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Hybrid method for studying the effect of the material change on the blade vibration behavior

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ABSTRACT

To increase the production efficiency of a typical turbine blade, it is necessary to change its material and hence its production technology. In this respect and in order to make sure that new material will not bring about the blade resonant vibrations, the common practice is to use the Campbell diagram approach. This approach only indicates the potential dangers for blade forced vibration. The purpose of this paper is to demonstrate the efficiency of a proposed hybrid procedure for the development of a reliable Campbell diagram, using an updated finite element model and a simplified test setup.

1. Introduction

High Cycle Fatigue (HCF) is known to be one of the major causes of blade failures in gas turbine operations. The airflow through a turbo-machine is inherently unsteady and has been recognized as the primary source of the blade vibration.

Turbine blade vibration is composed of three major sources: material (due to inherent material properties), structural (due to the frictional contacts), and aerodynamic (due to the motion-induced unsteady pressure around the blade). The design of aero gas turbine blades is an iterative process

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between structural and aerodynamic departments. The identification of vibration behavior of blades is one of the most important parameters in turbo machinery design. Other parameters such as operating temperature, geometry, operation rotational speed ,and material are also very important in blade cycle life and affect it by modifying blade modal behavior[1], which are discussed in this study.

One of the major tools in prevention of blade resonant vibration in design stage is the Campbell diagram. In order to develop such diagram, we have to either use full experimental setup measuring the natural frequencies of the blade in its operating conditions[2], or resort to a finite element based analysiss to extract such frequencies[3].

While the first option is accurate and expensive the second option is less accurate but relatively cheaper. In this paper a hybrid method will be introduced for Campbell diagram development that is based on a finite element model which is dynamically updated, using experimental data. The FE model that is updated and validated in two non-rotating and rotating stages, will be then used for Campbell diagram generation. The concept of the hybrid method is described below.

2. Hybrid Process Description

The process begins with generating the FE model of the blade and performing an analytical modal analysis on the non-rotating, non-heated blade. In parallel, and to validate (and update) the FE model, an experimental modal analysis will be performed on the blade.

The comparison of the two sets of results, i.e. experimental and analytical, will be used to update the FE model using an Inverse Eigen Sensitivity Method (IESM)[4].

The updated FE model is then used to calculate the modal parameters of the rotating, heated blade which will eventually lead to the development of the Campbell diagram. Before proceeding with the updated FE model for this purpose, we must make sure that the FE model that has been updated using non-rotating state modal parameters, is identically valid for rotating state of the blade, i.e. the same (or comparable) degree of correlation exists between experimental and analytical modal parameters for rotating state as it was for non-rotating state.

To achieve this goal, a test setup was designed that enabled the measurement of the blade natural frequencies in a rotating state, Fig. 7. The test setup was also modeled in the FE environment, Fig. 11

3. Prototype blade characteristics

The prototype blade that is considered for analysiss and test belongs to the second stage of a gas turbine and is shown in figure1.

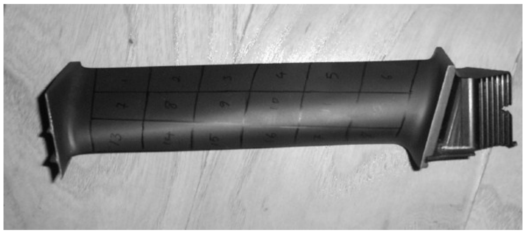


Fig 1. Prototype blade

Two different materials were considered for blades in this study, namely 1- Nimonic 115 super alloy used as original blade material, and 2- Inconel738LC used as the substitution material. Also, the original blade mass is 320 gr. Material properties of the two super alloys are given in table 1.

Table 1. Material properties

At 20 °C	$E\ (Gpa)$	$\rho\ (gr/cm^3)$
Nimonic115	212	7.86
Inconel738LC	200.6	8.11
At 700 °C	$E\ (Gpa)$	$\rho\ (gr/cm^3)$
Nimonic115	174	7.63
Inconel738LC	164	7.99

At 538 °C	$\alpha(10^{-6}/{}^\circ C)$
Nimonic115	13.3
Inconel738LC	13.95
At 871 °C	$\alpha(10^{-6}/{}^\circ C)$
Nimonic115	16.4
Inconel738LC	200.6

Regarding the operational condition of the blade, there is a special procedure for turbine warming up as follow:

The turbine starts at 100 ^{o}C at rotational speed of 1300 rpm, and then the temperature and rotational speed are increased up to operational points. Operational temperature and speed are 700 oC and 7000 rpm, respectively.

4. Analytical modal analysis (Static)

The first step in the analytical modal analysis is to develop the blade model in the FE environment. This was done by using a geometrical model created using CMM data of the blade, as shown in figure 2.

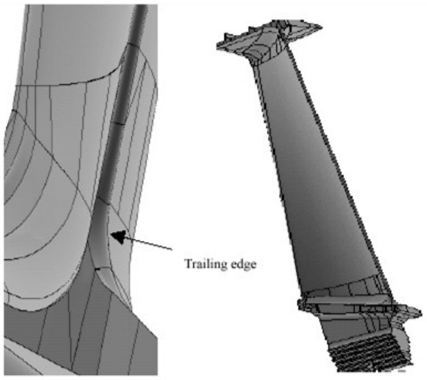


Fig 2. Geometrical model created by CMM data

The geometrical model was meshed and a modal analysis was performed, using the Nimonic material properties. The results for the 4 first modes are shown in figure 3.

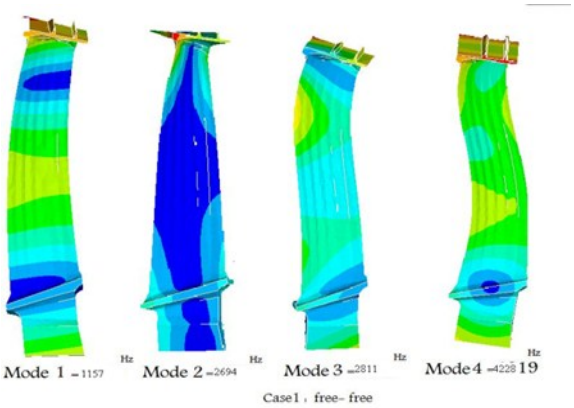


Fig 3. The 4 first mode shapes of the Nimonic blade

No. Of Elements 150880 & No. Of Nodes 31617

5. Experimental modal analysis (Static Test)

To extract the experimental modal parameters of the blade, the frequency response functions of the Nimonic blade were measured in 18 points in a free-free state. A typical measured FRF is shown in figure 4.

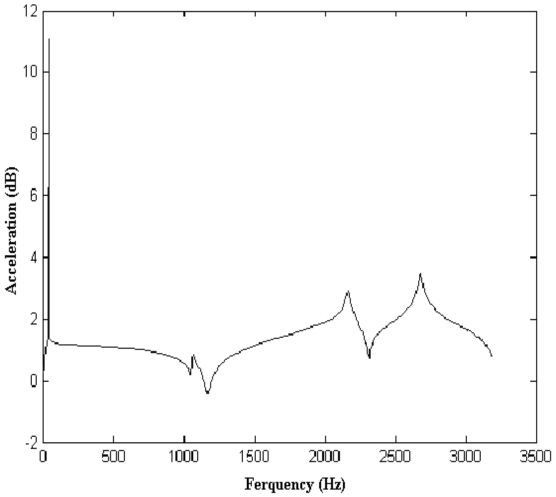


Fig 4. Measured Frequency response

Experimental modal analysiss were performed on the measured data and the results are presented and compared with analytical results in table 2.

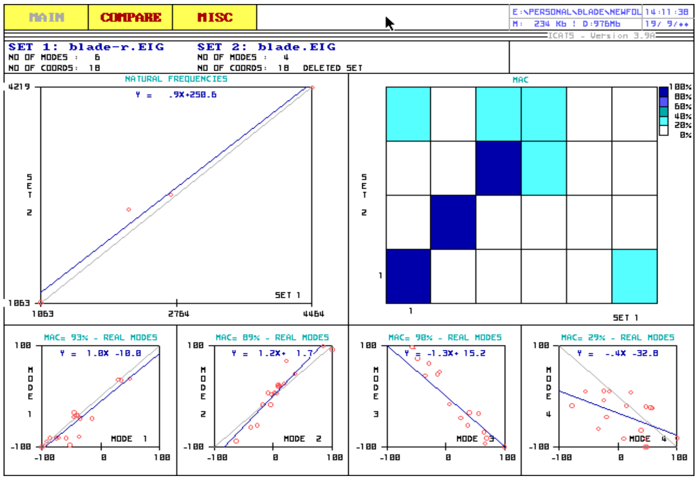
Table 2. Comparison between FE analysis and experimental results

	FEA Results [Hz]	Experimental Results [Hz]	Error%
1st mode	1070	1063	0.7%
2nd mode	2432	2162	12%
3rd mode	2649	2689	-1.5%
4th mode	4219	4464	-5.5%

Referring to table 2 good agreements between two groups of results can be observed. In the next step, we will try to further improve the correlation between the analytical and experimental results by updating the static FE model.

6. Finite Element Modal Updating

As mentioned above, before proceeding to the rotating state analysiss of the blade, it is necessary to update the FE model of the blade using the static (non-rotating) measured experimental modal parameters.

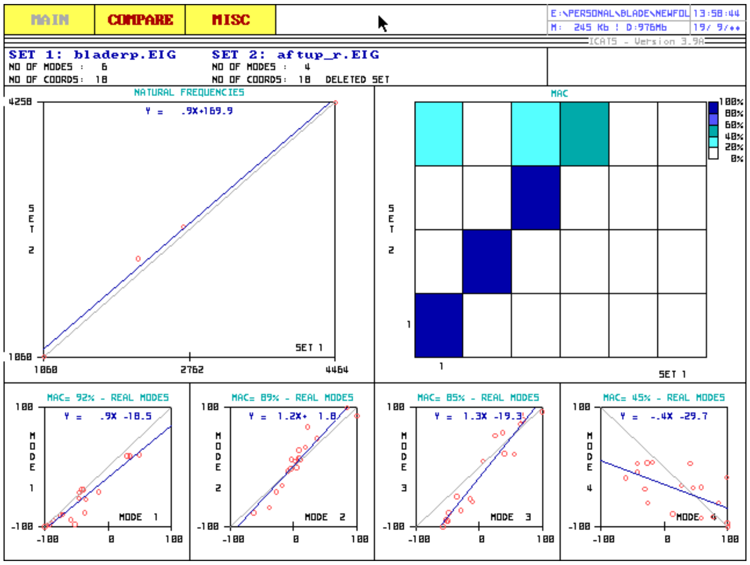


SET 1: blade-r.EIG
NO OF MODES : 6
NO OF COORDS: 18

SET 2: blade.EIG
NO OF MODES : 4
NO OF COORDS: 18 DELETED SET

Fig 5- MAC and modal vectors comparison –before updating

This was achieved by dividing the blade surface to 14 regions for which the modulus of elasticity and density were changed, in order to update the FE model. Figures 5 and 6 show the Natural frequencies, MAC and mode vector comparison for the FE model before and after updating, respectively. Also, table 3 represents the natural frequency and relative error values.



SET 1: bladerp.EIG
NO OF MODES : 6
NO OF COORDS: 18

SET 2: aftup_r.EIG
NO OF MODES : 4
NO OF COORDS: 18 DELETED SET

Fig 6- MAC and modal vectors comparison –after updating

As is evident from figures 5 and 6 and Table 3, the correlation between analytical and experimental results is increased and improved. The 4th mode natural frequency error is not decreased much, however the updating process has improved the modal vector correlation of this mode from 32% to 50%, as can be seen in the MAC tables.

Table 3. Comparison between FE analysis and experimental results- after updating

	FEA Results [Hz]	Experimental Results [Hz]	Error%
1st mode	1059	1063	-0.4%
2nd mode	2293	2162	6%
3rd mode	2695	2689	0.2%
4th mode	4258	4464	-4.6%

7. Validation of the updated FE model in the rotating state

To derive the Campbell diagram for the blade, it is necessary to perform the modal analysis on the updated FE model of the blade in a heated, rotating state, similar to its operational condition. Before undertaking this task, it is necessary to demonstrate that the FE model updated in the static condition is identically valid in rotating conditions too. If this validity is established then it will mean that Coriolis effects on the rotating blade are negligible, as these effects will render the statically updated FE model inaccurate for the case of the rotating model [5]. To achieve this, a test setup was designed as shown in figure 7. This test setup will enable us to rotate the blade in a lab condition and measure its response to excitation applied to the blade. Exciting the blade in a rotating state was a real challenge and was achieved by hitting the blade with a flexible hose that has copper tip attached to it. Also, the response of the blade was measured using an accelerometer that was attached to the bearing case(ball bearing), figure 7, To eliminate the effects of background noise coming from the rotating system, the appropriate windows,and trigger values were set on the B&K 3550, FFT analyzer. The excitation and response time signals, as well as auto spectrums of the response, are shown in figures 8 to 10.

Fig. 7. Experimental modal analysis test rig

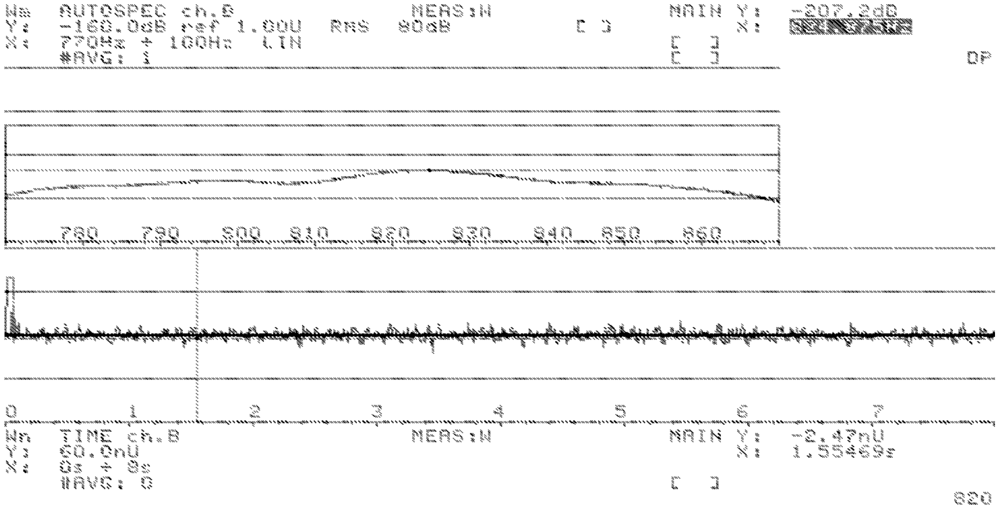
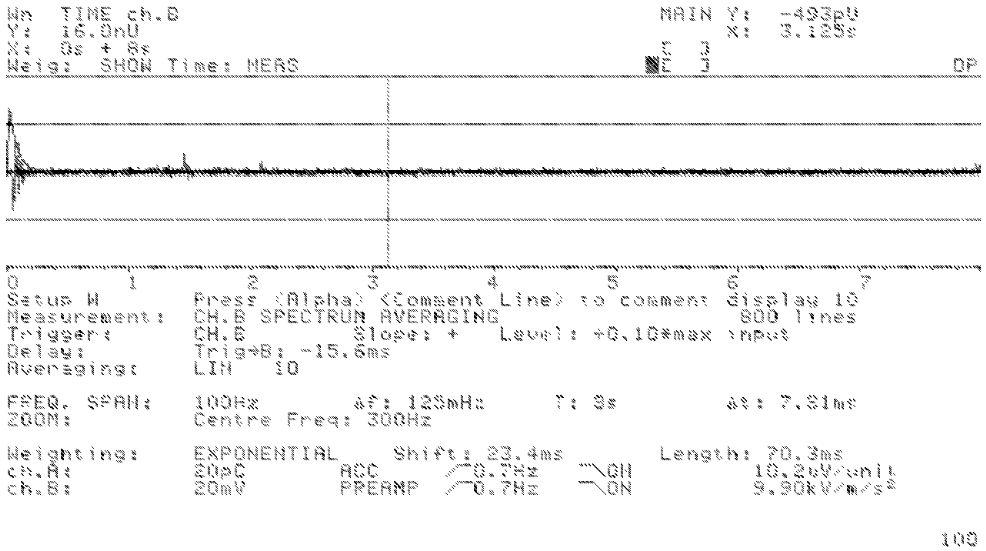


Fig 8- The response auto spectrum in 1500 rpm and the mode detected at 825 Hz. (above)

The response signal with a transient window (below)



```
Wm  TIME ch.B                                        MAIN Y:  -493pV
Y:  16.0nV                                                X:  3.125s
X:  0s + 8s                                          [  ]
Weig:  SHOW Time: MEAS                               █[  ]                      DP
```

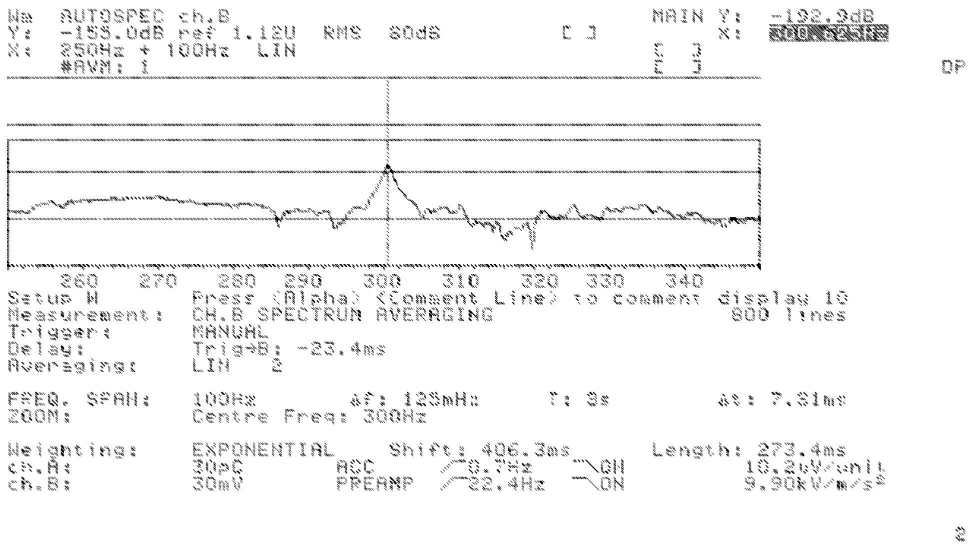
```
Setup W          Press <Alpha> <Comment Line> to comment display 10
Measurement:     CH.B SPECTRUM AVERAGING                   800 lines
Trigger:         CH.B        Slope: +   Level: +0.10*max input
Delay:           Trig→B: -15.6ms
Averaging:       LIN   10

FREQ. SPAN:      100Hz       Δf: 125mHz      T: 8s         Δt: 7.81ms
ZOOM:            Centre Freq: 300Hz

Weighting:       EXPONENTIAL    Shift: 23.4ms        Length: 70.3ms
ch.A:            20pC        ACC    ⁄‾0.7Hz   ‾\GH            10.2uV/unit
ch.B:            20mV        PREAMP ⁄‾0.7Hz   ‾\ON            9.90kV/m/s²

                                                                      100
```

Fig 9- The response signal and exponential window - 1500 rpm



```
Wm  AUTOSPEC ch.B                                    MAIN Y:  -192.9dB
Y:  -155.0dB ref 1.12V  RMS  80dB          [ ]            X:  [illegible]
X:  250Hz + 100Hz  LIN                               [  ]
    #AVM: 1                                          [  ]                      DP
```

```
Setup W          Press <Alpha> <Comment Line> to comment display 10
Measurement:     CH.B SPECTRUM AVERAGING                   800 lines
Trigger:         MANUAL
Delay:           Trig→B: -23.4ms
Averaging:       LIN   2

FREQ. SPAN:      100Hz       Δf: 125mHz      T: 8s         Δt: 7.81ms
ZOOM:            Centre Freq: 300Hz

Weighting:       EXPONENTIAL    Shift: 406.3ms       Length: 273.4ms
ch.A:            30pC        ACC    ⁄‾0.7Hz   ‾\GH            10.2uV/unit
ch.B:            30mV        PREAMP ⁄‾22.4Hz  ‾\ON            9.90kV/m/s²

                                                                        2
```

Fig 10- Response auto spectrum and the mode detected at 300 Hz.

In the next step, the FE model of the hub used in the test (which will be referred to as experimental hub hereinafter) was developed and integrated with the updated FE model, as shown in figure 11. It should be mentioned that the experimental hub was designed such that its first natural frequency was 4 times the first natural frequency of the blade and, as such, it was relatively rigid compared with the blade and did not interact with natural modes of the blade, i.e. the natural frequencies of the integrated blade-hub-shaft model were very close to those of the clamped blade model.

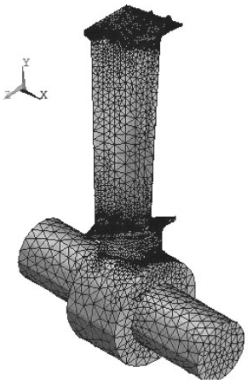


Fig 11. Updated blade, disk ,and shaft FE model

The results of the modal analysis performed on the "updated blade-hub-shaft" model in 1500 rpm are demonstrated in table 4.

Table 4- Comparison of the experimental and analytical modal parameters for the rotating state in 1500 rpm

4th mode (Hz)	3rd mode (Hz)	2nd mode (Hz)	1st mode (Hz)	
1840	1552	835	299	Blade+disk+shaft-Analytical
-	-	828	300	Blade+disk+shaft-Experimental

As is evident from table 4, there is a very good correlation between the FEA results and experimental results. Comparing the results of table 4 with the results of Table 3, it is observed that there is a better correlation between second natural frequencies in the rotating state than for the non-rotating state. The reason can be attributed to the fact that in a rotating state the root of the blade is almost fixed and has a lot less effect on the analytical results.

Now that the FE model is validated in a rotary state, one can proceed to the Campbell diagram derivation phase.

8. Campbell Diagram for the Blades with Original and Changed Materials

As explained above, the updated and validated FE model is now used to derive the Campbell diagram for the blade in original-material and changed-material states, as shown in figure 12.

As is evident from figure 12, mode 8 of the blade with Inconel 738 is close to the first order nozzle passage frequency and since the difference between these two frequencies are less than 5% (about 4%), further detailed studies might be necessary to prevent future problems.

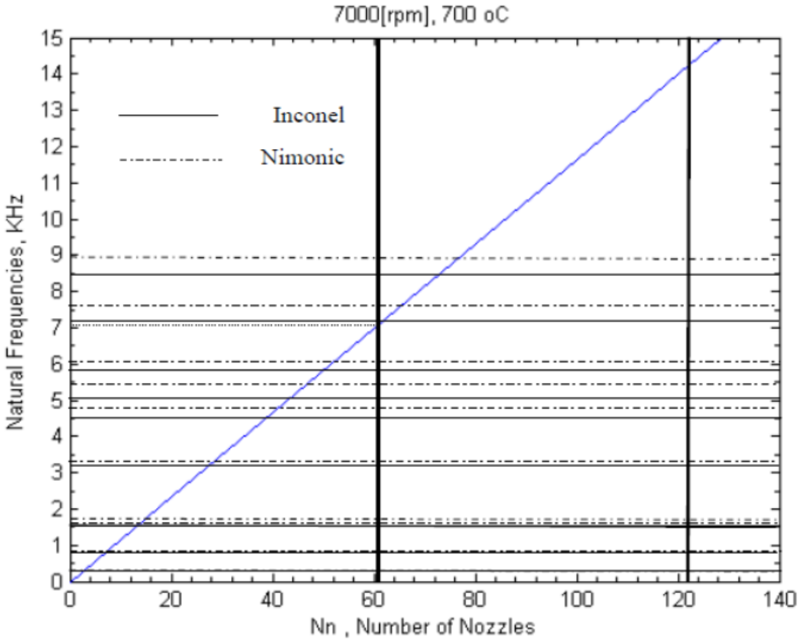


Fig. 12- Campbell diagram for the blade with original and changed materials

9. Conclusions and Remarks

In this paper, the potential danger associated with the changing of the material of a blade was studied. In this respect, a relatively cheap method was introduced for the derivation of the Campbell diagram for a turbine blade. The method which is based on using an updated FE model of the blade was validated using typical second stage turbine blade. Using the suggested method will enable us to derive highly reliable Campbell diagrams without resorting to very expensive test setups.

10. Acknowledgment:

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