MEC327

Prof. Elizabeth Cross

Comparative Analysis of Steam Turbine Dynamics An Executive Summary

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Contents

1 Analysis

For this analysis, 3 blades were assessed and their characteristics computed, with blade 1 being the provided blade data, blade 2 being the blade data captured in the lab, and blade 3 data being the personalised blade data provided.
The technical analysis of the blade characteristics begins with an observation of the data gathered during the testing, and processed by the MATLAB program. Natural frequency, frequency response functions (visualising the natural frequencies), damping ratio, and modal assurance criteria (MAC) were all computed from the test data in MATLAB, and can be seen below.

Blade 1 Natural Frequency	Blade 2 Natural Frequency	Blade 3 Natural Frequency
120.25	117.25	112
270.25	272.5	361.25
372.25	375.75	487.25
489.25	490.25	650.5
644.5	646.75	731

Table 1: Natural Frequencies for each Blade

The natural frequencies of each blade can be seen, with only minor differences between each. One notable characteristic of blade 3 is the wider range of natural frequencies, with the first being lower than that of the two other blades, but then the natural frequencies occur at higher frequencies than that of the other blades. Data for blade 1 and blade 2 are very similar in this respect, when compared to blade 3.
This effect can be observed better visually, when observing the FRFs of the 3 blades.

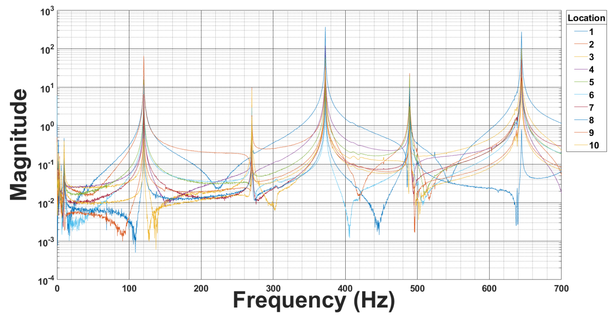

Figure 1: Frequency Response function for Blade 1

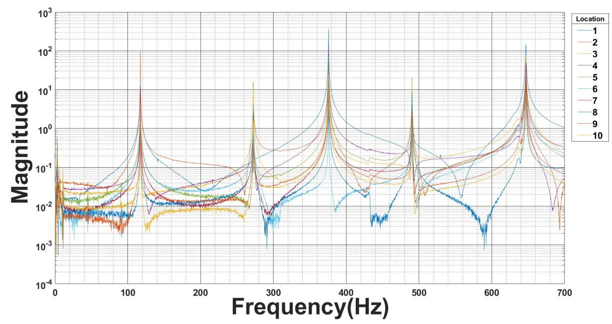

Figure 2: Frequency Response function for Blade 2

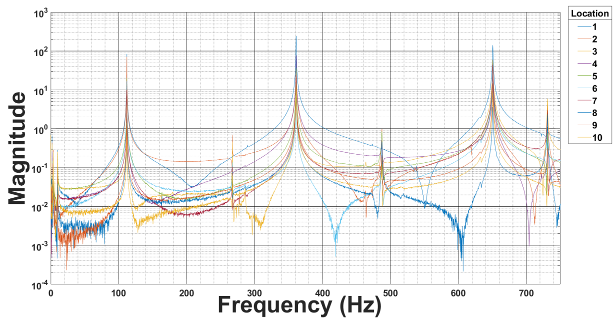

Figure 3: Frequency Response function for Blade 3

The observed peaks being the presence of a natural frequency of the blade where large oscillations can be observed regardless of excitation position. For blade 3, the fifth natural frequency appears to occur at 731 Hz, as verified by the FRF plot.

Blade 1 Damping Ratio	Blade 2 Damping Ratio	Blade 3 Damping Ratio
0.0013000	0.0007402	0.0007414
0.000400	0.0009183	0.000833
0.000300	0.0002771	0.0002773
0.000300	0.0002189	0.0002154
0.000300	0.0004717	0.0004634

Table 2: Damping Ratios for corresponding natural frequencies for each Blade

Collection of mode shape data was performed using mode shape analysis functions, and MAC algorithms to find similarities between two mode shape matrices.

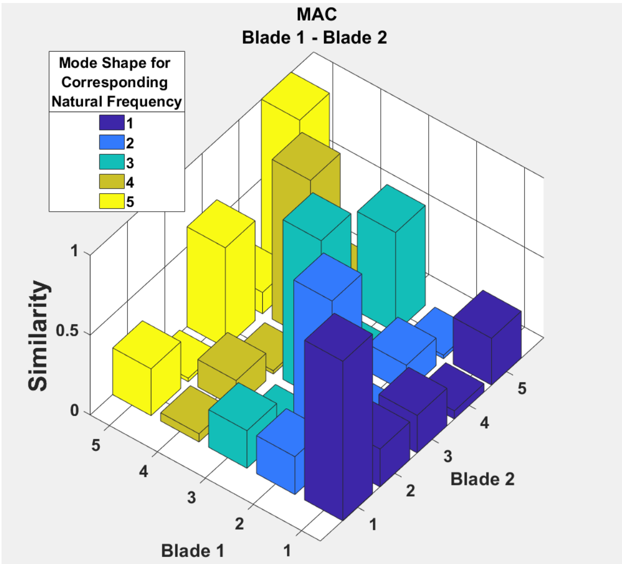

Figure 4: MAC for Blades 1 and 2

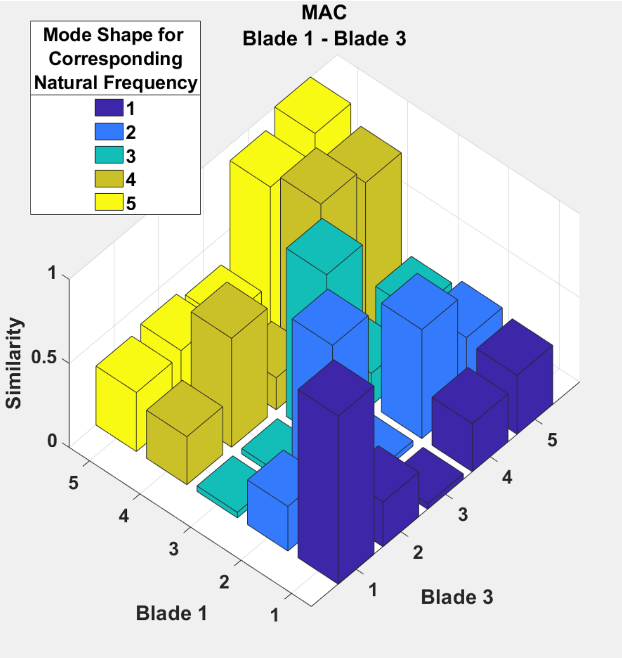

Figure 5: MAC for Blades 1 and 3

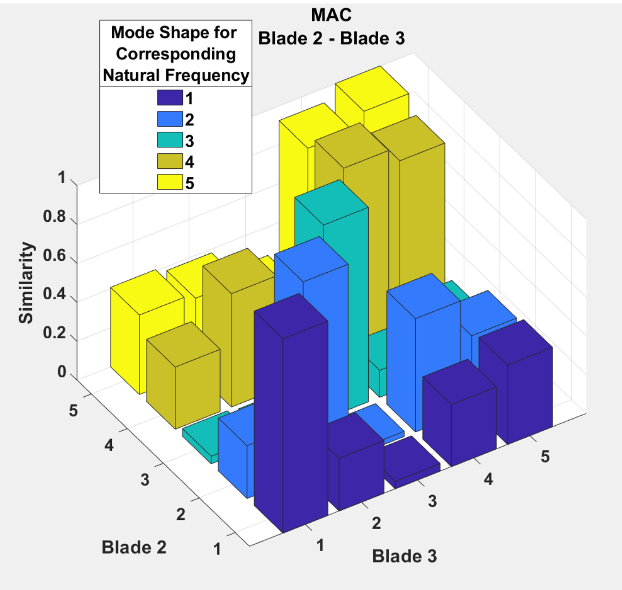

Figure 6: MAC for Blades 2 and 3

The data presented was plotted using an existing function written by Farshchin, Ph.D (2015). Use of function provided is Lawful and following producers license - The corresponding raw data for each modal assurance criteria study is included in appendix 1.

The data demonstrates that the blades are responding to excitation in very similar modes, and are dynamically comparable. This is because the similarity of corresponding mode shapes between blades is close to or equal to 1. The similarity between differing mode shapes of different blades is relatively low, and shows little correlation, with the exception of blade 3, where the 4th and 5th modes are similar to both the 4th, and 5th modes of the other blades. The reason for this may not be known at this point, as the MAC is independent of the frequency at which a component is oscillating, only the mode shape itself. Typically, MACs are used to compare experimental results with simulated Finite Element Analyses, to validate or improve computational study, but are justified in this case due to the direct comparison of seemingly identical turbine blades. See discussion (section 3) for further justification for such an analysis. The SDOF validation for the 3 blades can be seen in appendix 2.

2 Dynamic Properties

The large discrepancy in natural frequency at the third mode between blade 3 the others is indicative of a 'missing' natural frequency, as the subsequent natural frequencies lie close to that of the succeeding in blades 1 and 2. (i.e. the 3rd natural frequency of blade 3 lies close to the 4th natural frequency of blades 1 and 2, the 4th lies close to the 5th, and so on). Most likely, this is caused due to the accelerometer on blade 3 (measuring excitation) being positioned at a node of the third mode shape. This will cause an apparent 'ghost' frequency, where no response is measured. This claim is supported by the data obtained by the MAC analysis, as they indicate very little difference between the second mode shape of the blades, indicating the different second mode response is not caused by an inherent dynamic or geometric difference, and instead by measurement error. This is also supported by the very low modal similarity of the second mode of blade 3, and the third mode of blades 1 and 2, suggesting that the mode at the second observed natural frequency is in fact not the second mode shape of the blade.

Another large discrepancy can be seen in the damping ratio data for blade 1 when compared to blades 2 and 3. Unlike the others, blade 1 has a relatively consistent damping ratio, with a significantly higher value at the first mode, and then relatively similar values for higher modes. The steady-state behaviour of the blade 1 damping ratio at higher frequencies indicates that the damping behaviour of the blade is independent of resonating frequency, and instead relies on another, potentially external, factor. Reasons for this may be the mounting method for the blade, with higher mounting pressures at points of contact perhaps stiffening sections of the blade at lower frequencies, and exacerbating excitation at higher frequencies - indicative of a greater stiffness. A lower stiffness will reduce the dynamic performance of a component at low frequencies, and then begin to act as an isolator for the mass at higher frequencies, the opposite of which is observed for blade 1.

3 Discussion

The importance of dynamic analysis is clearly demonstrated here, as the physical dynamic properties of a component can be measured, which can be useful for applications in industry. In the context of turbomachinery, this demonstrates the importance of resonant analysis, as manufacturers can observe and document the frequencies at which a turbine blade may be more likely to fail, due to higher stresses experienced at resonance. Yet, the most significant justification for the study of modal analysis presents itself when the MAC study of blade 3 is taken into consideration. Aligned with what has been established, engineers may consider the third blade for use at 270Hz, as at first it appears other blades would deteriorate at a faster rate, and so blade 3 is a favourable choice. **However**, through the MAC analysis, it can be seen that the blade does in fact resonate at that frequency, and so the blade is no less likely to fail than the others. This highlights the value of dynamic analysis for industry, as it leads to avoiding potential loss of money and life, from turbine blade failure.

4 Postscript

4.1 Appendix 1

MAC Tabular Data

			Blade 1		
	1.0000	0.2373	0.2355	0.0512	0.2929
	0.2373	1.0000	0.0111	0.1720	0.0219
Blade 2	0.2355	0.0111	1.0000	0.0208	0.6259
	0.0512	0.1720	0.0208	1.0000	0.1334
	0.2929	0.0219	0.6259	0.1334	1.0000

			Blade 1		
	1.0000	0.2652	0.0327	0.2854	0.3508
	0.2652	1.0000	0.0316	0.6474	0.3773
Blade 3	0.0327	0.0316	1.0000	0.1897	0.3280
	0.2854	0.6474	0.1897	1.0000	0.9039
	0.3508	0.3773	0.3280	0.9039	1.0000

			Blade 2		
	1.0000	0.2718	0.0392	0.3194	0.4086
	0.2718	1.0000	0.0265	0.5829	0.3808
Blade 3	0.0392	0.0265	1.0000	0.1390	0.2536
	0.3194	0.5829	0.1390	1.0000	0.9241
	0.4086	0.3808	0.2536	0.9241	1.0000

Figure 7: MAC Tabular Data

4.2 Appendix 2

SDOF Verification

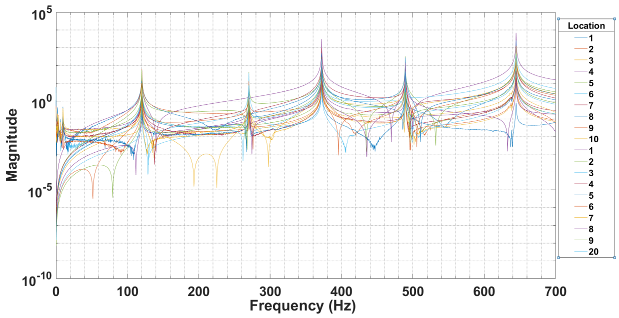

Figure 8

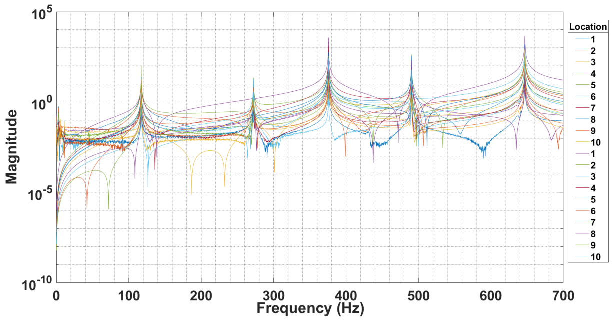

Figure 9

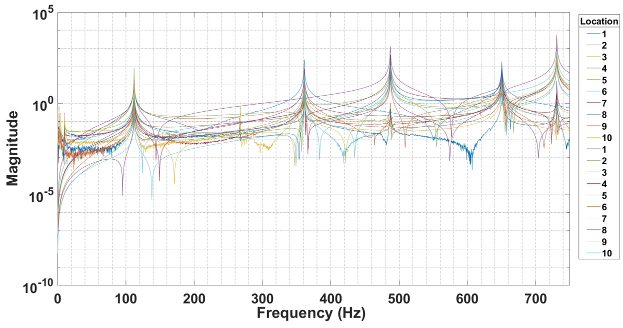

Figure 10

References

[Code for Figures 4, 5, 6] Copyright (c) 2015, Mohammad Farshchin, Ph.D, University of Michigan (2015), Function 1 pp.l1-l16.

[Blade Data] Cross, E. (2022). MEC327 Coursework – University of Sheffield.