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Simulation by the finite element method of The crankshaft material under the influence of the variable wind force

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Abstract

This paper presents the comparison of two different approaches for crankcase structural analysis. The first approach is aconventional quasi-static simulation, which will not be detailed in this work and the second approach involves determiningthe dynamic loading generated by the crankshaft torsional, flexural and axial vibrations on the crankcase. The accuracy ofthis approach consists in the development of a robust mathematical model that can couple the dynamic characteristics ofthe crankshaft and the crankcase, representing realistically the interaction between both components. The methodology toevaluate these dynamic responses is referred to as hybrid simulation, which consists of the solution of the dynamics of anE-MBS (Elastic Multi Body System) coupled with consecutive FEA (Finite Element Analysis).

Keywords: crankcase, dynamic, FEM, crankshaft, material.

1.Introduction

Considering the trends for internal combustion enginesand specifically talking about diesel engines for commercialvehicles, it is clear that while the power output increases forevery new application, the mass, fuel consumption andengine final cost must develop in the opposite direction. Fromthese points of view, one relevant aspect is that thedownsizing of the engines is an actual trend [1].The engine considered in this study was calibrated toachieve Euro-V legislation targets without significantchanges in basic components utilized in previousapplications. This new engine performance presents 6.5%higher power output and 18% higher torque.Taking into account these input variables, the cranksha ftand crankcase durability were calculated consideringconventional methodologies which are accepted worldwideby many engine manufacturers, i.e. dynamic simulations for the crankshaft and quasi-static simulation for the crankcase atmain bearings region.After the end of the calibration phase and after somedurability tests at the dynamometer, some cracks occurred inthe block at the flywheel side in a fillet area, close to the mainbearing cap. The first conclusion from the failure analysisindicated some possible unbalance of the propeller shaft atthe test bed, but a deeper analysis of the problem highlightedthat the bending vibrations of the crankshaft was the rootcause of the crankcase failures.There are some possibilities to reduce crankshaft flexuralvibrations by increasing its bending stiffness. One effectivedesign change is to increase the overlap of the crankshaft bymodifying the crankpin and/or journal diameters. However,this proposal has many side impacts on the design affectingmany other components, including the crankcase, whichcould cause increased engine mass and final product costs.Since reducing the component's dynamic response bychanging the overlap is prohibitive, the adopted solution t[2].

Finite Element Model for Crankcase Structural Analysis The dynamic loads computed during steady state E-MBS simulations at the main bearing shells, are then transferred to the second FE model where the stresses at predefined timesteps are evaluated and considered for the subsequent fatigueanalysis.The FE model at this point shall be able to simulate theassembly and working stress in all regions of the crankcase.For that, the press fit of the main bearing shells, bolttightening forces and ladder frame geometric deviations mustbe taken into account.Figure 8 shows the finite element model for crankcasestructural analysis. It is important to note that the dynamicloads from E-MBS model are used as inputs in combinationwith bolt tightening, press fits, etc. The crankshaft shall notbe modeled here, what results in a considerable CPU timesaving for each interaction.

2. Finite element analysis and modeling

3.1 Mesh

The finite element method allows a mesh adapted to the problem dealt with. For explicit codes it is the same, and some additional stresses are added because of the wave paths in the structure. This paragraph presents the mesh used for the blade. During assem rent materials, so the contact is without int

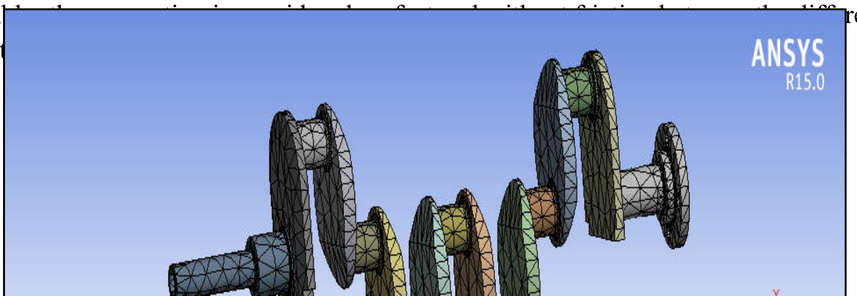


Fig.1. mesh of the crank shaft

3.2 Materials for manufacturing a wind crank shaft:

3.2.1 Problem of how to choose the material

The choice of materials for the manufacture of a blade is a critical task as it directly affects the design and fabrication methods used. Again, for materials with good structural properties, the two main qualities of the ideal material are their ratios of high strength / density (S/ρ) and stiffness / density (E/ρ)[6]. In this study we will work on glass fiber reinforced plastic (GFRP) and Carbon Fiber Reinforced Plastics (CFRP).

Table 1. Properties of the materials[5]

Reinforcing Material	Yield Strength (MPa)	Tensile Strength (MPa)	Elastic Modulus (GPa)	Strain at Break percent
GFRP	N/A	480-1,600	35-51	1.2-3.1
CFRP	N/A	1,720-3,690	120-580	0.5-1.9

3. Modal analysis and identification of stresses and deformations

Static analysis

The manufacturing material used is: alloy steel

Onexerce on the crankshaft a pressure of 35 bar

We fix the crankshaft in the side of Tourillon and let the side of Queueon movement, we do all our study on the .crankshaft on this basis

3.1 Modal deformities

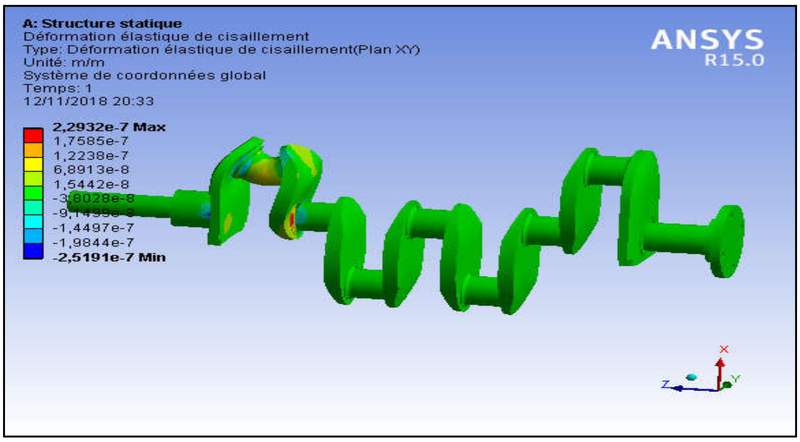


Fig.2. modes in terms the frequency for the

Through the previous tables, which study both vibration as well as deformation in terms of modes, we noticed that GFRP is more resistant than CFRP, as it achieved a lower percentage in terms of vibration as well as in terms of deformation.

Working hypotheses

4.1. Elasticity solution

The general form of an anisotropic stress–strain relationship can be defined as:

$$\sigma = D\varepsilon \qquad (01)$$

or

$$\varepsilon = C\sigma \qquad (02)$$

where **D** and **C** are anisotropic material stiffness and compliance matrices, respectively. Eq. (15) can also be written in a component form,

$$\varepsilon_i = C_{ij}\sigma_j \qquad i,j = 1,2,3,\dots\dots\dots,6 \qquad (03)$$

which only requires 36 independent constants c_{ij} due to symmetry properties. For an orthotropic material which has three mutually orthogonal planes of elastic symmetry, Eq. (03) is reduced to:

$$\begin{Bmatrix}\varepsilon_1\\ \varepsilon_2\\ \varepsilon_3\\ \varepsilon_4\\ \varepsilon_5\\ \varepsilon_6\end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0\\ C_{12} & C_{22} & C_{23} & 0 & 0 & 0\\ C_{13} & C_{23} & C_{33} & 0 & 0 & 0\\ 0 & 0 & 0 & C_{44} & 0 & 0\\ 0 & 0 & 0 & 0 & C_{55} & 0\\ 0 & 0 & 0 & 0 & 0 & C_{66}\end{bmatrix}\begin{Bmatrix}\sigma_1\\ \sigma_2\\ \sigma_3\\ \sigma_4\\ \sigma_5\\ \sigma_6\end{Bmatrix} \qquad (04)$$

and there remain only 5 independent constants for a transversely isotropic material,

$$\begin{Bmatrix}\varepsilon_1\\ \varepsilon_2\\ \varepsilon_3\\ \varepsilon_4\\ \varepsilon_5\\ \varepsilon_6\end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0\\ C_{12} & C_{11} & C_{13} & 0 & 0 & 0\\ C_{13} & C_{13} & C_{33} & 0 & 0 & 0\\ 0 & 0 & 0 & 2(C_{11}-C_{12}) & 0 & 0\\ 0 & 0 & 0 & 0 & C_{44} & 0\\ 0 & 0 & 0 & 0 & 0 & C_{44}\end{bmatrix}\begin{Bmatrix}\sigma_1\\ \sigma_2\\ \sigma_3\\ \sigma_4\\ \sigma_5\\ \sigma_6\end{Bmatrix} \qquad (05)$$

In the following table, we represent the Distributions of The equivalent stresses (Von Misses), and the equivalent strain on the blade and the displacement.

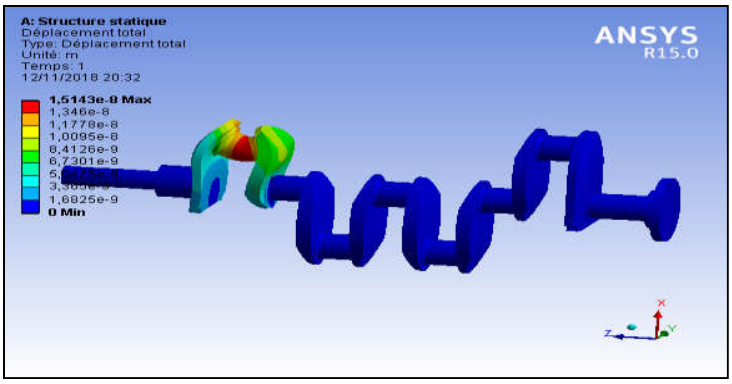


Fig.3. displacement in terms the lode

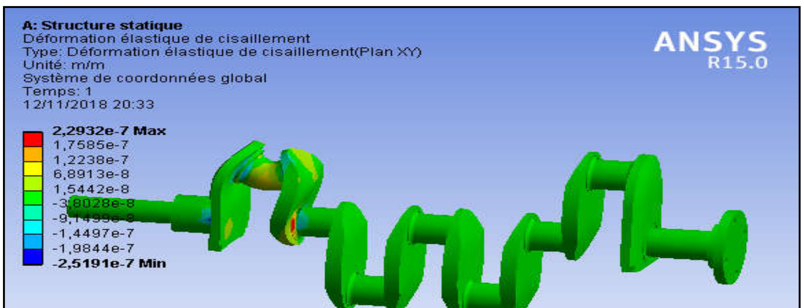


Fig.4. strain in terms the lode

As is evident in each column of the table, which is the change in lode in terms of stresses and strain, as well as the displacement to Glass fiber reinforced plastic and Carbon fiber reinforced plastic as we note that the value of stresses is approximately equal in the first and second material while it was greater in CFRP and this makes the GFRP better than the first[5]..

Modal analysis

Own natural frequencies

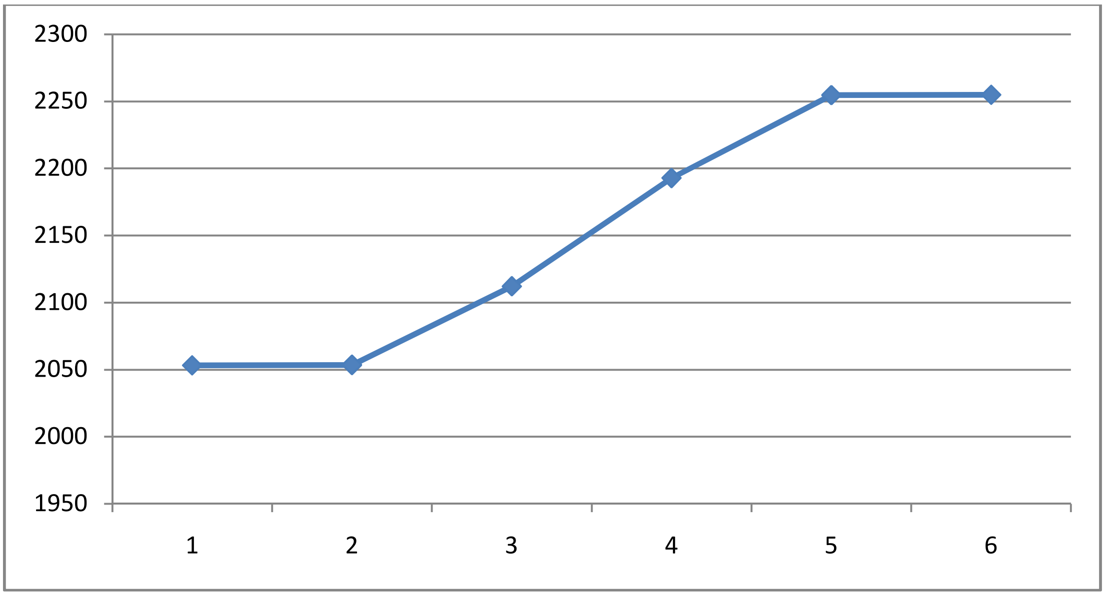
Eigen frequency trace according to 6 displacement models

The frequencies of the crankshaft are presented in the table below:

I.01.Table of natural frequencies

Fashion Number	Fashion Type	Frequency [HZ]
1	1ere battement	2053.2
2	2émé battement	2053.5
3	1er Trainé	2112.1
4	3émé battement	2192.8
5	4émé battement	2254.6
6	**1er Torsion**	**2254.8**

Frequency graph according to modes



4. Conclusions

The crankshaft consists of several aligned journals, on which it rotates. Between these bearings are offset pins, on which the connecting rods are mounted. The crankpins are connected to the bearings by masses.

The masses allow the dynamic balancing of the crankshaft. Their purpose is to reduce the vibrations due to the reciprocating movement of the pistons as well as to the asymmetry of the crank device.

The diameter of the crankpins needs to be large to reduce the vibrations that result from bending of the crankshaft.

The crankshaft carries, at the end designed to transmit power. At the other end, a correct shape ensures the timing of the distribution control gear and of the pulleys for the belt drive of the auxiliary members.

A crankshaft must be balanced statically and dynamically. Balancing the crankshaft is essential to reduce the engine vibrations caused by the forces and moments produced by the pressure of the gases in the cylinders and by the alternating and rotating parts, and to reduce the loads exerted on the bearings of the tree line. Two types of forces are generated by the moving parts linked to the crankshaft: centrifugal forces and alternative forces which cause vibrations.

Centrifugal forces

The crankshaft is statically balanced when the result of the centrifugal forces is zero, that is to say when the center of gravity is on the axis of rotation. A statically balanced crankshaft is not necessarily dynamically balanced, that is to say that it can give rise, when it is in rotation, to a bending moment due to centrifugal forces located in different planes.

Alternative forces

The crankpins also undergo forces due to the masses animated by an alternating movement. These forces, caused by variations in the speed of the piston and the connecting rod, are subdivided into alternative forces of the first and second order.

References

[1] *Y. Golfman, Hybrid anisotropic materials for wind power turbine blades. CRC Press, 2016.*

[2] *C. W. Kensche, "Fatigue of composites for wind turbines," International journal of fatigue, vol. 28, no. 10, pp. 1363-1374, 2006.*

[3] *T. Chelirem, "Etude dynamique d'une pale d'éolienne éfilée," 2010.*

[4] *U. S. a. R. R. J. Windte, "Validation of the RANS-Simulation of Laminar Separation Bubbles on Airfoils'," Aerospace Science and Technology, vol. 10, pp. 484 – 494, 2006.*

[5] *A. Nanni, A. De Luca, and H. J. Zadeh, Reinforced concrete with FRP bars: Mechanics and design. CRC Press, 2014.*

[6] *A. Chellil, "Etude des charges aéroélastiques d'une pale en composite," 2013.*

[7] *Y.-L. Lee, J. Pan, R. Hathaway, and M. Barkey, Fatigue testing and analysis: theory and practice. Butterworth-Heinemann, 2005.*