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DESIGN AND OPTIMIZATION OF AN AIRCRAFT PROPELLER FOR TUNED TORSIONAL VIBRATION DAMPING

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ABSTRACT

This paper analyzes the design process of an aircraft propeller for a piston engine. The propeller should also damp the main critical torsional frequency of the crankshaft. The first step was the calculation of the geometrical parameters of two different blades: one according to Larrabee's procedure and the other one according to the Theodorsen's theory. The evaluation of the effect of aerodynamics and centrifugal loads has required the union of the results come from CFD (Computational Fluid Dynamics) and the ones come from the CSM (Computational Structural Mechanics), through the execution of several one way FSI (Fluid Structure Interaction) analyses. The results allowed making pre-stressed modal analyses, which gave the opportunity to identify the kinds of propeller having the fundamental frequency coincident with the main resonance frequency of the crankshaft. The final design is a blade having the deformed shape of the optimum aerodynamic design.

Keywords: aircraft propeller, crankshaft; torsional vibrations, tuned damper.

INTRODUCTION

Torsional vibrations are a typical problem of crankshafts of piston engines because of the periodical behaviour of the generated torque [1]. Such vibrations can be dangerous for the structural integrity of the shaft and for this reason the installation of torsional vibration dampers or decouplers is essential between the propeller and the reduction unit.

The described phenomenon concerns many aeronautical piston-engines. In the last years, it has been noted that torsional vibration problems can be avoided by using a GFRP (Glass Fiber Reinforced Plastic) Propeller. These problems come out again if the propeller is made of other traditional materials like aluminium alloys or CFRP (Carbon Fibre Reinforced Plastic). Such a fact led to think that a blade made of composite material can absorb the energy generated by shaft vibrations. This study came to light on the basis of this possibility, with the aim to identify the essential parameters affecting the propeller structural behaviour.

The design criteria to realize a propeller able to damp torsional vibrations of the crankshaft was then searched.

THE STARTING POINT

Observing how GFRP "damping" propeller works, it has been noted that blades suffered of high bending deformations. This fact led to think that the materials properties of these propellers allow behaviour connected similar to a dynamic absorber. In this specific case the crankshaft is taken into account, considering a harmonic force applied to the system, as described in Figure-1:

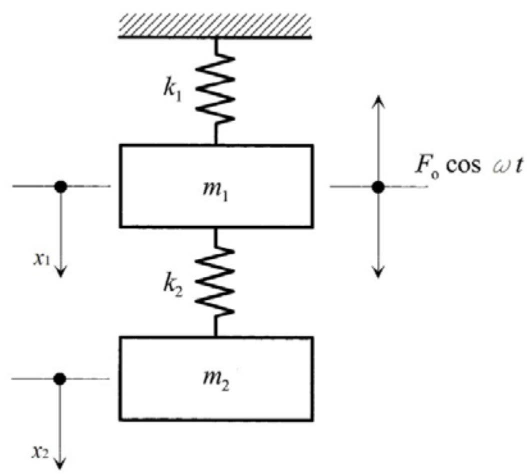

Figure-1. Scheme of the dynamic absorber.

The scheme shows a two degrees of freedom system, where the oscillation amplitudes of the masses can be calculated starting from the D'Alembert's equations, obtaining:

$$\begin{cases} X_{10} = \dfrac{k_2 - m_2\omega^2}{(k_1 + k_2 - m_1\omega^2)(k_2 - m_2\omega^2) - k_2^2} F_0 \\ X_{20} = \dfrac{k_2}{(k_1 + k_2 - m_1\omega^2)(k_2 - m_2\omega^2) - k_2^2} F_0 \end{cases}, \quad (1)$$

in which is clear that the oscillation of m_1 mass can be eliminated if the pulse excitation is equal to:

$$\omega = \sqrt{k_2 / m_2}\ . \quad (2)$$

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On the basis of this concept, it was made the decision to design a propeller for a well known Diesel engine, the aeronautical version of the FIAT 1.9 JTD.

Previous studies showed that this engine has got a resonance condition at a speed equal to 3030 rpm **(**Figure-2) that correspond to a resonance frequency of 50.5 Hz.

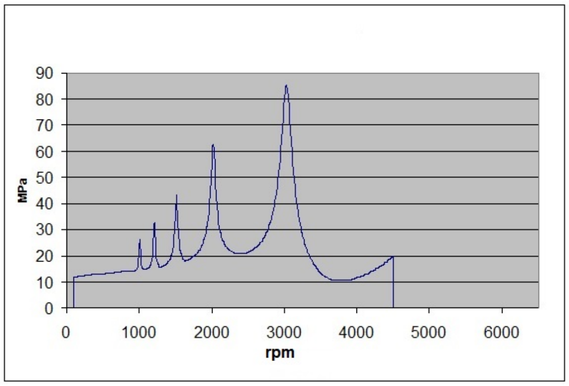

Figure-2. Shear stress profile.

Figure-2 shows several stress peaks, but the only to take into account is the one at 3030 rpm; in fact, this is the only one that exceeds the operating Lloyd curve. The resonance frequency of the propeller can be calculated with (3). The PRSU (Propeller Speed Reduction Unit) has a gear ratio *i* of 11/7.

$$f_p = \frac{f_s}{i} = \frac{50.5}{11/7} \approx 32.2 \text{ Hz.} \qquad (3)$$

Kevlar, Carbon and Glass Reinforced Plastic were used in the calculations with the properties shown in Table-1:

Table-1. Materials properties.

	CFRP	GFRP	KFRP
Young's modulus (GPa)	160	40	75
Poisson's ratio	0.3	0.25	0.34
Ultimate Tensile Strength (MPa)	1000	1000	1300
Density (g/cm^3)	1.6	1.9	1.4

AERODYNAMIC BLADE DESIGN

Two different procedures to design blades were implemented in order to be able to study possible differences in blade damping behaviour. The first methos used is the Larrabee's one. This is based on the theory of the blade element. In our case, the scripts made by Epperle have been used [2]. The result is very thin blades with a very large chord. The second method is the Crigles's procedure, based on Theodorsen's theory [3]. This second method outputs very tapered blades, that have the section of maximum chord closer to the hub. This geometry increases stiffness and strength.

The starting data are summarized in Table-2.

Table-2. Propeller requirements.

Propeller speed	**2800**	**rpm**
Aircraft velocity	100	m/s
Power	200	HP
Number of blades	2	[-]

The environmental conditions are ISA+0°C ground level. Due to previous experiences on racing propellers it was decided to choose a two blade propeller with Clark Y airfoil.

There are different ways to define the optimum diameter, but in this case the maximum value to avoid compressibility problems was chosen. The outside diameter is then 1,700 mm with tip velocity of about 0.81 Mach [4].

These data were implemented in the Epperle's scripts, which the result shown in Figure-3.

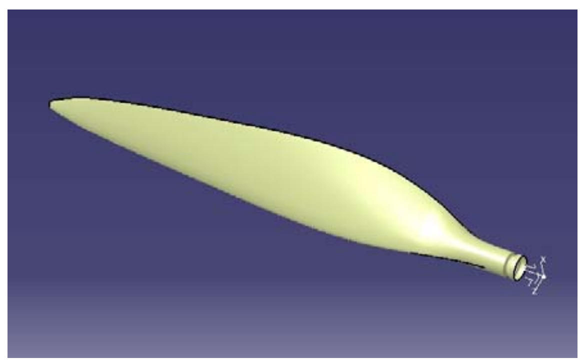

Figure-3. Larrabee blade.

Afterwards the Theodorsen's theory has been applied to design the blade of Figure-4.

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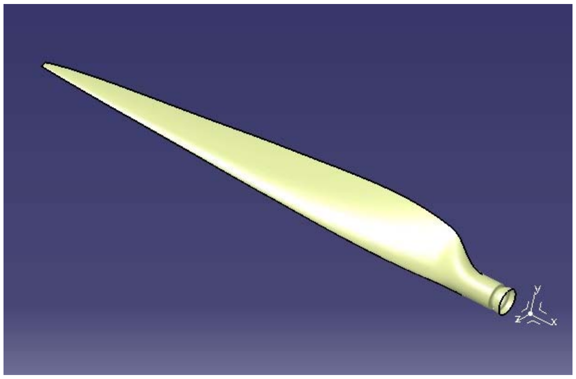

Figure-4. Theodorsen blade.

For technological reasons a constant thickness of the blade walls has been imposed. Thickness values ranging from 1 mm to 4 mm are compatible with a good manufacturing quality.

FSI ANALYSES

The effect of loads acting on the propeller, in operating conditions, causes a modification of the natural frequencies of the system. In order to realize the pre-stressed modal analyses, the execution of fluid-structure interaction study was necessary to evaluate the effect of different kinds of load. The first step concerned the analysis of the aerodynamic load through the use of a CFD (Computational Fluid Dynamic) solver, therefore a computation domain has been defined (Figure-5).

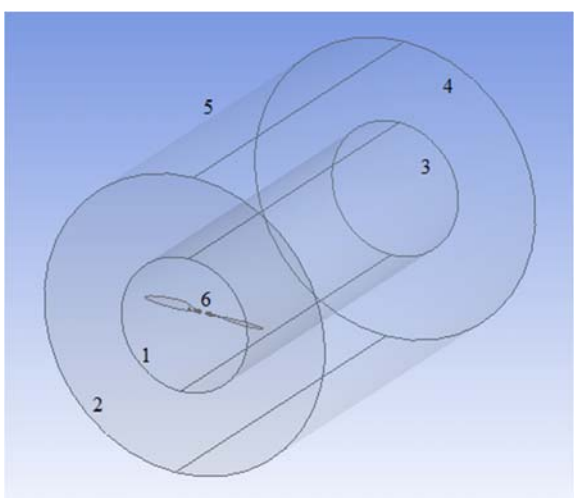

Figure-5. CFD domain.

As can be noted in Figure-5, the geometry of the blades has been subtracted using a Boolean operation and the domain is divided in two sub-domains: the inner one is rotating at the propeller speed, while the external one is static.

The boundary conditions imposed were:

- Velocity Inlet on surface 1 of Figure-5: the value is related to the speed of the aircraft.
- Velocity Inlet equal to zero on surface 2 of Figure-5.
- Null Pressure Outlet on surfaces 3 and 4 of Figure-5.
- Wall Unspecified on surface 5 of Figure-5.
- Wall Stationary on surfaces 6 of Figure-5, which representing the blades geometry.

The mesh has been modelled using tetrahedral elements.

Through these two models, the pressures acting on the blades surfaces have been identified; whereupon the results have been exported into the FEA (Finite Element Analysis) Static Structural model.

The second step involved the setting of the pre-stressed modal analyses and for this reason the propeller mesh has been created using 4 nodes isoparametric multilayer shell elements. The thickness of the blades could then be modified without re-meshing. Considering the constraints, the two blades have been coupled to the hub model through a perfect adhesion model (congruence of displacement at nodes). The aluminium alloy hub was constrained at the propeller shaft interface with a fixed end (Figure-6).

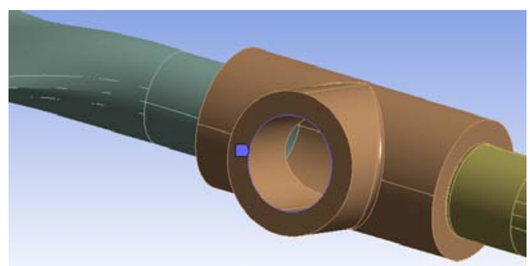

Figure-6. Fixed support on the hub.

PRESTRESSED MODAL ANALYSES

Several models have been defined by varying the thickness of the shell elements, the lay-up and the composite material. For each model a static analysis, followed by a modal analysis, have been performed, in order to define the natural frequencies of the system. Among the several configurations studied, three were found characterized by a first natural frequency not much different from the target value of 32.2 Hz:

- First configuration: a propeller with Larrabee blades in GFRP of 2.5 mm wall thickness. This model is characterized by a single natural frequency of 40.4 Hz in the operating range.
- Second configuration: a propeller with Larrabee blades in Kevlar of 1.3 mm wall thickness. This model is characterized by a single natural frequency of 52 Hz in the operating range.
- Third configuration: a propeller with Theodorsen blades in GFRP of 2 mm walls thickness,

characterized by a single natural frequency of 43.4 Hz in the operating range.

These three frequencies don't coincide with the target value; therefore a technique widely used in the helicopter blades has been adopted. A mass has been inserted in the tips of the blades, in order to match the crankshaft natural frequency required for tuned damping. Taking as reference the three models just described, the following results have been obtained:

- A mass of 120 g has been added to the blades of the first configuration to obtain a fundamental frequency of 32.5 Hz (Figure-7).

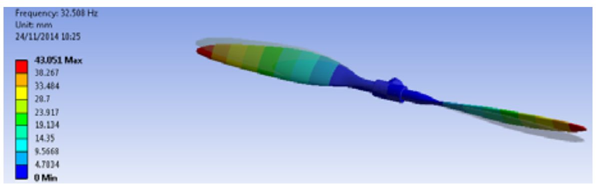

Figure-7. Modal shape of the first configuration.

- A mass of 145 g has been added to the blades of the second configuration to obtain a fundamental frequency of 32.9 Hz (Figure-8).

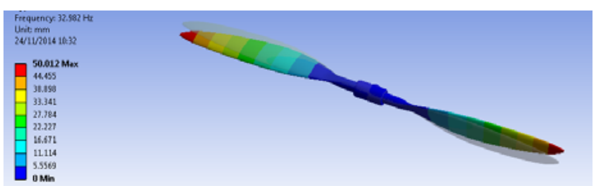

Figure-8. Modal shape of the second configuration.

- A mass of 120 g has been added to the blades of the third configuration to obtain a fundamental frequency of 32.7 Hz (Figure-9).

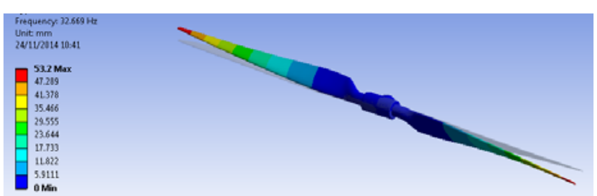

Figure-9. Modal shape of the third configuration.

Figures 7-9 show, in all three cases, a flapping (flexural) mode in resonance conditions matches with acceptable precision the required natural frequency of 32.7 Hz [5].

FINAL OPTIMIZATION

The GFRP-Larrabee blade Figure-3 and Figure-7 is a good compromise between costs, mass and performance.

The need to optimize the shape is due to the fact that in operative conditions the blade shape changes due to the loads, causing an aerodynamic efficiency reduction. For this reason, the orientation of the various profiles has been modified to obtain a blade that, in maximum performance conditions (2800 rpm and 100 m/s), has the Larrabee optimized shape (Figure-10).

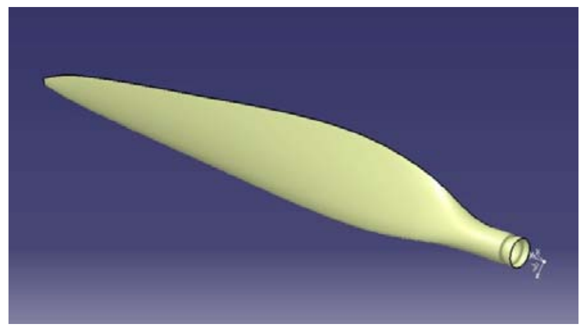

Figure-10. The optimized blade.

Figure-11 shows that the deformed optimized propeller has a nearly optimum shape.

Figure-11. Deformed propeller in operative conditions.

From this new CAD model, the FSI analysis and the pre-stressed modal analysis have been executed again, from which is resulted that this kind of propeller is characterized by a fundamental frequency of 32.5 Hz (Figure-12).

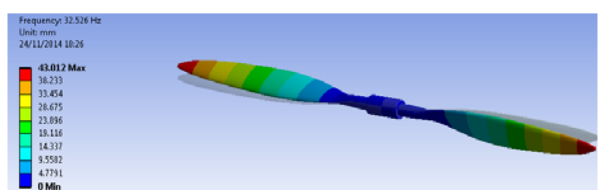

Figure-12. Modal shape of the optimized propeller.

Finally the FSI analysis have been used to compute the Safety Factors (SF). It was found that the SF lower values were close to the hub (Figure-13) but nevertheless the SF results equal to 15 practically

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anywhere, while in the worse points its value never decreases below 7.

Figure-13. The SFs.

A practical design rule says that a propeller must be characterized by a factor of safety equal to 15 in every point of the system, thus, considering the results showed in Figure-13, another simulation has been performed. In this case the propeller has been submitted to a centrifugal force equal to the twice of the maximum rotational speed in real conditions; therefore a rotational speed of 5600 rpm has been imposed to the structure and it was found that the higher stress calculated was 987.8 MPa (Figure-14), which is lower than the UTS (Ultimate Tensile Strength) of the material.

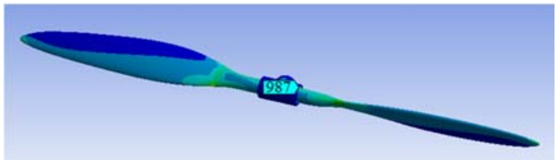

Figure-14. Static analysis at 5,600 rpm.

CONCLUSIONS

The present study demonstrates that is possible to design a propeller usable as a torsional vibration damper of the crankshaft. In fact the new CAE (Computer Aided Engineering) design tools, combined with the correct material model and a suitable mass distribution, allow the achievement of this result. The material choice is the most critical issue; in fact a small Young's Modulus and acceptable UTS (Ultimate Tensile Stress) are necessary.

By using a propeller working as an absorber, crankshaft damping devices can be eliminated. In this way the total mass can be reduced of more than 8 kg and the reliability can be improved by reduced the number of parts.

Symbols

X_{10}	Oscillation amplitude of the 1st mass of the 2 DOF system	[m]
X_{20}	Oscillation amplitude of the 2nd mass of the 2 DOF system	[m]
m_1	1st mass of the 2 DOF system	[kg]
m_2	2nd mass of the 2 DOF system	[kg]
k_1	1st spring stiffness of the 2 DOF system	[N/m]
k_2	2nd spring stiffness of the 2 DOF system	[N/]
ω	Pulse excitation	[1/s]
F_0	Amplitude of the harmonic force	[N]
f_s	Resonance frequency of the crankshaft	[Hz]
f_p	Fundamental frequency of the propeller	[Hz]
i	Gear ratio of the PSRU	[-]
t	Time	[s]

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