NUMERICAL AND EXPERIMENTAL ANALYSES REGARDING THE EIGEN FREQUENCIES OF A CENTRIFUGAL COMPRESSOR MADE OF COMPOSITES

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The composite materials are well known for their excellent combination of high structural stiffness and low weight. Their inherent anisotropy allows the designer to tailor the material in order to achieve the desired performance requirements. The objective of this work is to partially validate the design and manufacturing of a centrifugal rotor made of composite materials using the autoclave technology. The partial validation consists in investigating the dynamic behaviour of the centrifugal impeller, by both numerical and experimental analysis. The results obtained can be used in correlating the layers distribution, the design and the technology in order to obtain optimal results for the composite centrifugal compressor (low weight and low cost).

Keywords: comparative study, carbon fibre composites, experimental modal analysis, finite element analysis

1. Introduction

Since the first commercial passenger aircraft, the design and manufacturing of aircraft has evolved considerably, with a tendency to reduce fuel consumption as well as pollutant emissions, consisting mainly in carbon dioxide and nitrogen oxides, with direct impact on climate change or global warming. An emerging increase of flights was observed in last 10 years, from 100.000/day in 2009 compared to 150.000/day in 2019, and the number is increasing [1]. As mass is one of the key factors for fuel consumption with direct implications in pollutant emissions, a major decision in the development of aviation industry was to use

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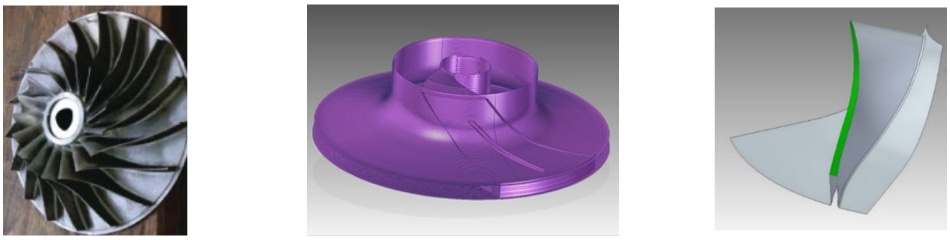
materials with lower density and high mechanical properties that will contribute substantially in decreasing the emissions and in increasing travel distance as less fuel will be consumed for the same travel distance. Advanced composite materials were considered in primary and secondary structures in aerospace industry, applicable from fuselages to even gas turbines. Gas turbines are the most versatile turbomachines, and can be used in a variety of ways and configurations for important industries such as power generation, oil extraction or aviation [2, 3]. Polymer composite materials (PMC) have been also used in the cold section of the aircraft engines, mainly in the compression section for development of composite stators or casings (General Electric [4], Airbus [5], Volvo, [6]).

The centrifugal compressor is a turbomachine that compresses a working fluid by applying inertial forces on it with the help of blades mounted on a rotor in rotating motion [7, 8]. As part of the compression section of the gas turbine, the main component under study in this paper, the centrifugal rotor, is considered to be manufactured out of carbon fibre composite materials (CFRP), thus increasing the possibility for further mass improvements. The usage of composite materials is due to their high specific strength (strength/density) and high specific modulus (modulus/density). However, several aspects have to be considered in regard to the design, analysis and fabrication of such complex components. Firstly, the rotor blades are the most critical parts, as they have a key role in the efficiency and performance of the gas turbine, and also, they are exposed to very turbulent conditions and high pressures [9-11]. In order to ensure a high design reliability, computational fluid dynamic analysis (CFD) and finite element analysis (FEA) shall be employed. A close relation must be kept between analysis and design in order to have a centrifugal rotor design that combines both dynamic behaviour, high efficiency and high mechanical stability [9, 12]. The dynamic behaviour is generally given by the rotation of centrifugal rotor and its interaction with the working fluid. Another important aspect is the fabrication technology, as it gives the dimensional accuracy of the component. The use of improper fabrication technology can generate profile deviations, delamination or micro-cracking that can lead to failure of the component, and in some cases failure during operations, which can be extremely dangerous. For this application, autoclave technology was selected as it combines pressure, vacuum and high polymerisation temperature to ensure that each layer of the composite materials is strongly bonded, eliminates airgaps between layers and respects the geometrical design requirements by using corresponding moulds.

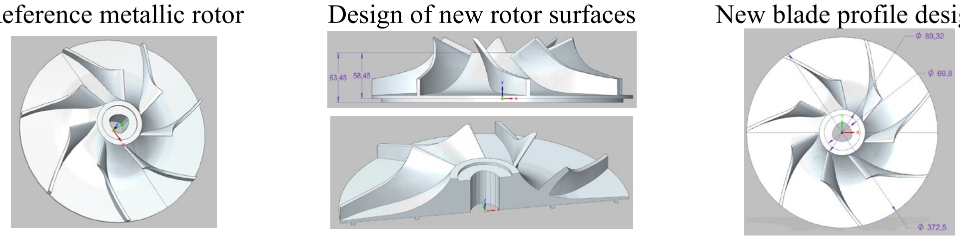
The particular case study performed in this paper was a centrifugal rotor manufactured from CFRP materials using the autoclave technology. The composite rotor was subjected to FEA and modal testing, to determine and verify the vibration characteristics of the impeller using both experimental and numerical methods.

2. Design and manufacturing of the compressor rotor

Using the metallic (17-4 PH) rotor reference from the CCAE 9-125 air compressor produced by the Romanian Research and Development Institute for Gas Turbines COMOTI, a new centrifugal rotor design was created using SolidEdge software (version 2019). During design iterations, CFD analysis was performed using Numeca Fine/Turbo solver and Spalart Allmaras turbulence model (upwind 2 discretization scheme [13]) and FEA studies (using ANSYS software) were performed, generating a new, optimised centrifugal rotor. The final geometrical model consisted in reducing the number of blades from 17 and 3 mm thickness to 7 blades and 6 mm thickness (3 mm wall thickness and 3 mm inner clearance), as presented in Figure 1 [14, 15, 16]. The material used in this study was a pre-impregnated (prepreg) material, HexPlyM49/42%/200T2X2/CHS, having 42 %vol. epoxy resin content and high strength carbon fibres. The material properties are presented in Table 1.



Reference metallic rotor Design of new rotor surfaces New blade profile design



Final centrifugal rotor design

Fig. 1. New centrifugal rotor design [14, 15, 16]

Table 1

HexPlyM49/42%/200T2X2/CHS material properties [17]

M49/42%/200T2x2/CHS-3k	**Value according to the manufacturer**
Density [g / cm3]	1.47
Young's modulus [N / mm^2]	55 000
Poisson's coefficient	0.28
Thermal expansion coefficient [ppm/°C]	2.1 x 10^{-6}
Tensile strength [N / mm^2]	850-900
Flexural strength [N / mm^2]	940
Compression strength [N / mm^2]	670
Fatigue strength [N / mm^2]	550
Shear strength modulus (GPa)	5
Interlaminar shear strength [N / mm^2]	63

The centrifugal rotor was manufactured in a three-steps process. During the first stage, the rotor blades were manufactured using autoclave technology and using a two-component metallic mould that corresponds to the blade design model, one mould represents the suction side and the second mould represents the pressure side. Using the two-component moulds, 7 blades were manufactured at a temperature of 120°C, 7 bar pressure and 0.9 mbar vacuum in the vacuum bag for a period of 120 minutes (Figure 2). The second manufacturing stage consisted in creating the rotor by bonding all 7 individual composite blades. For this process, another metallic mould was used, in order to create the disc of the rotor. On the mould, several layers of prepreg were laid onto which all 7 blades were positioned using multiple acrylonitrile butadiene styrene (ABS) moulds. For the third manufacturing stage, the multiple composite layers were placed on the rotor disc back side, to form the entire rotor assembly (disc and blades), as seen in Figure 3. A set of metallic rings was placed on the back side of the rotor to ensure the necessary removal material for balancing operations. The final composite centrifugal rotor is presented in Figure 4.

(a) (b)

Fig. 2. (a) Two-component mould used to manufacture the (b) composite rotor blades

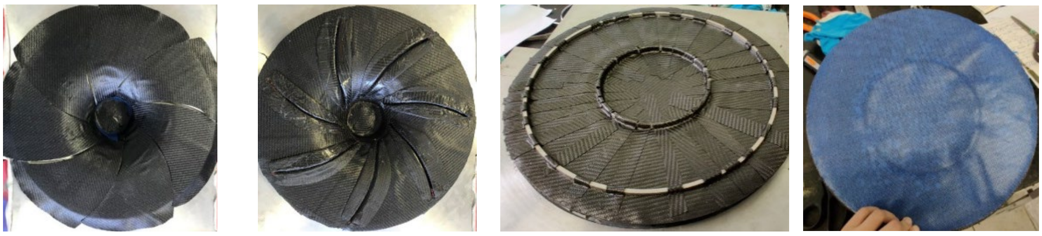


Fig. 3. Second and third manufacturing stages for the composite centrifugal rotor

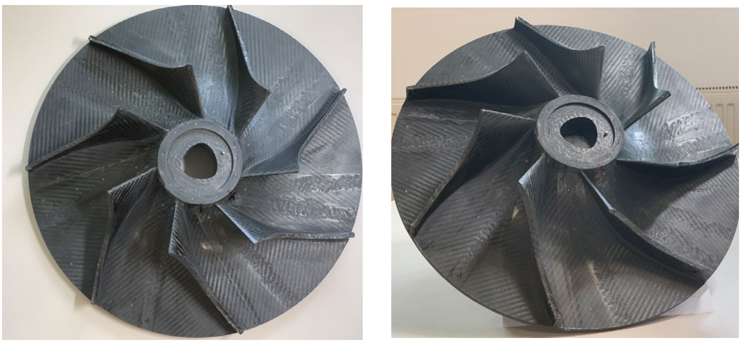


Fig. 4. Final composite centrifugal rotor

3. Compressor rotor modal analysis

Modal analysis plays an important role in the design and optimization of dynamic structures such as rotating machines. This method is employed to determine the natural frequencies and mode shapes of a component when it is subjected to resonance conditions. Modal characters are inherent of the component. Centrifugal impellers are high speed rotating components that have a high chance of rotor failure by attaining resonance frequencies. While rotating, stresses are created in the impeller due to its inertial effects which may change the inherent character of the impeller namely the natural frequencies and mode shapes.

3.1. Mesh model

The FEA modal analysis of the centrifugal impeller made of CFRP materials was performed in order to identify the natural frequencies and mode shape by using the ANSYS-ICEM FEA package. For this analysis the 3D CAD model presented in Figure 1 was imported in the ANSYS workbench. As the impeller has a complex geometrical design, hexahedron elements were used to generate the mesh in order to obtain better results. The generated mesh is presented in Figure 5.

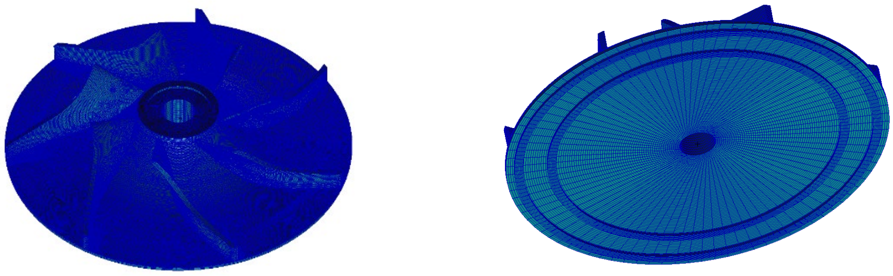
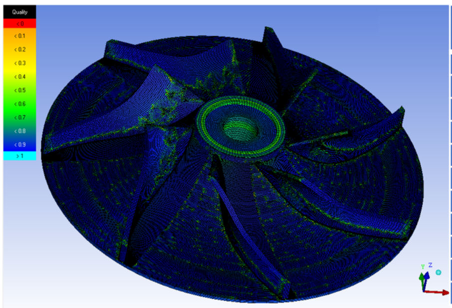


Fig. 5. Centrifugal impeller mesh model

The model contains a total number of 379643 hexahedron elements. A quality histogram was generated in ANSYS-ICEM and it is presented in Figure 6.



Quality Index	No. Of elements (%)	Quality Index	No. Of elements (%)
0.95 -> 1.0 :	157,446 (56.302%)	0.45 -> 0.5 :	1,239 (0.443%)
0.9 -> 0.95 :	49,369 (17.654%)	0.4 -> 0.45 :	796 (0.285%)
0.9 -> 0.95 :	49,369 (17.654%)	0.35 -> 0.4 :	454 (0.162%)
0.85 -> 0.9 :	23,367 (8.356%)	0.3 -> 0.35 :	238 (0.085%)
0.8 -> 0.85 :	17,709 (6.333%)	0.25 -> 0.3 :	119 (0.043%)
0.75 -> 0.8 :	9,487 (3.393%)	0.2 -> 0.25 :	0 (0.000%)
0.7 -> 0.75 :	6,260 (2.239%)	0.15 -> 0.2 :	0 (0.000%)
0.65 -> 0.7 :	4,910 (1.756%)	0.1 -> 0.15 :	0 (0.000%)
0.6 -> 0.65 :	3,370 (1.205%)	0.05 -> 0.1 :	0 (0.000%)
0.55 -> 0.6 :	3,361 (1.202%)	0.0 -> 0.05 :	0 (0.000%)
0.5 -> 0.55 :	1,518 (0.543%)		

Fig. 6. Centrifugal impeller quality histogram

3.2. Boundary conditions

The boundary conditions used in the modal analysis of the centrifugal rotor made of composite materials were rotor fixed in Z=0 plane and axial symmetry on the circular contour near the rotor inlet. These conditions are described also in [14, ,18, 19].

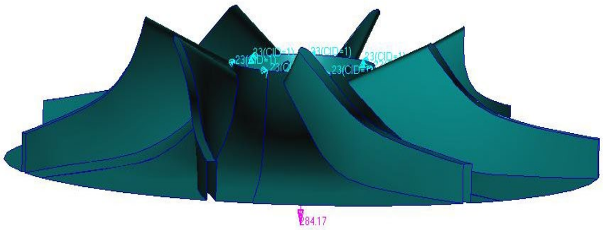


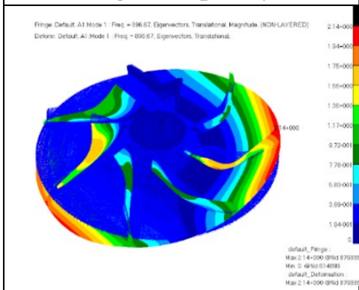
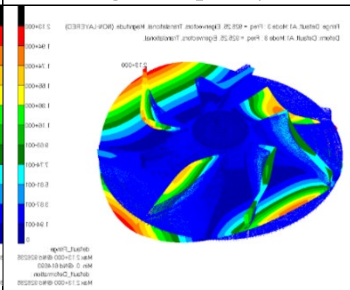
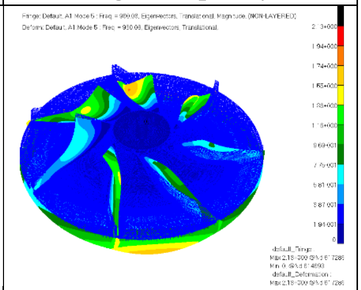
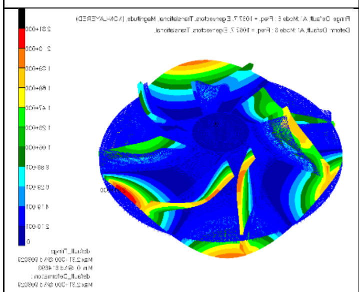
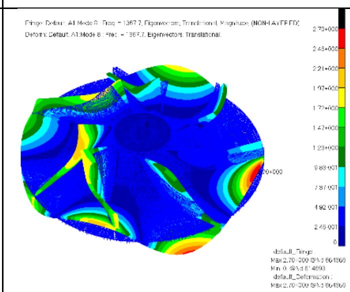
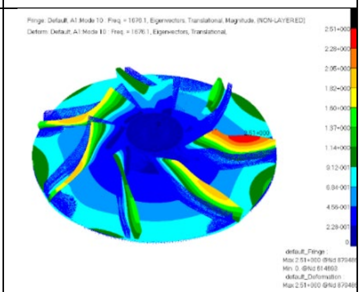
Fig. 7. Centrifugal impeller – boundary conditions

3.3. Numerical analysis results

Following the analyses performed for the composite centrifugal rotor, the graphical results are presented in Table 2 below.

Table 2

Eigen frequency values obtained for the composite centrifugal rotor

Eigen vector – [mm] / Eigen frequency	Eigen vector – [mm] / Eigen frequency	Eigen vector – [mm] / Eigen frequency
		
F1=897 Hz	*F2=925 Hz*	*F3 = 960 Hz*
		
F4=1068 Hz	*F5 = 1368 Hz*	*F6 = 1676 Hz*

The analysis of the Eigen vectors shows that the vibration modes are actually complex modes obtained by the combining/interaction of the Eigen frequencies of the composite rotor components used to defined the mesh model, such as:

- Metallic balancing rings, characterized by homogenous, isotropic material, having axial symmetry;
- Disc, modelled with an orthotropic material, having an axial symmetry with a periodicity of 3 elements per circumference (due to the K profile shape);
- Blades, modelled using a laminated, orthotropic material, with a geometry characterized by axial symmetry, having 7 elements per circumference.

One can observe that the main modal shapes (the first three Eigen frequencies) obtained are characterized by the bending of the disk. Moreover, the blades have a significant role in these main modal shapes.

In order to compare the theoretical results with the experimental ones, a frequency response analysis was performed for the entire rotor assembly, modelled with 1D elements considering the configuration used in the experimentation phase.

The 1D model is presented in Figure 8. The rotor shaft was modelled with shaft type elements, while the centrifugal rotor was modelled as a material point, placed in the rotor shaft center mass, having the inertial properties of the 3D rotor shaft (mass, moment of inertia). The frequency response of the rotor assembly is presented in Figure 9, for a frequency band from 0 to 1000 Hz. One can see a critical frequency, F1 = 660 Hz, which may be identified following the Ping test.

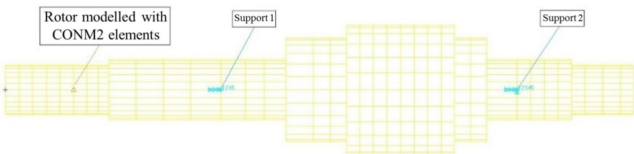


Fig. 8. 1D model used for the Ping test simulation

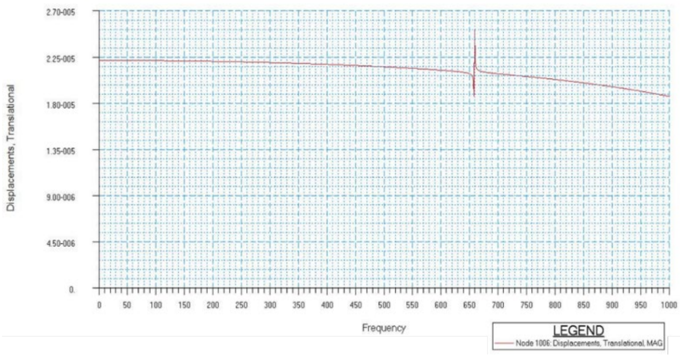


Fig. 9. Frequency response – Radial displacement of the rotor's center mass

Table 3

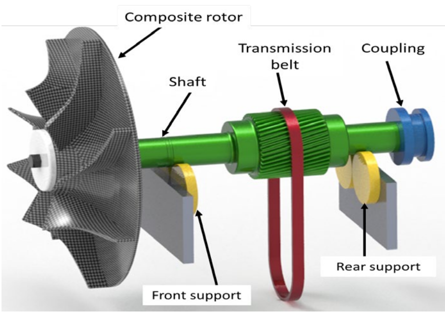
Eigen frequencies of the rotor assembly determined by numerical simulation based on the testing configuration

Nr.	Frequency [Hz]	Observations
1	660	Eigen frequency of the rotor assembly for the ping test conditions
2	897	First Eigen frequency of the rotor (first nodal diameter)
3	925	Second Eigen frequency of the rotor (second nodal diameter)
4	960	Third Eigen frequency of the rotor
5	1068	Fourth Eigen frequency of the rotor (third nodal diameter)
6	1368	Complex vibration mode
7	1676	Complex vibration mode

4. Experimental analysis

The experimental modal analysis is one of the methods employed to determine the natural frequencies of an assembly and for the current study, the impact hammer method was selected, allowing the Frequency Response Function (FRF) to be measured by means of an analyser which is based on the Fast Fourier Transform (FFT), as a rapid, convenient and low-cost method [20]. The centrifugal rotor was excited using a non-instrumented hammer, and the response was measured by miniature accelerometers which have a mass of 0.5 grams. Using these ultra-low weight accelerometers lead to a minimal influence of the blades' frequency response, respectively of the centrifugal compressor rotor. The main characteristics of the process used to determine the composite centrifugal impeller Eigen frequencies are:

- Centrifugal rotor mounted on the shaft, which is then placed on two supports (bearings) – Fig. 10;
- The uniaxial accelerometers are placed in points indicated (in the first phase an accelerometer was placed on each blade, while for the rotor analysis three accelerometers were placed on its disk – Figure 11;



a) CAD model

b) Testing configuration

Fig. 10. Rotor assembly configuration used in the modal analysis experimentation

The modal analysis was performed using the following equipment:
- Multichannel data acquisition system Sirius from Dewesoft;
- 3 Dytran 3224A1 accelerometers (0.5 g each).

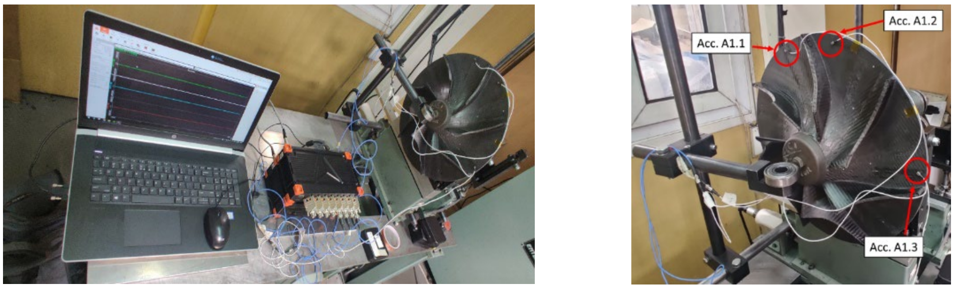


Fig. 11. Experimentation configuration for the composite centrifugal rotor modal analysis

The sampling frequency used for the vibrations signal set to 50 kS/s. The measuring procedure of the Eigen frequencies of the rotor consisted in mounting an accelerometer on each blade, using cyanoacrylate adhesive which ensures a rigid fixing, avoiding possible damping or non-linear phenomenon. The next step consisted of applying three shocks on each blade in different points on the blade surface. The raw signals post-processing presumes in selecting each shock and performing an FFT analysis in the linear mediation frequency domain. Figure 12 presents the first shock results. The resulting spectre indicates all the resonant frequencies considering the analysed frequency domain. Figure 13 presents the experimental data obtained from the shock applied to the first teste rotor blade. From the spectral analysis presented in Figure 12, it is observed that the first frequency value is 455 Hz. As frequency increases, the spectre at 645 Hz is found only at accelerometer A1.2 (Figure 11) while for 687 Hz, accelerometers A1.1 and A1.3 have an increased amplitude, differing than A1.2 which has a low amplitude. This can be explained due to positioning of A1.2 in a vibration mode node.

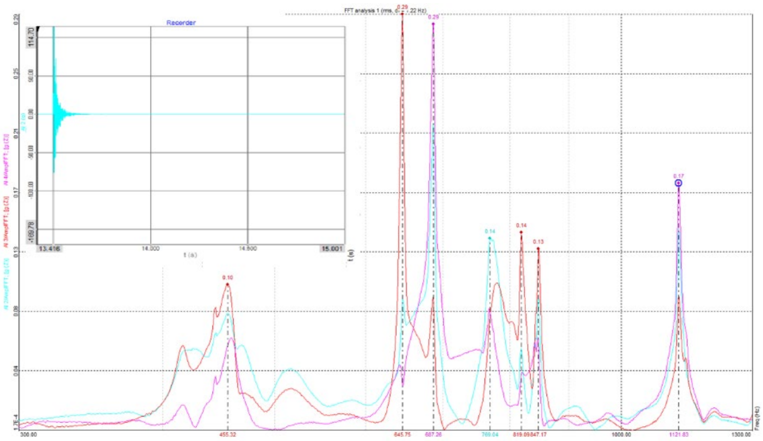


Fig. 12. First shock results obtained for the composite rotor

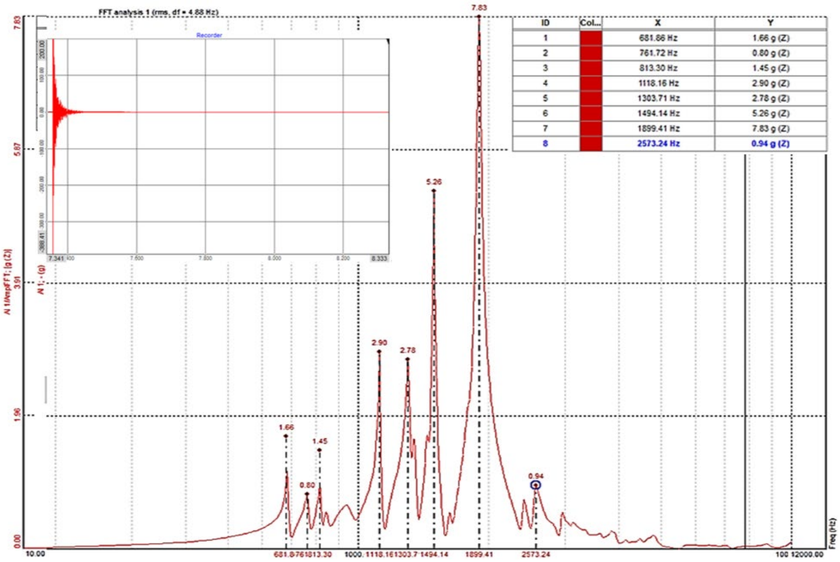


Fig. 13. First shock results obtained for the first rotor blade

The results in Figure 13 indicates that the first resonant frequency is 1988 Hz, with the highest amplitude. Second and third shock responses suggest the same frequencies (Figure 14) but with different amplitudes. Differences in amplitude were caused by the fact that the application position of each shock as well as the energy applied varied with each shock. Applying the shock to different points lead to excitation of the vibration mode node at that point, which leads to an increase in the amplitude response of that frequency. Also, the variation of the shock energy leads to a variable response in amplitude.

5. Comparison between the numerical and experimental evaluation

To have a better understanding on the modal behaviour and response of the centrifugal composite rotor, a parallel analysis has been made, presenting in Figure 14, the comparison between the FEM analysis and experimental data results.

Ping test					FEM analysis	Ping test on each blade	
Freq. [Hz]	1st shock [Hz]	2nd shock [Hz]	3rd shock [Hz]	Avg. Freq. [Hz]	Freq. [Hz]	Blade no.	Freq. [Hz]
F1	455	457	456	456	660	~~F1~~	658
F2	646	645	644	645	897	F2	756
F3	687	-	682	685	925	F3	876
F4	769	-	762	766	960	F4	1119
F5	819	849	845	838	1068	F5	1393
F6	1121	1123	1136	1128	1368	F6	1724
F7					1676		

FEM and experimental Ping test results

Ping test on rotor [Hz]	FEA analysis [Hz]	Maximum relative deviation [%]
F2 = 645	F1 = 660	2.3%
F3 = 685	F1 = 660	3.3%
F5 = 838	F2 = 897	6.4%
F6 = 1127	F5 = 1068	5.5%
Ping test on blades [Hz]	**FEA analysis [Hz]**	**Maximum relative deviation [%]**
F1 = 658	F1 = 660	0.3%
F3 = 876	F2 = 897	2.3%
F4 = 1119	F5 = 1068	4.8%
F5 = 1393	F6 = 1368	1.8%
F6 = 1724	F7 = 1676	2.8%

The maximum relative deviation

Fig. 14. Comparison between the FEM and experimental analysis

The results presented in Figure 14 represent the correlations between the measured experimental frequencies obtained during the Ping test performed on the composite rotor disc and blades, separately, and the modal analysis results. It was found that F2 and F3 frequencies can be associated with the first resonance frequency of the rotor assembly, F5 is associated to the first nodal diameter while the F6 is associated to the third nodal diameter. Concerning the composite rotor blades, F1 corresponds to the rotor assembly (more exactly the shaft vibration is transmitted to the blades and disc). The same applies for F2 and F3 frequencies. F4 is associated to the third nodal diameter and to F5 from modal analysis, F5 is associated to the F6 from modal analysis and F6 corresponds to F7 results from modal analysis. It is to be noted that the Ping test simulation has been performed through a frequency response type analysis to which the excitation amplitude remains constant. The Ping test consisted in applying a shock load whose frequency domain includes different amplitudes. However, is possible that the shock excitation during the Ping test to have a component with a frequency of 455 Hz and a higher amplitude. Moreover, the fixing system is different between the numerical and experimental testing, as during the numerical analysis the shaft-rotor assembly is mounted on the two supports as in Figure 8 while for the experimental analysis (Ping test), the assembly is mounted on the balancing machine supports and using the machine transmission belt to fix the rotor, as in Figure 10. The latter, induces an oscillating vibration mode, due to the elastic behaviour of the transmission belt and the oscillating stand for the balancing machine.

6. Conclusions

An investigation has been carried out to examine in detail the vibration characteristic of a new centrifugal impeller realized from CFRP materials.

The comparison between the experimental analysis results with that of the modal analysis (analysis of Eigen modes, frequency response) shows that the finite element model is adequate to the real one within the limits of the calculated deviations. The deviations related to the test results can be explained by the variation of the constructive parameters (uniform arrangement of the resin, homogeneous mechanical properties) as well as by the influence of the spectral components of the shock stress. It is expected that with the optimization of the technology of making the centrifugal compressor rotor from composite materials, the deviations between the theoretical and experimental results will be reduced.

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