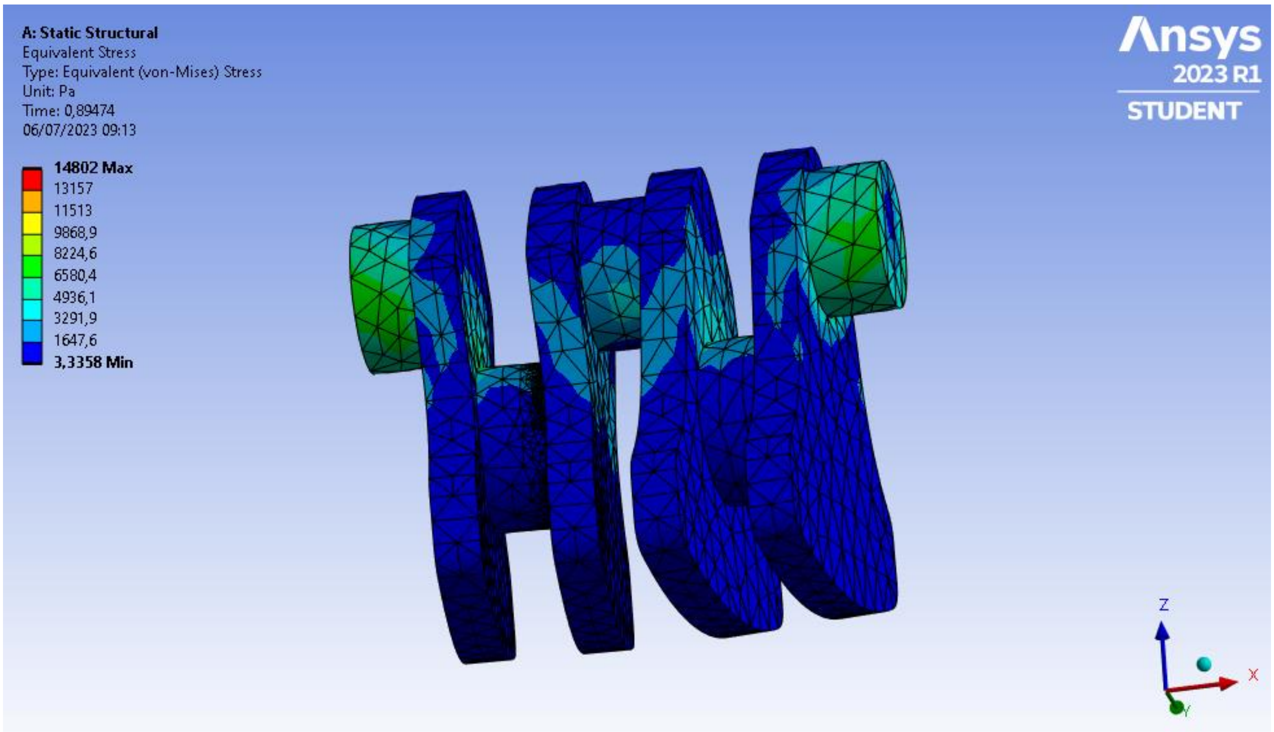

Figure 7 – Fatigue analysis in ANSYS

4. Conclusions

In summary, the completion of an unbalance analysis is essential for components like crankshafts. This study focused on conducting an unbalance analysis for a crankshaft, which revealed a certain degree of deviation from the inertia axis. The analysis utilized a Teknikao brand balancer, specifically the NK750 model, to assess the unbalance. A balancing grade of G 250, aligned with the ISO 1940 grades, was applied to cater to fast diesel engine crankshafts with 4 cylinders. The examined component had a weight of 13.86 kg. All relevant parameters were inputted into the calibration software, enabling a comprehensive analysis. This study highlights the importance of conducting thorough unbalance analyses to ensure the optimal functioning and longevity of crankshaft components. this study investigated the analysis of unbalance and fatigue in a crankshaft. A numerical analysis was performed in ANSYS, using the von Mises theory to observe fatigue failure occurrence. It was evident that in the lower regions, where the load was lower, the likelihood of fatigue failure was reduced. Conversely, in the lateral supports where there was higher overload, the likelihood of fatigue failure was higher.

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References

Budynas, R. G., & Nisbeth, J. K. (2016). Elementos de Máquinas de Shigley-10ª Edição. *McGraw Hill Brazil.*

Coelho, G. A. T. (2013). Balanceamento dinâmico de um simulador de máquinas rotativas.

Pereira, M. V. S. (2018). Avaliação estrutural de eixo virabrequim sob fadiga multiaxial. Ph.D. thesis, PUC-Rio.

Rao, S. S. (2018). Mechanical vibration. *Pearson Education*, United Kingdom, 6th edition. ISBN 1-292-17860-4.

Silva, R. A. R. d. (2013). Modelagem e simulação do mecanismo biela manivela e análise estrutural de biela automotiva.

Wowk, V. (1998). Machinery vibration: balancing. *McGraw Hill*, United Kingdom, 1st edition. ISBN 0071348611.