

## Chapter 2

# Experimental Non-linear Modal Testing of an Aircraft Engine Casing Assembly

Dario Di Maio, Paul Bennett, Christoph Schwingshackl, and David J. Ewins

**Abstract** This paper aims to present experimental work on an aircraft engine casing assembly. Nowadays single components of casings can be modeled with such high accuracy that they can be validated by carrying out the model validation process using measured data from a sector of the entire assembly. This smart validation process can be achieved by carrying out the modal analysis with a Scanning LDV (Laser Doppler Vibrometer) system which allows good spatial resolution of the measured mode shapes. The validation process can be assumed valid under linear response conditions obtainable for low vibration amplitudes. Casings are typically connected together by joints which may or may not respond non-linearly under high levels of vibration. Therefore, prior to conducting any non-linear validation, the mode(s) responding non-linearly must be identified beforehand in order to correctly specify the non-linear modal testing required. The work presented here will use a large civil engine casing assembly comprising a Combustion Chamber Outer Casing (CCOC), High Intermediate Pressure Turbine Casing (HIPTC) and Low Pressure Turbine Casing (LPTC.) The Fine Mesh Finite Element Model (FMFEM) was successfully validated using linear modal analysis test data. One of the objectives of this work is to define the key points for conducting non-linear modal testing of such large casing assemblies and sub-assemblies. One outcome of the experimental work was a set of recommendations for performing measurements, which should be carried out within the frequency bandwidth selected during the model validation process. Experimentally derived non-linear response curves are presented in this paper.

**Keywords** Joints • Non-linear testing • Aircraft engine casing • Scanning LDV

## 2.1 Introduction

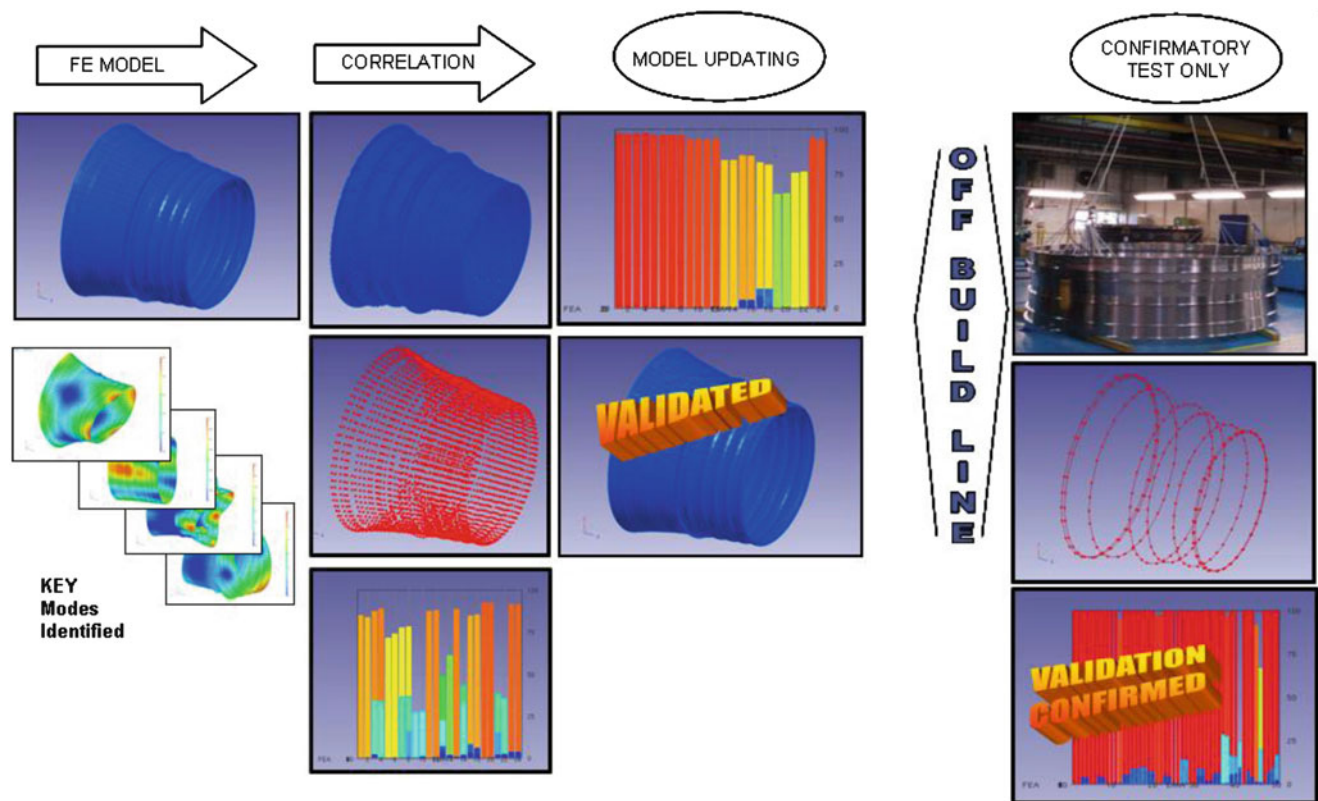
A civil aircraft engine casing assembly comprising a Combustion Chamber Outer Casing (CCOC), High Intermediate Pressure Turbine Casing (HIPTC) and Low Pressure Turbine Casing (LPTC) Fine Mesh Finite Element Model (FMFEM) was successfully validated using linear modal analysis test data [1] and (Private communication University of Bristol-Rolls-Royce). One of the objectives of this project was to define guide-lines for conducting non-linear modal testing of such large casing assemblies and sub-assemblies. The key aim was to produce a set of recommendations for performing measurements, which should be carried out within the frequency bandwidth selected during the model validation process. The validation process of the full assembly was performed using two measurement methods: (i) accelerometers and a roving instrumented impact hammer and (ii) using a Scanning Laser Doppler Vibrometry (SLDV) system and electromagnetic shaker. The first validation was carried out with the casing assembly supported vertically by bungees, whereas the second was tested with the assembly suspended horizontally.

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**Fig. 2.1** Schematic of a validation of an assembly model

Non-linear testing was initially started with the casing suspended horizontally. It was later decided to rotate the test structure and continue testing with it suspended vertically because some of the natural frequencies obtained from the horizontal tests had shifted slightly.

The assembly has over 50 modes below 1 kHz which makes selection of the most suitable modes for non linear testing difficult.

The testing was divided into five distinct stages

- (i) Linear FEM model validation
- (ii) Mode shape selection
- (iii) Qualitative assessment of non linearity using force control method (low excitation levels)
- (iv) Quantitative assessment of non linearity of selected resonances using force control method
- (v) Higher amplitude modal testing using amplitude control method of those modes identified as non linear.

Using this approach those resonances likely to exhibit the strongest non linear behaviours can be quickly shortlisted. For the assembly the bolted flanges are responsible for most (if not all) of the non linearity of the structure. The testing presented here was focused mainly on excitation of modes with significant casing flange response.

For some engine components, the 'physics' of the undamped FE model are completely represented when analysed in a Free-Free state (i.e. no boundary conditions or external loads applied). In other words, the accuracy of the normal modes prediction of such FE models (i.e. the natural frequencies and mode shapes) are most strongly dependant on the fidelity of the FE model geometry, the material properties used, selection of element type(s) and the degree of FE mesh refinement employed. In fact, for lightly damped, solid, isotropic components at room temperature, the accuracy of the FE model normal modes analysis, for all practical purposes, depends only on these parameters for low levels of excitation.

Early on in the development program, prior to the manufacture of hardware, only nominal geometry models are usually available. Even so, it is clear that if FE models of high geometric fidelity and mesh refinement are generated (i.e. FEMs) the resulting predictions of frequencies and mode shapes can be used, well in advance of any Modal Testing, to help validate the less refined and more idealised Whole Engine Models (WEM) required early on in an Engine Development Program. The effort required to produce FEMs is now considerably less than might be expected (typically only 2–3 days) depending on the component complexity and the quality of the 3D geometry initially provided. Figure 2.1 shows how a validation of a

linear WEM assembly model can be performed very quickly using FMFEM models where modal testing plays an important part in confirming the quality of the validation process.

At this time it is felt that some limited, confirmatory testing, of the actual component hardware would still be advantageous. This would primarily be to confirm that the test hardware supplied was close to its design intent. It has been shown recently that the use of new technologies such as Scanning LDV systems can be used to rapidly validate FMFEMs. Such short test times can be fitted neatly within modern development engine build programmes with minimal disruption. One of the key advantages of performing modal testing with an SLDV system is its ability to measure mode shapes with high spatial resolution very quickly. Visualization of the mode shapes helped to rapidly identify those resonances which were more likely to exhibit a non-linear response. Reviewing such high fidelity depictions of the assembly mode shapes was very useful for understanding which modes were likely to exercise the casing flanges, the greatest potential source of the assembly non-linearity. This approach was found to be extremely useful for the initial experimental assessment of the casing's likely non linear responses.

## 2.2 Test Planning for Rapid Validation of FMFEM Axisymmetric Casings and Assemblies

### 2.2.1 *Contrast with WEM Model Validation Testing*

For WEM models current validation processes require that all key modes are acquired during the modal test model correlation and to permit any subsequent Computational Model Updating (CMU) required later. When Test Planning for FMFEMs only those key modes need to be acquired which confirm, with high confidence, that the FMFEM closely represents the dynamic behaviour of the test component and so was a valid reference for previous validation of the WEM model.

Modal Test Planning and Validation of WEM models starts with the WEM model itself, however when a FMFEM model has already been used to 'validate' the less detailed WEM model earlier in the engine development programme then a smaller scale modal test can be planned. For example, Test Planning the validation of WEM models will usually make provision to acquire both orthogonal modes from each mode pair. However, when validating large, *strongly* axisymmetric FMFEM models usually only one orthogonal mode from each pair need be acquired on test, since a strong correlation with one of the orthogonal modes would imply a similarly strong correlation for the other. High confidence in the FMFEM also makes it very unlikely that the modal validation test data will need to be used as a reference for any CMU later.

### 2.2.2 *Reducing Modal Test Times*

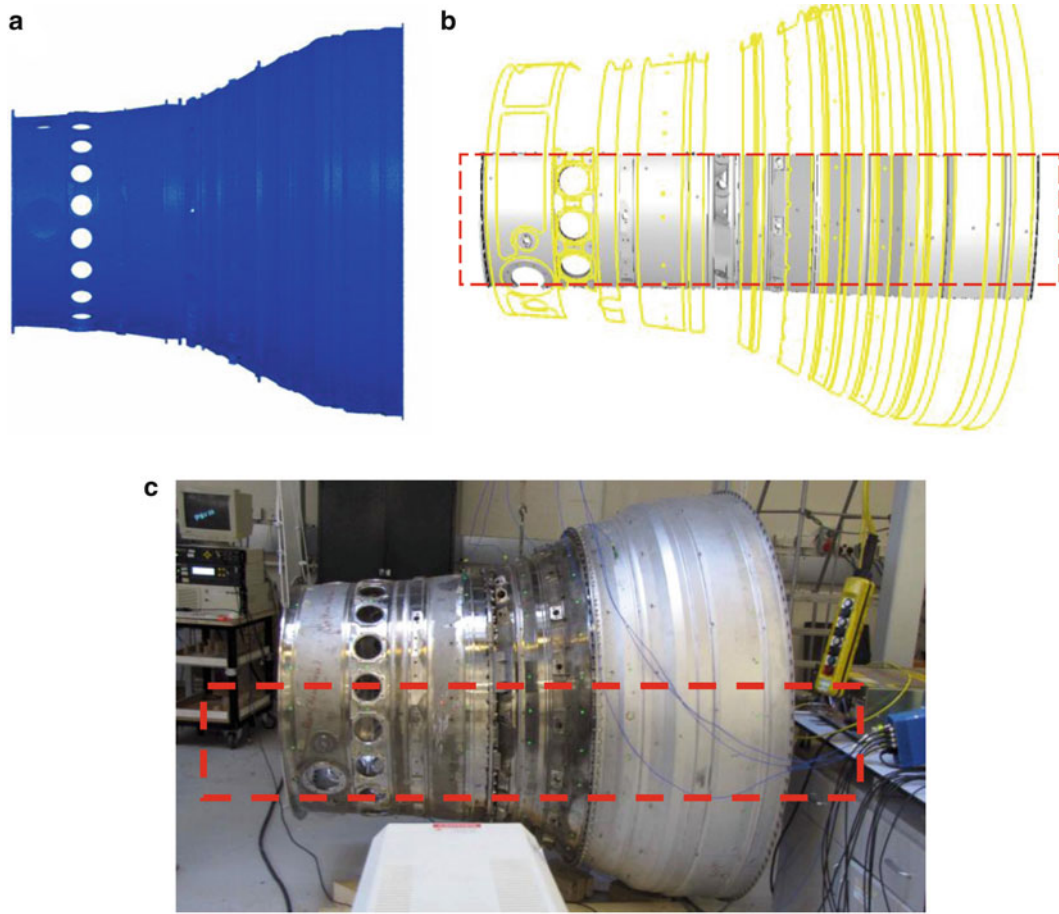
There are three main areas in which to speed up the validation testing required for FMFEMs. This section illustrates some suggestions to reduce the model validation test time required for FMFEM components and/or subassemblies from delivery of the hardware to the test area.

For large casing structures modal test setup times can still be lengthy despite the apparent simplicity of the Free-Free boundary conditions required. However, by using the FMFEM model it becomes possible to predict the test component's fundamental Free-Free mode and mass with high confidence. Using this information the number and stiffness of the suspension points required can be calculated in advance and can be made ready prior to the test hardware's arrival.

Correctly marking up the test hardware prior to modal testing is critical, as it locates the test positions carefully chosen during Test Planning onto the actual test hardware. Mistakes made during marking-up are often difficult to evaluate post test and will artificially reduce the correlation with the FMFEM model.

The large amount of geometric detail in FMFEMs can be utilised to simplify marking-up of the test hardware enormously. A large number of features (e.g. flange holes, casing holes, casing bosses, etc.) within the FMFEM can now also be accurately located on the test hardware itself. Higher initial confidence in the FMFEM also means that fewer test points are required for validation, i.e. additional test points that would otherwise have been specified to extract both orthogonal mode pairs or to aid mode visualisation are no longer required.

It will be shown that for large axisymmetric component casings (and subassemblies) test data will usually only be required from a sector of the test hardware in order to validate its corresponding FMFEM model with high confidence. Since only one side of the component needs to be tested, non-contact 'line of sight' dynamic measurement methods (e.g. SLDV) can



**Fig. 2.2** FE model (a), target sector (b) and test setup (c). (a) Mesh Finite Element Model (FMFEM) of the COC/HPTIPT/LPT casing assembly. (b) Target sector used for the Test planning of COC/HPTIPT/LPT casing assembly FMFEM. (c) Setup showing the target sector of the FMFEM on the real COC/HPTIPT/LPT casing assembly

be employed efficiently. When SLDV is used the acquisition can also be automated to give high density area scans. It is essential to confirm in advance, using Test Planning, that the chosen sector scan is likely to provide sufficient high quality data in order to validate the target FMFEM modes.

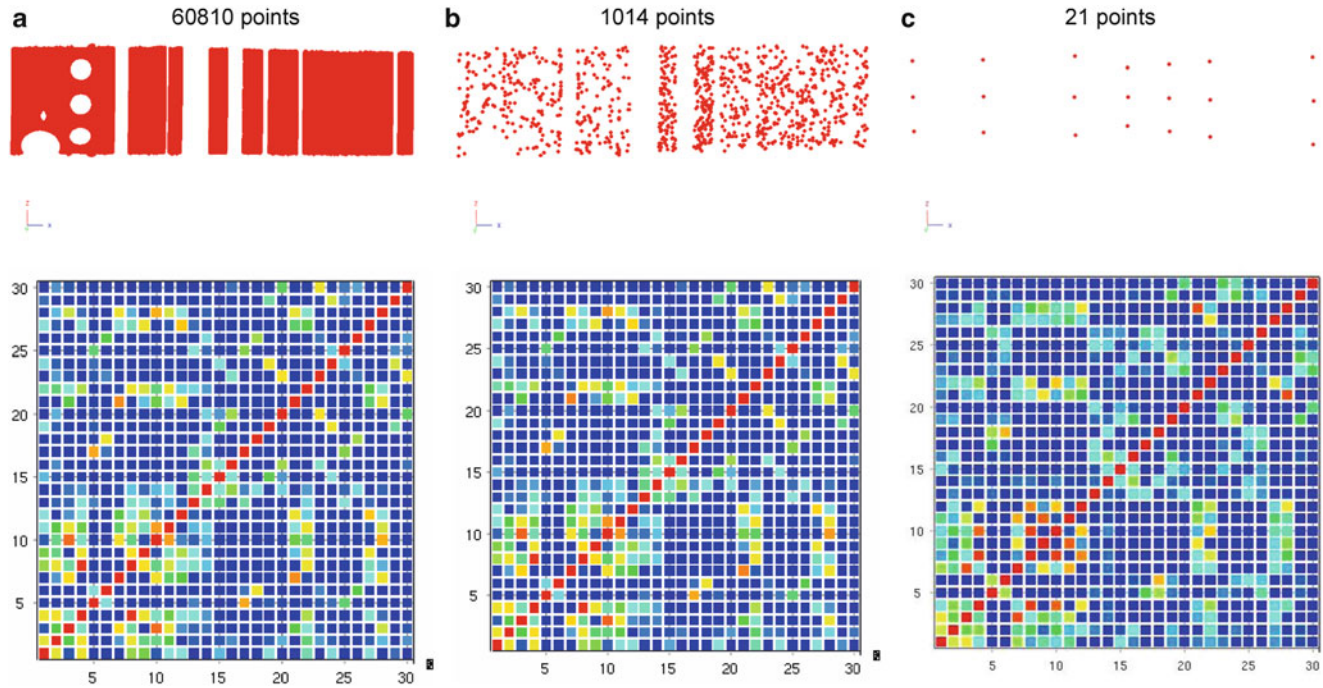
### 2.2.3 Sector Test Planning for Large Axisymmetric FMFEMs

Figure 2.2a shows a FMFEM model of a large Civil Engine casing sub-assembly. This FMFEM model has over 12 million degrees of freedom and the casing flanges have been modelled (simplistically) as rigidly fixed together. Using the Rolls-Royce in-house FE code it is possible to quickly ‘hide’ all but a sector of the full assembly model, Fig. 2.2b, and export selected nodal displacements from that sector only. Figure 2.2c shows how closely the geometry of the FMFEM sector matches the actual test hardware; a number of the same features can be clearly identified on both (e.g. casing holes, bosses, etc.). Note, not all sector faces are chosen for inclusion in Test Planning, highly angular faces (e.g. bosses, flange radii, etc.) have not been included and so appear uniformly dark blue (i.e. no displacement data exported) in Fig. 2.2c.

The first 30 FMFEM modes were included in Test Planning. The displacement data and sector FE mesh were combined and converted into a format that could be read directly into FEMtools [2]. Using FEMtools it was possible to quickly produce an autoMAC of the sector displacement patterns from the first 30 FMFEM modes. This was one of the key aims of Test Planning – to eliminate or minimise any spatial aliasing, i.e. to ensure that enough test points are included so that *all modes of interest* could be unambiguously differentiated.

In this way testing a ‘sector’ of the FMFEM was ‘simulated’ in order to determine how many test points might be required to clearly differentiate the first 30 modes of the sub-assembly on test. The autoMAC of a good test plan will have





**Fig. 2.3** Predicted variation in the autoMAC of first 30 modes of the FMFEM sector due to a reduction in test point density

low off-diagonal terms for all modes of interest. Figure 2.3 shows the effect on the autoMAC for a reducing number of test points assumed from (a) all 60,810 points exported from the FMFEM; to (b) a subset of 1,014 points; to (c) where only 21 points were selected. Assuming 60,810 test points is obviously impractical, however, the resulting autoMAC does provide a useful datum against which to assess how decreasing test point density might adversely affect the resulting test.

It can be seen that a reduction from 60,810 to 1,014 test points, a far more likely scan density for this structure, has had little detrimental effect on the autoMAC. A Test Plan is usually deemed satisfactory if the off diagonal terms of the simulated test autoMAC are less than 40% for all key modes unless they are already well separated ( $>25\%$ ) in frequency. In other words, no two modes on test should look too much alike unless they can also be differentiated (with high confidence) using frequency difference alone. The near identical autoMAC's of Fig. 2.3a, b show that using 1,014 test points would be functionally as good as using 60,810 points.

What may be a little more surprising is that the autoMAC from a very small (but carefully chosen) number of test points, i.e. the 21 points of Fig. 2.3c, is still reasonable. While there are now some obvious differences when compared with the larger tests, its autoMAC suggests that test data from just these 21 points would still lead to a satisfactory modal test.

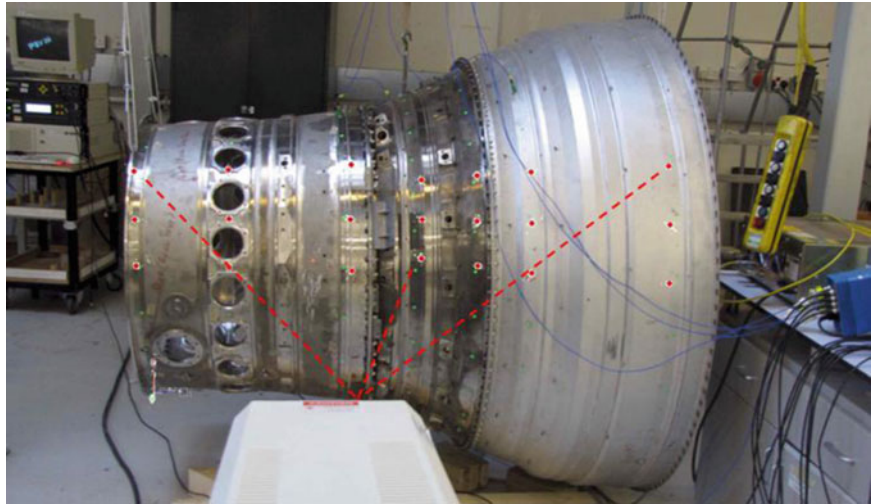
The number and location of the accelerometers required for the modal test should always be confirmed using Test Simulation. A key advantage for FMFEM validation testing is that it becomes possible, during Test Simulation, to confirm a number of suitable accelerometer positions which also take advantage of the very high geometry fidelity to the actual test hardware. Through careful Test Planning, the number and position of the test accelerometers required can be chosen such that they are simple to attach accurately, and very quickly, to the test hardware.

### 2.3 Test Case: SILOET 2.3.2: Full Assembly COC\HPTIPT\LPT FMFEM

In order to demonstrate the applicability of the above approach it was decided to conduct some rapid testing of the large Civil casing sub-assembly shown in of Fig. 2.2a, at the University of Bristol as part of the SILOET Research Program. The Test Planning and Test Simulation outlined in Sect. 2.2.3 were carried out by Rolls-Royce and an impact hammer modal test was conducted at the University of Bristol using just the 21 points of Figs. 2.3c and 2.10 off pre-selected reference accelerometer positions around the assembly. The location of these 21 points on the test hardware has been shown in Fig. 2.4.

The results of the testing are shown in Table 2.1 where it can be seen that there is strong correlation with the FMFEM predictions for at least 1 of each orthogonal mode pair up to mode pair 12. Table 2.1 compares the FMFEM correlation

**Fig. 2.4** Location of the 21 measurement points



**Table 2.1** Correlation of FMFEM model against modal test data. Comparison of rapid sector test FMFEM correlation with that of the full modal test

Mode pair	FMFEM mode	Hz	TEST mode	Hz	Full test Freq. diff. (%)	MAC	Rapid test Freq. diff. (%)	MAC
1	1	39.3	1	39.1	0.4	97.8	−0.9	91.3
	2	40.1	2	39.7	1.1	94.4	2.5	80.9
2	3	53.7	3	53.1	1.1	99.0	1.1	96.3
	4	53.8	4	53.2	1.2	99.2	1.1	89.1
3	5	58.7	5	58.5	0.3	97.4	−0.4	95.2
	6	58.7	6	58.9	−0.3	97.1	0.4	90
4	7	84.5	7	82.3	2.7	99.5	2.7	97.8
	8	84.6	8	82.4	2.7	99.1	2.7	94.4
5	9	129.3	9	125.7	2.9	98.8	2.8	96.1
	10	129.3	10	125.8	2.8	99.0	2.9	90.6
6	11	185.5	11	180.5	2.8	97.3	2.8	98.5
	12	185.6	12	180.6	2.8	96.9	2.8	79.3
7	13	201.8	13	187.7	7.5	85.9	7.5	92.2
	14	203.0	14	189.4	7.2	96.3	7.2	95.2
8	15	206.8	15	201.1	2.9	96.9	0.4	97.1
	16	209.9	16	206.0	1.9	97.3	4.4	96.4
9	17	241.2	17	229.8	4.9	92.2	4.9	94.8
	18	242.4	18	230.3	5.3	91.6	5.3	89.5
10	19	244.8	19	232.7	5.2	98.4	5.2	97.3
	20	245.0	20	233.2	5.0	93.1	5.1	90.9
11	21	249.6	21	242.6	2.9	98.9	2.9	96.6
	22	249.7	22	243.0	2.8	99.1	2.8	93.4
12	23	290.1	23	283.5	2.3	99.4	1.9	93.5
	24	290.5	24	284.8	2.0	99.4	2.5	96.2

results of the 21 point sector test and those of the larger 200 point whole annulus test. The 21 point ‘sector’ test and 200 point ‘full’ modal test results compare well in almost all cases, and very well for at least one of the orthogonal modes in each pair – which was the original aim of the Test Planning. For both tests mode pair 7, 9 and 10 are outside the current FMFEM acceptance criteria and this is likely due to the simplistic modelling of the flange joints in the FMFEM (fully fixed) causing it to over predict the frequencies for these modes. The advantage of SLDV scanning is that it can be completely automated and gives very high test point density (typically 1,000+ points). However, it generally uses a ‘contacting’ excitation system (e.g. a small Electromagnetic (EM) shaker) which can have a negative impact on the extraction of the orthogonal mode pairs of the large axisymmetric components/sub-assemblies and also on their measured frequencies (due to mass attachment effects).

## 2.4 Test Setup for Non-linear Large Assembly Tests

### 2.4.1 Preliminary Measurements

The initial assessment of the casing assembly's non linearity was started with it suspended horizontally. The ten off reference accelerometer positions used during the linear validation of the assembly were also used for the non linear testing to provide continuity. Figure 2.5 shows the initial non linear test set-up, and the LDS V200 electromagnetic shaker (20 N dynamic range) attached to the HPIPT of the casing.

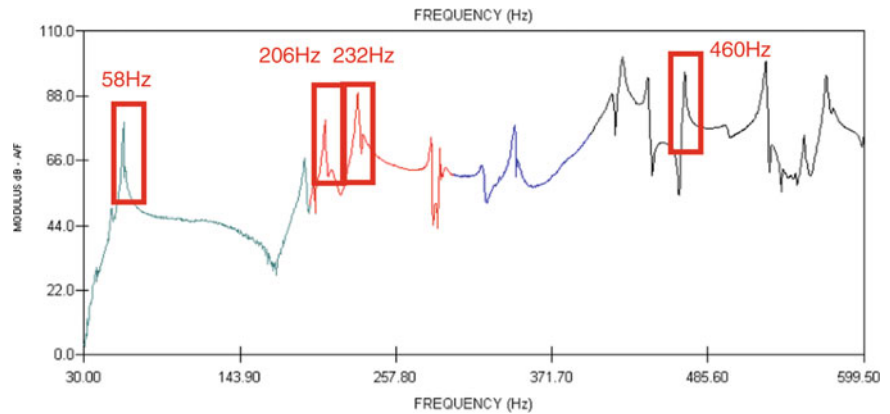
Although a commercially available system was used for the non-linear testing, a customised acquisition panel was created in LabVIEW (a National Instruments software package) to support the initial investigation. The test data acquired was analysed using the ICATS modal analysis software suite [3].

A broadband sine step excitation (500 Hz bandwidth) was selected to excite a pre-chosen subset of the casing resonances for non linear assessment. Figure 2.6 shows in red boxes the resonances selected at 58, 206, 232 and 460 Hz, respectively. Neither the force nor amplitude was experimentally controlled but instead, as a first pass check, the selected resonances were excited at increasing levels in order to visually detect any changes in the resulting FRFs, that might be suggestive of underlying non linearity.

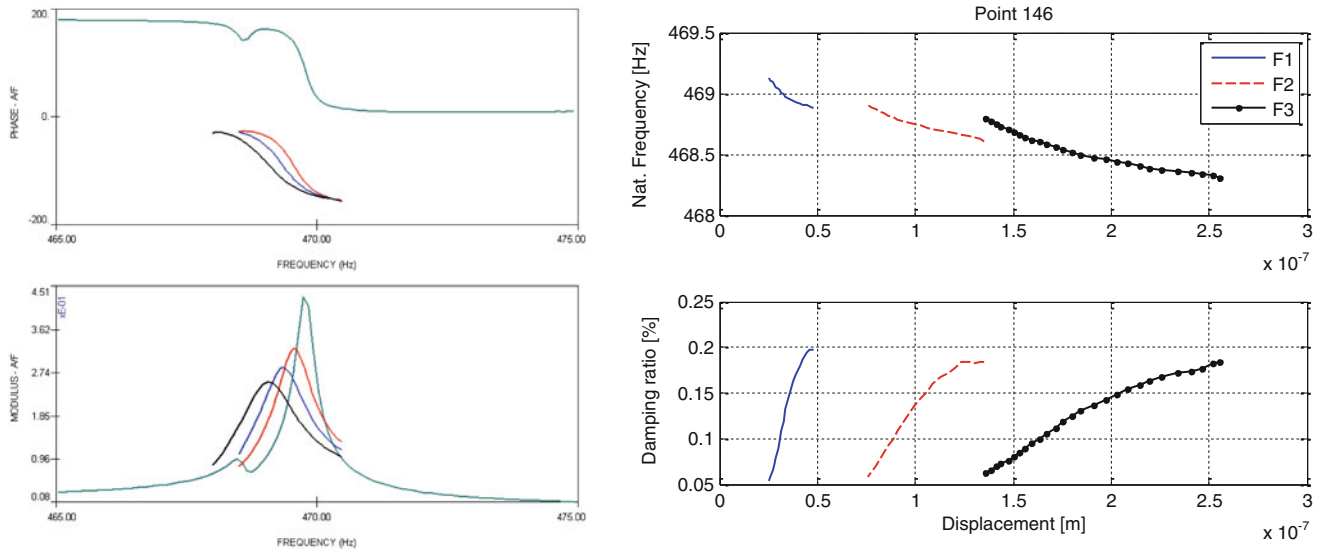
As a quick first pass check, neither the force nor amplitude was experimentally controlled but instead the selected resonances were excited at increasing levels in order to visually detect any changes in the resulting FRFs, which might be suggestive of underlying non linearity. In order to excite to higher amplitudes a further trial was carried out with a larger shaker LDS V400 which has a 190 N dynamic range.



**Fig. 2.5** Test rig setup horizontally



**Fig. 2.6** Broad band excitation for resonance's selected



**Fig. 2.7** Post-processing of FRFs using CONCERTO software

The non linear behaviour of the structure was confirmed by the frequency shift of the FRFs. Figure 2.7 shows the FRFs measured around 467 Hz when post processed by CONCERTO [4]. The plots show how the natural frequency varies with increased displacement, confirming the non linear behaviour of the assembly.

### 2.4.2 Importance of Vibration Settling Time

An important check before conducting a modal test is to assess the settling time of response, especially for lightly damped structures. The longer the settling time the longer the pause required before test measurement acquisition. Figure 2.8 shows a time history plot of all ten off accelerometers attached to the assembly for the excitation of the first of the resonances shown in Fig. 2.6. It is clear that a suitable time of 5 s might be required for vibrations to settle down for the casing assembly. Figure 2.9 shows the effect that having an incorrect settling time can have on the FRFs. Different structures can have very different settling times and so a preliminary check would be a sensible.

### 2.4.3 Final Test Setup and Preliminary Non-linear Testing

Following the initial assessment, it was decided to suspend the structure vertically. Despite the successful validation of the assembly model from test data obtained both vertically and horizontally supported, it was felt that the straps looping longitudinally along the assembly (when hung horizontally) was contributing significantly to the structural damping at the contact locations. Other suspension methods were considered, but none provided as much confidence when testing at larger amplitudes as hanging the assembly vertically suspended from the much stiffer CCOC. In addition, when a comparison of FRFs was made (using a reduced modal test) with the specimen supported both horizontally and vertically, they were found to be different, as shown in Fig. 2.10. The FRFs shown were both measured at position 146 with the excitation at position 25.

With the preliminary testing giving confidence that the assembly was exhibiting non linear behaviour, it was important to consider whether it would be possible to quickly identify which modes might respond more non-linearly than the others. The FMFEM mode shapes below 500 Hz were reviewed and a modal test carried out along a sector of the test assembly. The details of the test planning and FE model correlation are outlined in Sect. 2.2. As the bolted flanges were the most likely sources of non linearity those mode shapes with large deflections across the flange areas were considered first. For example the mode shape at 129 Hz (shown on the left-hand side of Fig. 2.11) responds locally around the unconstrained edge of the LPT whereas the mode shape at 202 Hz (shown on the right-hand side of Fig. 2.11) responds significantly across CCOC-HPIPT and HPIPT-LPT bolted flanges.



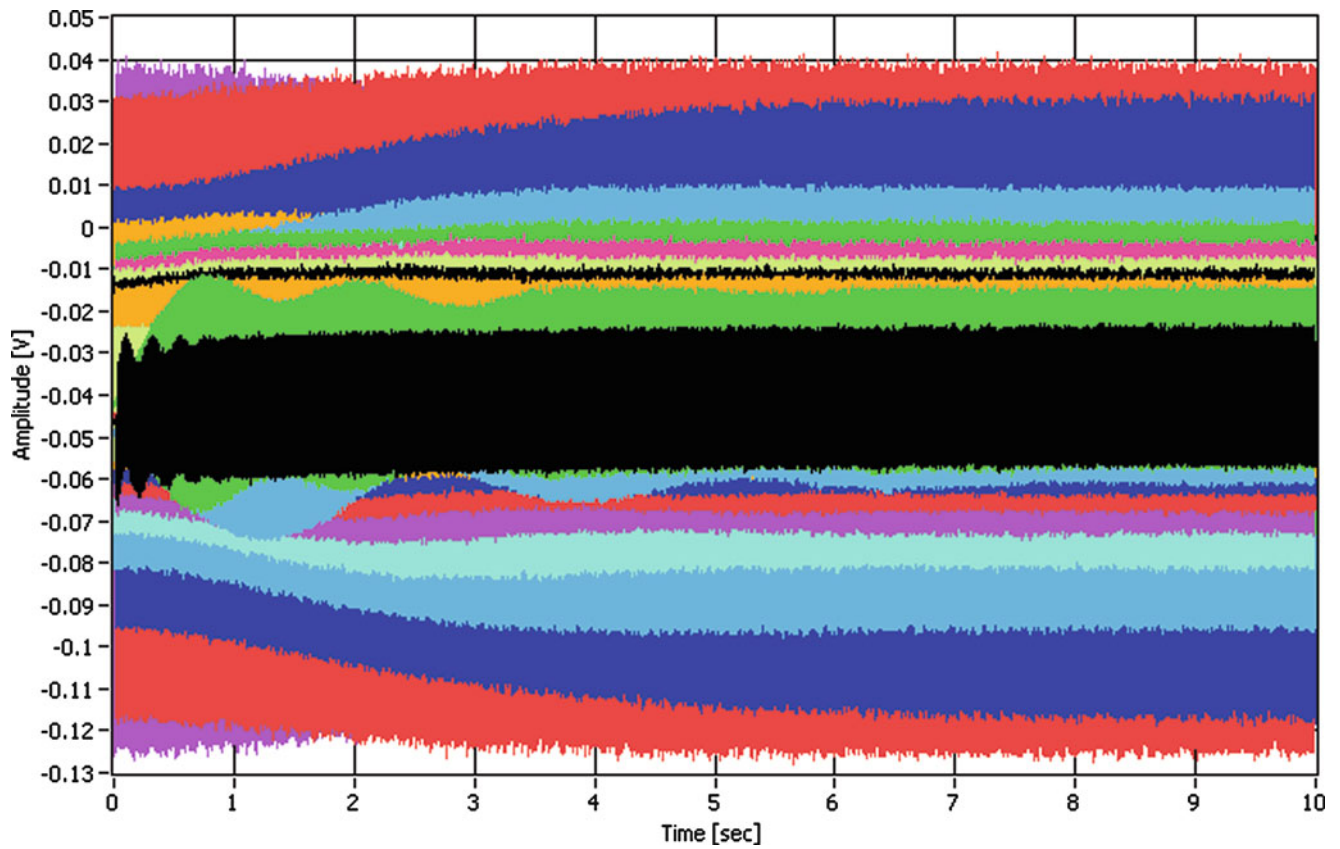
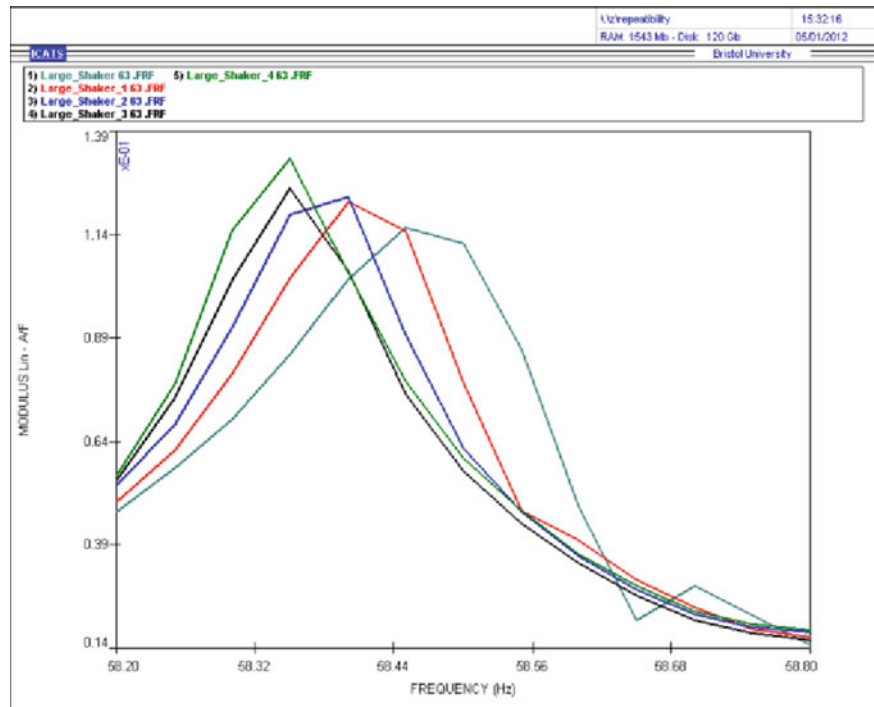
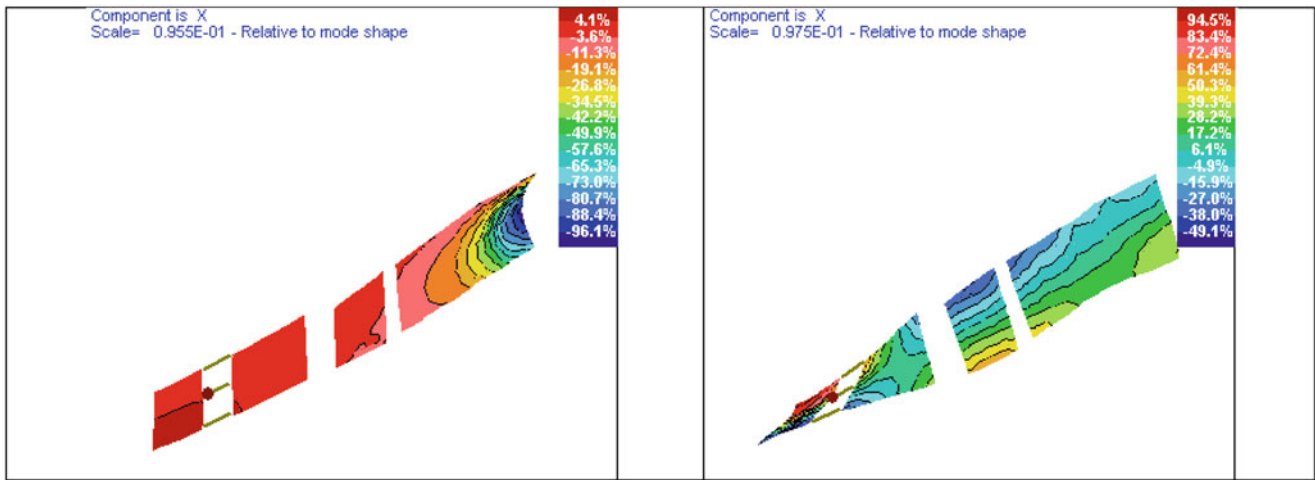
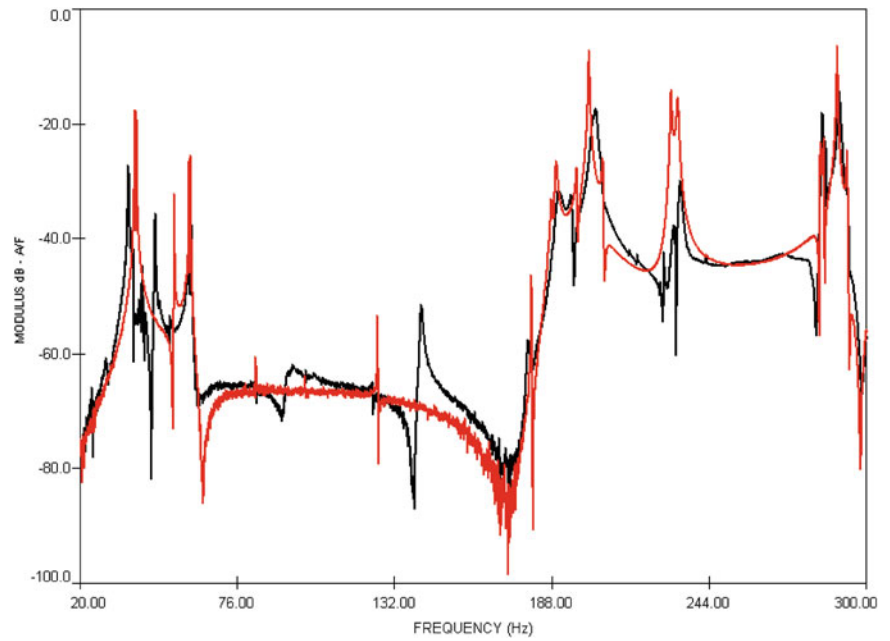


Fig. 2.8 Acquired time signal

Fig. 2.9 Effect of incorrect settling time during testing



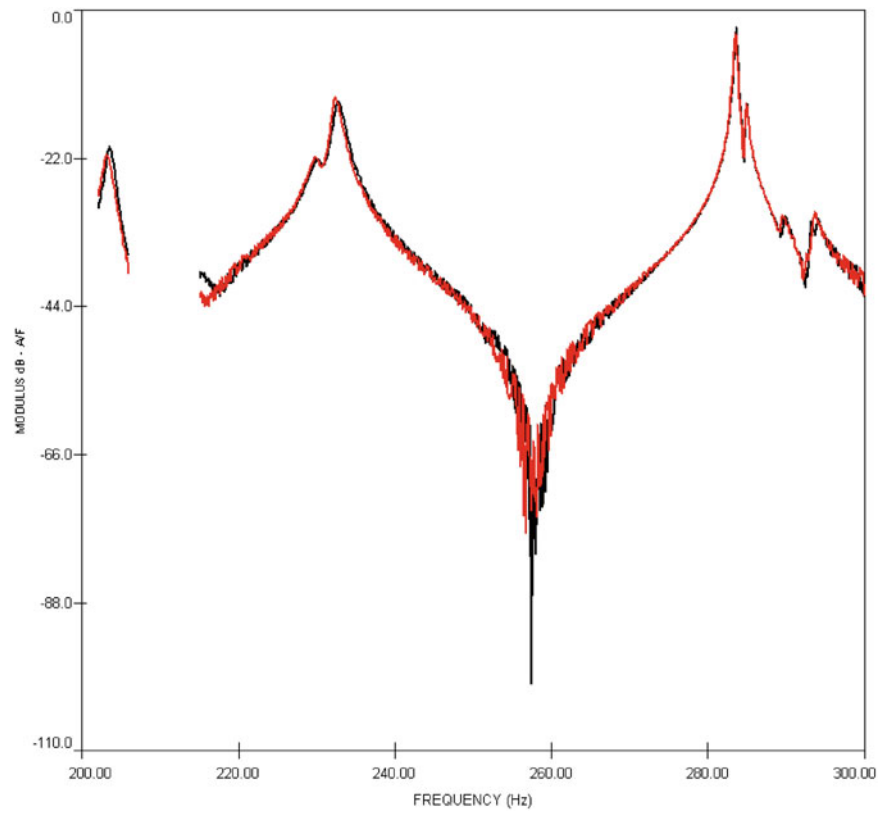
**Fig. 2.10** Comparison of FRFs measured horizontally (*black*) and vertically (*red*) (color figure online)



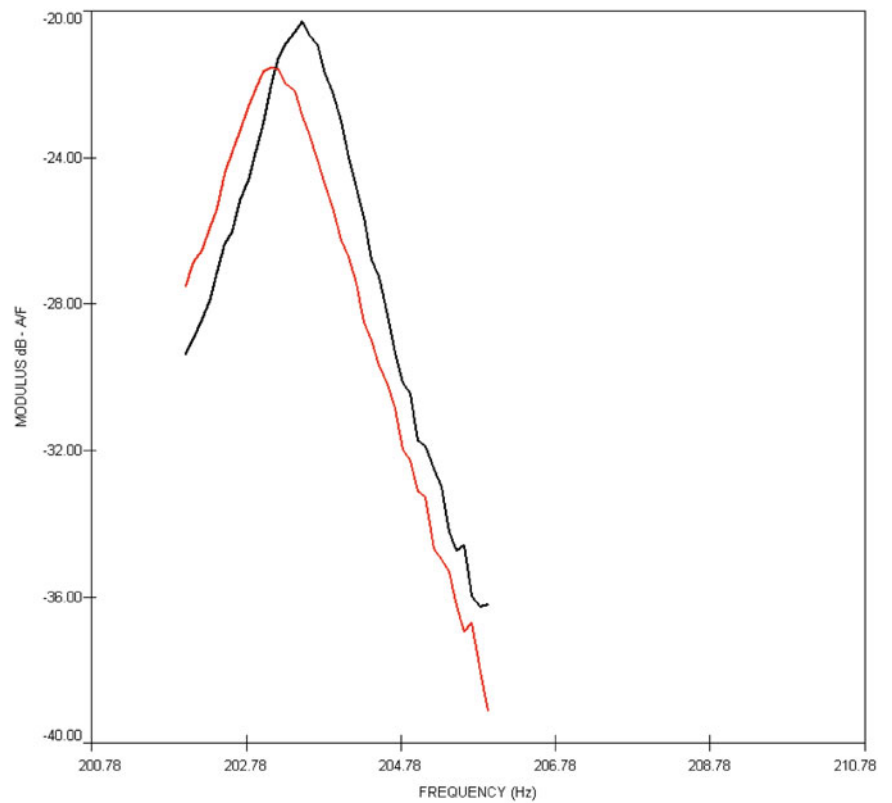
**Fig. 2.11** Identification of resonances for non linear testing using mode shapes

This qualitative assessment of the resonances using mode shapes from experimental data (or FEM predictions if available) can help to quickly shortlist the resonances to be tested further. The validity of this approach was tested by measuring FRFs at low and high excitation levels and comparing modes which presented small and large frequency shifts. Figure 2.12 shows two FRFs measured using low and high excitation levels; measurements were again carried out without force control. Figures 2.13 and 2.14 show a zoom of the FRFs where Figs. 2.15 and 2.16 show the mode shapes, respectively. These examples have been chosen to demonstrate that mode shapes can be used as good indicators when selecting resonances for non linear testing. The mode shown in Fig. 2.15 looks more likely to exercise the flanges, and has a corresponding high frequency shift its FRFs (as shown in Fig. 2.13), than the far more local (to the LPT) mode shape of which has a correspondingly small change in the FRFs as shown in Fig. 2.14. The mode shapes of such casings can be extremely useful tools for identifying areas of such test structures which are likely to respond linearly. This approach was used to down select the resonances for further, higher amplitude, non linear testing.

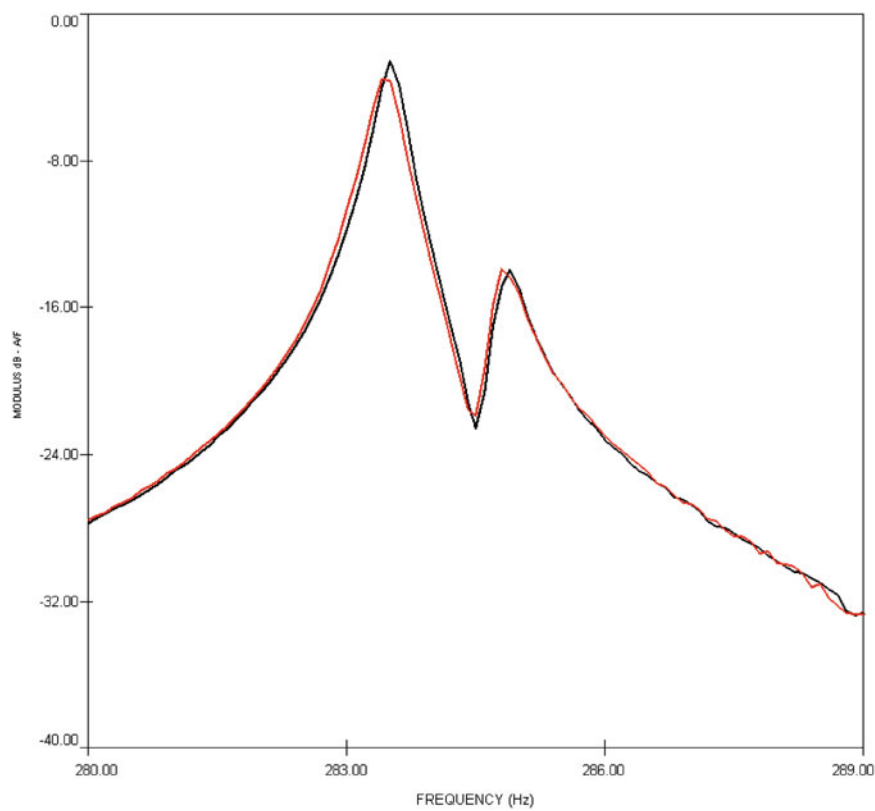
**Fig. 2.12** FRFs obtained from *high* and *low* level of excitations; *red* and *black*, respectively (color figure online)



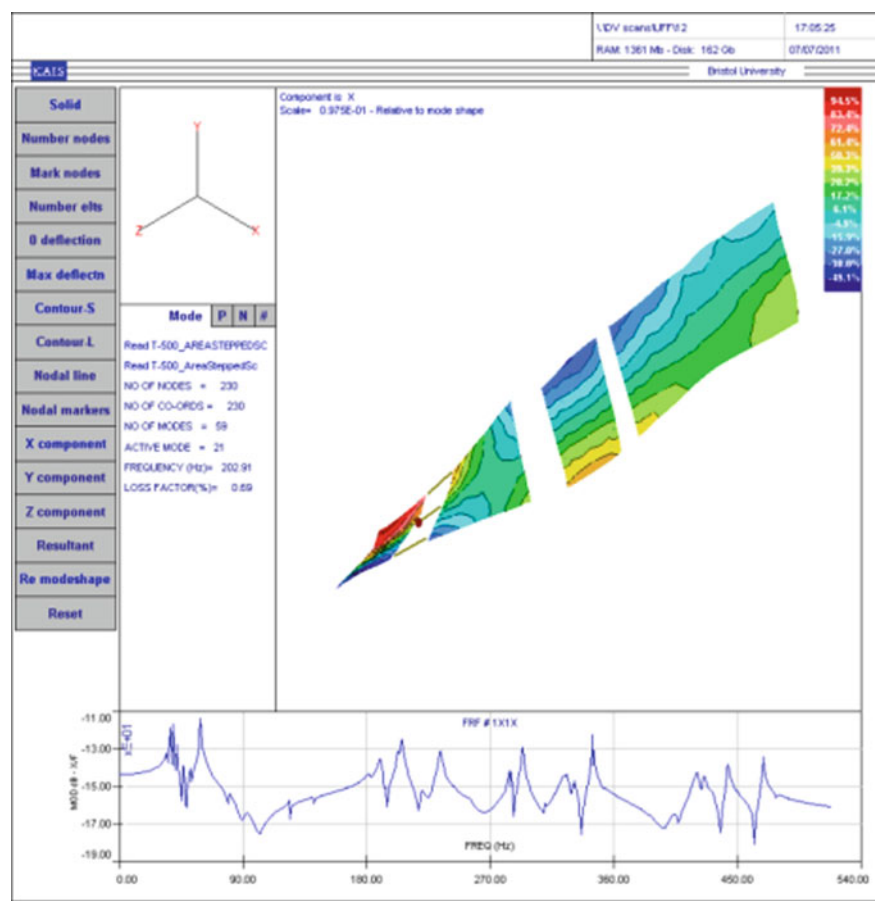
**Fig. 2.13** FRFs (zoom) around 202 Hz



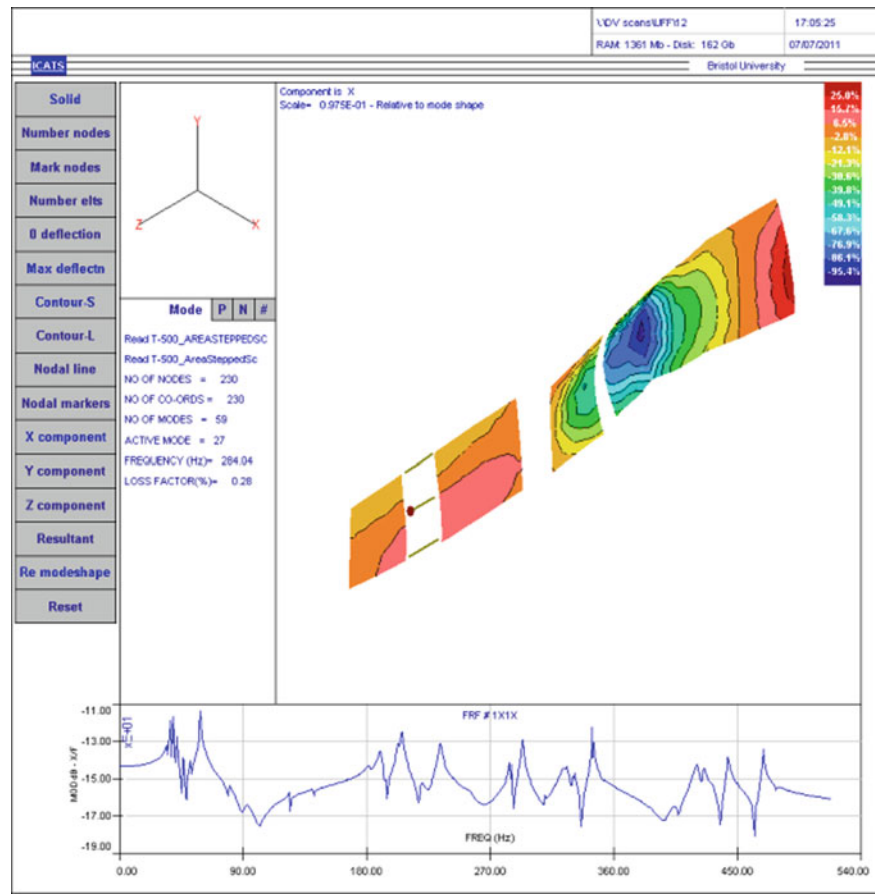
**Fig. 2.14** FRFs (zoom) around 284 Hz



**Fig. 2.15** Mode shape at 202.9 Hz





**Fig. 2.16** Mode shape at 284 Hz

## 2.5 Non-linear Testing Method

This section aims to design guidance for conducting sensibly non linear modal testing. A schematic of the process used for the initial study of the non linear behaviour of the casing assembly has been shown in Fig. 2.17.

Having measured some non linearity from the casing assembly, it was decided to produce a test matrix so as to quantify more precisely which resonances would respond non-linearly at higher excitation levels. The test matrix was also designed to highlight more clearly the influence of excitation position. It was decided to use a commercially available system for carrying out the testing. The Dynamics laboratory of Blade (University of Bristol) has an LMS data acquisition system and this was used for the remainder of the work presented. The LMS system has several excitation options and the decision was made to use the stepped sine excitation with either (i) no force or amplitude control; (ii) force control or (iii) amplitude control. The third option is the only one which permits a subsequent linear modal analysis of the resulting FRFs. Using amplitude control, response at every frequency point is measured for a given acceleration level and the output FRFs are effectively linearised and so can be processed using existing linear modal analysis tools.

The test matrix covered:

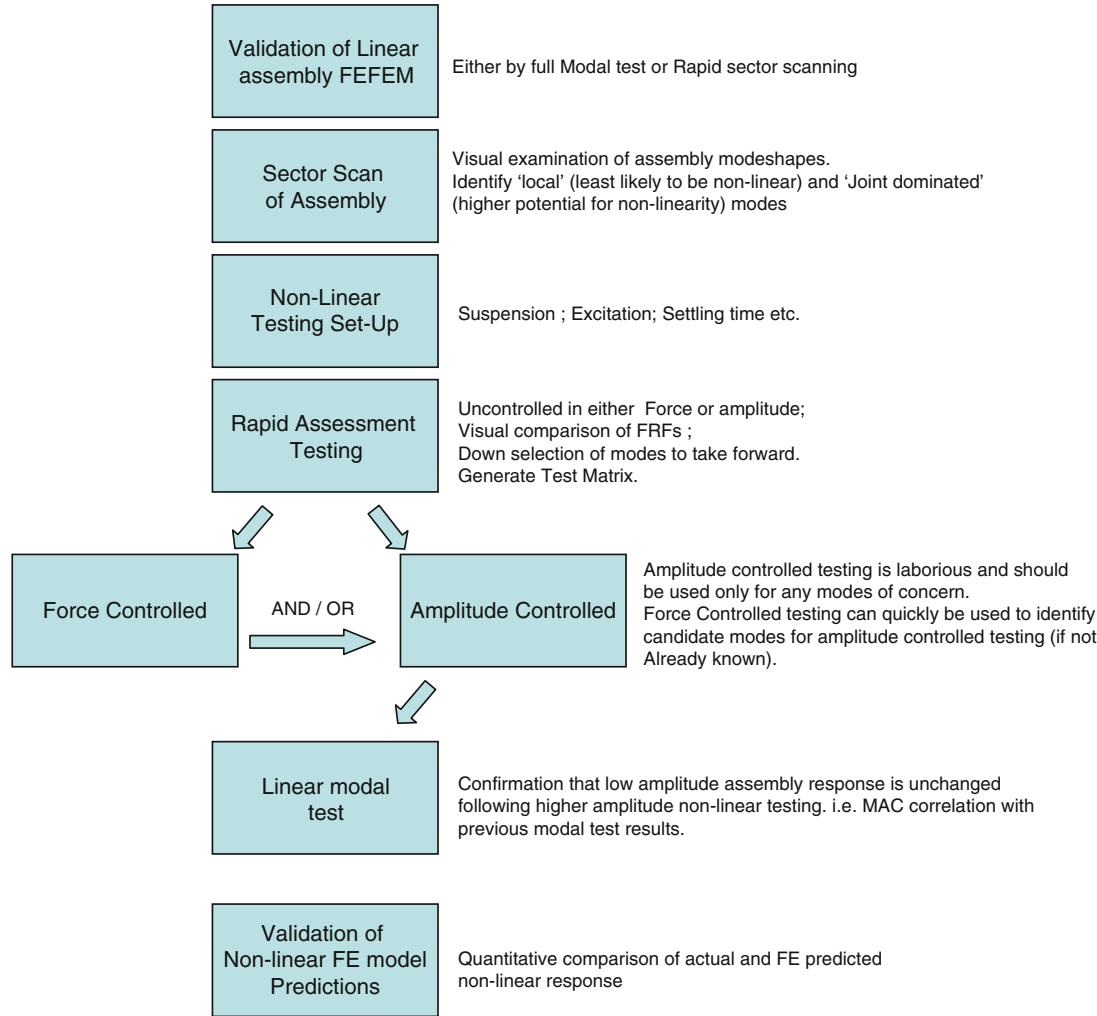
- (i) Two frequency bandwidths
- (ii) Excitation of the test structure at six different positions; two points for each component
- (iii) Excitation at three excitation levels using the force control method

and

- (iv) Excitation at three acceleration levels using the amplitude control method.

When attaching the shaker to the CCOC casing (e.g. as in Figs. 2.18, 2.19, 2.20, 2.21, 2.22, and 2.23), care was taken to ensure that it was correctly aligned with the stinger rod. Misalignment of the shaker and its stinger rod can 'corrupt' the excitation and so compromise post-processing of the test data.

It was noticed that some measurement parameters specified in the test matrix were redundant. For example, initially three levels of excitation force had been chosen but it was later decided, to reduce the testing time, this could be lowered to just



**Fig. 2.17** Schematic process for non-linear behavior study

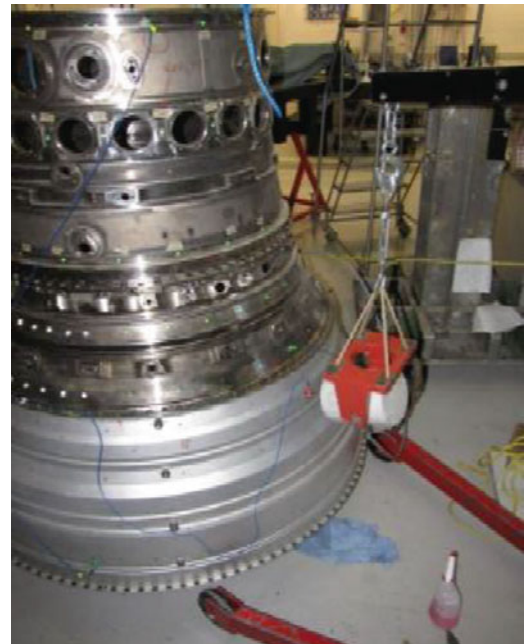
**Fig. 2.18** Position 1



**Fig. 2.19** Position 2**Fig. 2.20** Position 3

two. To assess any natural frequency shift of the test modes excitation levels of either 1 and 30 N, or 1 and 50 N were used. The lowest force level (i.e. 1 N) could always be achieved whereas the higher force levels could not always be achieved. It can be noted that FRFs measured for the lowest force levels looked, on occasion, quite noisy. This was a consequence of the poor Signal-To-Noise-Ratio (SNR) at low excitation levels and for future testing a minimum allowable SNR should be specified. For some modes the excitation levels of 50 N could also not be achieved either because the force control could not be maintained and/or of the power of the shaker was inadequate at that frequency. However, as this was intended to be an explorative approach, to be confirmed later by more rigorous testing using amplitude control; this was considered acceptable.

Measurements of FRFs for low (black) and high (red) excitation levels are presented in Fig. 2.24 for the response at the point 119. Measurement point 119 was chosen because of its proximity to the flange, as shown in Fig. 2.25, and thereby

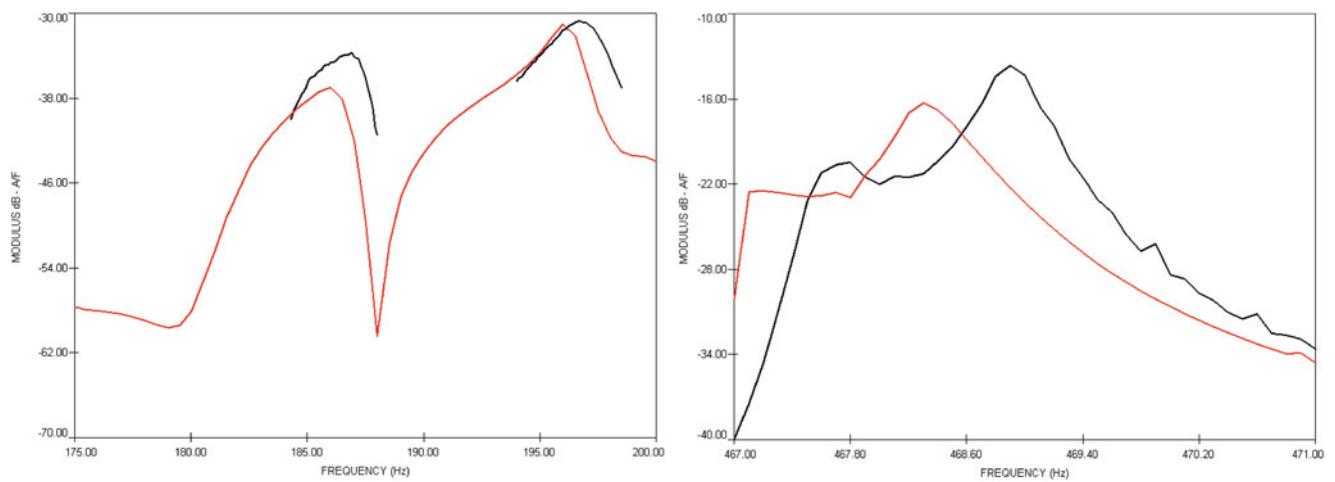
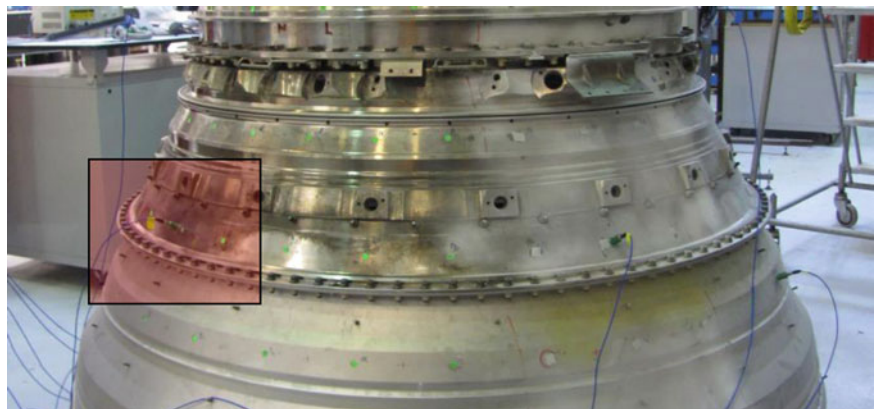
**Fig. 2.21** Position 4**Fig. 2.22** Position 5

thought to be capable of capturing more non-linear effects than the other accelerometers. This point was also used as the reference when carrying out amplitude controlled measurements.

All modes selected exhibited a frequency shift with increased excitation force, due to the non-linear behaviour of the flange joints. The quantitative assessment of the non linearity of the target modes was carried out using amplitude control so as to be able to analyse FRFs using currently available linear modal analysis tools.

As stated above the accelerometer position for the amplitude controlled testing was measurement point 119. Only one position at the time can be used for amplitude control, as the response from the remaining nine off accelerometers will be uncontrolled. If required, amplitude control measurements could have been carried out individually for the other accelerometers positions taking each one in turn. However, due to time limitations this could not be done as it would have increased the required testing time nearly tenfold. Careful selection of which measurement point(s) to use for amplitude controlled testing is essential to avoid wasting the available, and usually highly limited, test time. Using of the preliminary (and much quicker) force controlled testing can act as a effective guide to avoid poor choices.



**Fig. 2.23** Position 6**Fig. 2.24** FRFs measured at point 119 with different force levels, high in *red* and low in *black*, respectively. Excitation position 1 (color figure online)**Fig. 2.25** Response measurement point 119, *red box* (color figure online)

**Fig. 2.26** Natural frequency shift with increased acceleration with excitation position 1 and response measurement at point 119

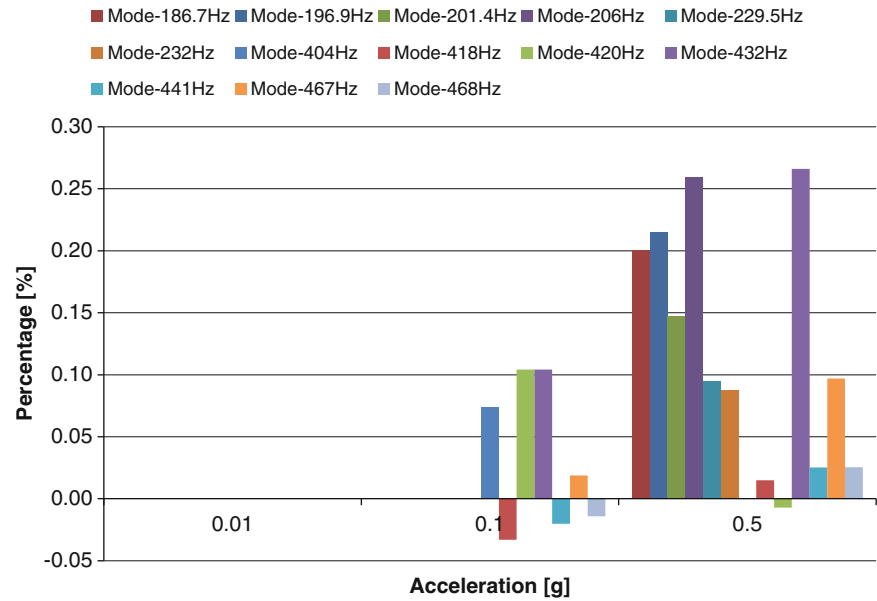


Figure 2.26 shows the observed frequency shifts (when compared with the linear modal analysis results) of the target resonances, for increasing levels of acceleration. FRF measurements were performed for most resonances at 0.01, 0.1 and 0.5 g. However, it can be seen that some resonances were not measured at 0.1 g. The aim of the post-processing was to be able to identify those resonances which were responding more non-linearly than the others. The modes at 186.7, 196.9, 206 and 432 Hz exhibited noticeable non-linear behaviour compared with the other modes. Although, the frequency shifts seen are quite small, such plots indicate which resonances are good candidates for higher amplitude testing.

### 2.5.1 Test Results for SILOET 2.3.2 Test Case

Completion of the test matrix produced some interesting results. The preliminary scoping performed using the force control method was able to identify the key resonances to be selected for amplitude control testing. The amplitude controlled testing focussed on achieving much higher levels of vibration in order to observe more consistent non linear behaviour of the structure. Based on the testing carried out it was decided to use excitation position 1 because of the higher quality test data that resulted. Positions 2–6 were discarded because

(i) Not all modes could be excited,

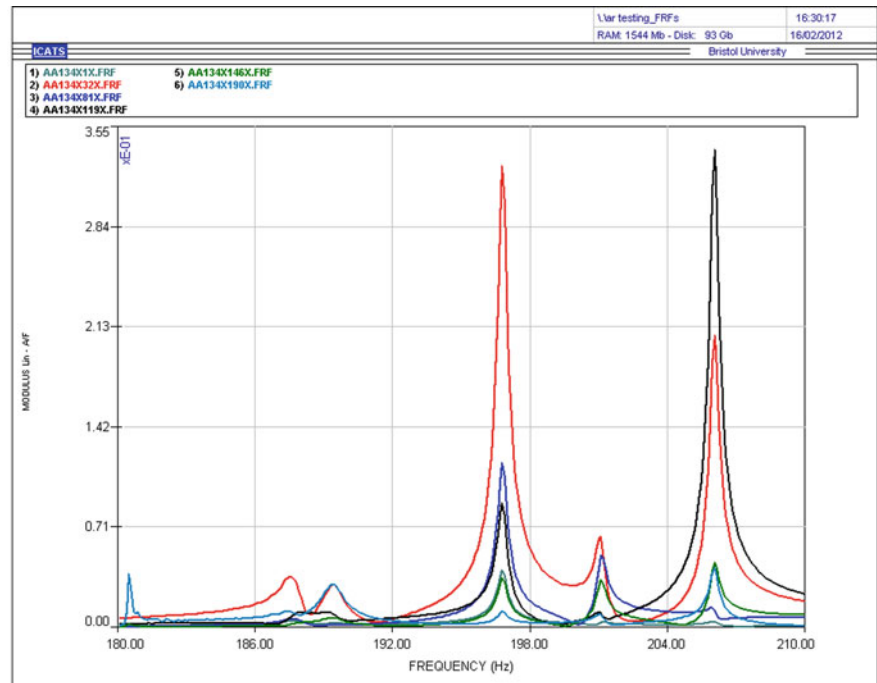
and

(ii) When modes were excited they did not present non linear behaviour.

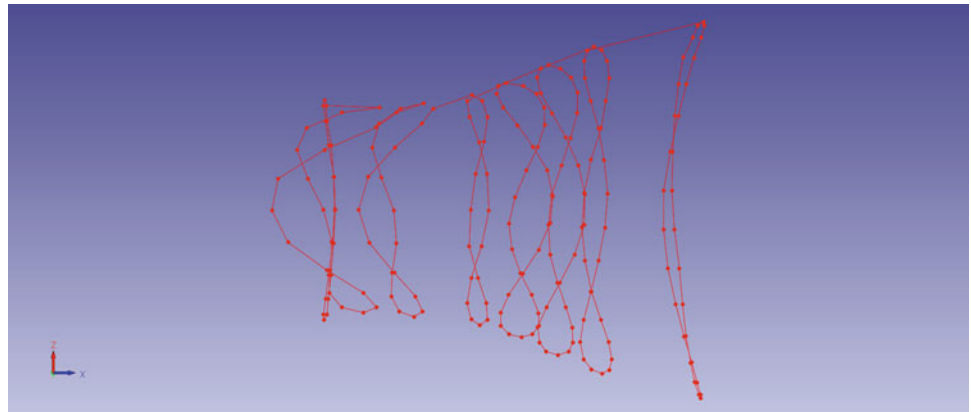
Measurements were repeated for two modes (i) 194.6 Hz and (ii) 205.4 Hz. The first was not predicted by the FMFEM even though it was measured during both the VIVACE [1] and current SILOET testing of the fully assembled structure. Figure 2.27 shows a set of FRFs obtained during the linear modal analysis; the excitation position for the impact hammer was at point 134. The plot shows clearly the presence of a mode at approx. 194.6 Hz which was not predicted. The most likely explanation is that the FMFEM, which had a simplistic ‘fully fixed’ boundary conditions between the flange mating faces, could not correctly predict modes which have significant separation of the flange faces of the bolted test structure. Figures 2.28 and 2.29 present the mode shape in the plane Z-X and Y-X, respectively. Table 2.1 showed that there are other mode pairs (7, 9 and 10) for which the FMFEM poorly predicted the assembly frequencies, again strongly suggesting that, although quick, there is a need to replace the overly stiff ‘fully fixed’ representation of the bolting with something more detailed. Figures 2.30 and 2.31 show mode shapes measured at 194.6 and 205.4 Hz, respectively, using SLDV measurement method.

Mode at 194.6 Hz was tested up to nine excitation levels, 0.1, 0.5, 1, 1.5, 2, 3, 4, 4.5 and 5 g, respectively. Mode at 205.4 Hz was tested using excitation levels of 0.1, 1, 2, 3, 5, 7, 9 and 11 g, respectively. Modal analysis was performed for calculating natural frequencies and damping loss factors, as shown in Figs. 2.32 and 2.33, respectively. It is of interest that

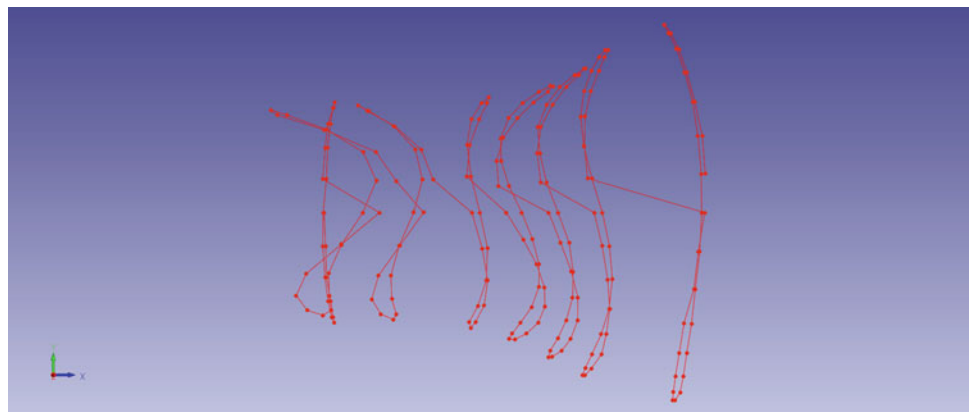
**Fig. 2.27** FRFs measured with impact hammer (excitation point-134) response at points 32 (top red), 1, 81, 119, 146 and 190, respectively (color figure online)



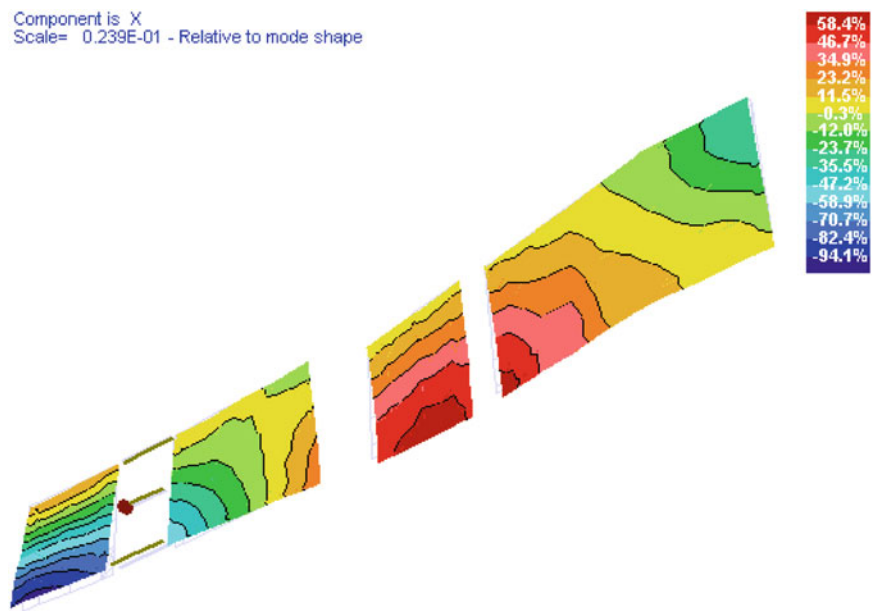
**Fig. 2.28** Mode of vibration at 196 Hz (plane Z-X)



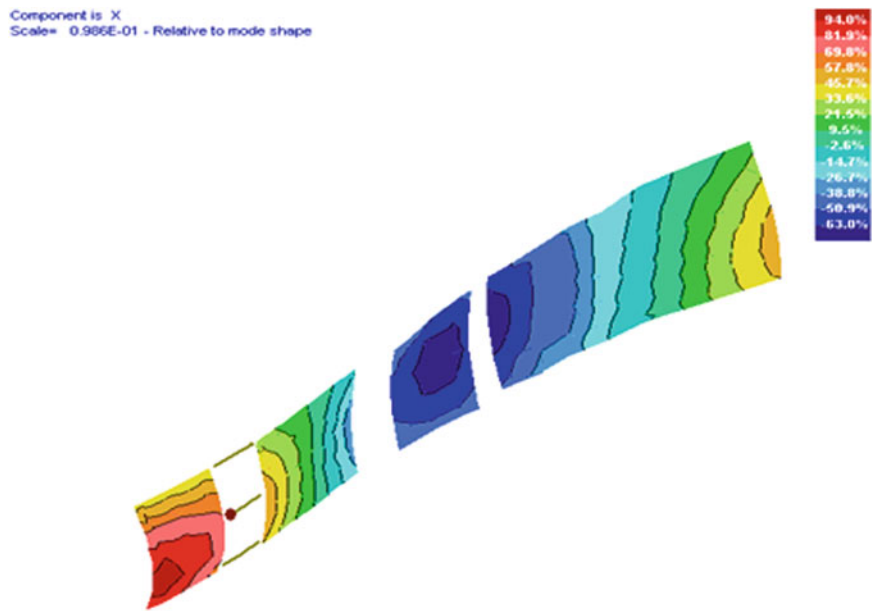
**Fig. 2.29** Mode of vibration at 196 Hz (plane Y-X)



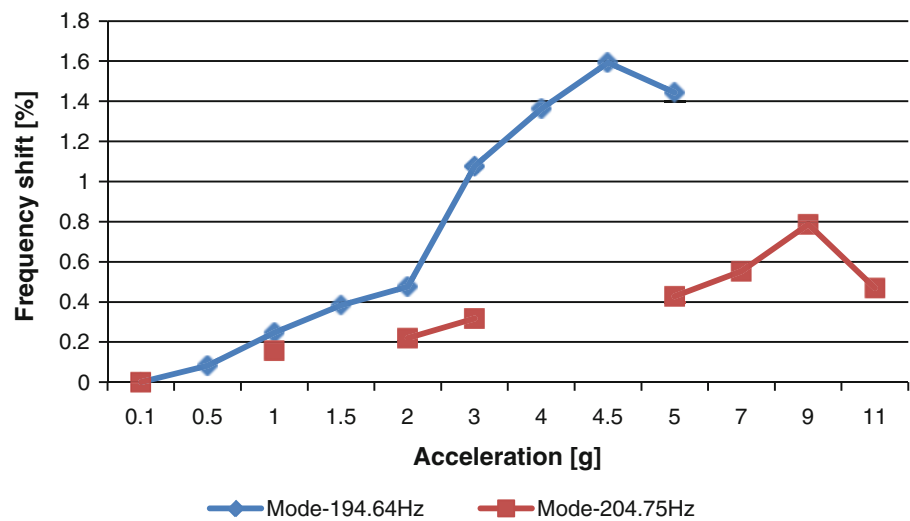
**Fig. 2.30** Measured mode at 194.6 Hz using sector scanning SLDV measurement method



**Fig. 2.31** Measured mode at 205.4 Hz using sector scanning SLDV measurement method

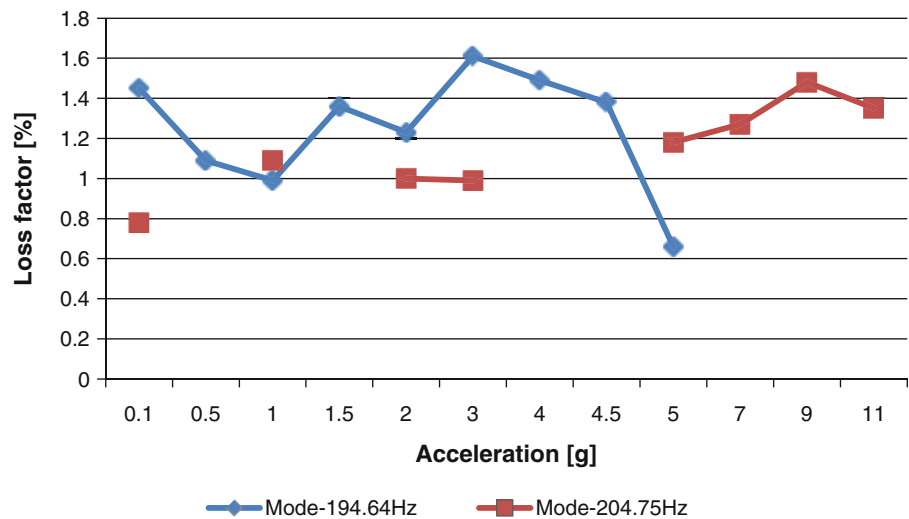


**Fig. 2.32** Comparison of frequency shifts at different vibration levels





**Fig. 2.33** Comparison of loss factors at different vibration levels



both frequency shift plots show a drop for the last measurement that is 5 and 11 g, respectively. These two FRFs present a quite flat shape and it is possible that either the modal analysis was not very accurate because the small number of frequency point for the curve fitting or response behaviour of the flanges not fully captured at these vibration levels with FRFs. Such experiments are at the limit of the commercial software's capability. The results at 5 and 11 g should be treated with caution until they have been confirmed by additional testing.

The results presented here tend to confirm that the approach of Fig. 2.17 for performing non linear modal testing is valid. The initial selection of modes using either FMFEM or experimental mode shapes helped to quickly shortlist the resonances for the modal testing. Screening these modes using the force control method helped to quickly identify the resonances to consider for the amplitude control testing. Finally, the FRFs measured using amplitude control helped confirm the resonances whose non linear behaviour should be studied in greater detail.

While it is clear that FMFEMs are capable of predicting modes of single components and also of sub-assemblies, they can omit or miscalculate those resonances which exhibit strong non linearity.

## 2.6 Conclusions

A set of criteria have been listed so as to give guidance for non linear testing of large casing assemblies. It is important to perform test planning using the FEM so as to identify the best excitation positions for EM shaker. The experimental work with a civil aircraft engine casing has shown that some of the positions chosen for exciting the structure were not optimal because some modes could not be excited either sufficiently, or at all. There are no additional precautions in suspending the test structure than the ones already applied for linear testing.

The alignment of shaker and stinger rod must be very accurate. This becomes even more important as the levels of excitation are far higher than those of linear testing. Any misalignment between shaker-stinger-load cell will introduce unnecessary errors.

The settling time must be estimated correctly before trials are started. It was shown that an incorrect choice of settling time can lead to misinterpretation of the acquired FRFs. This can be as simple as acquiring the time history data from the test structure for a single impact where the time taken for the vibrations to decay would make a suitable settling time.

The use of mode shapes obtained either from FE models or from modal analysis performed with SLDV measurement method on sector(s) of a specimen is very beneficial for identification of resonances. All mode shapes presenting large levels of vibrations in the areas of, for example, bolted flanges can be confidently selected for the next measurement phase.

Quick measurements made at low and high excitation levels, but without controlling the excitation force, can highlight signs of non linear behaviours in the test structure. Although only a qualitative assessment it is still very helpful in the early stages of testing.

Measurement of FRFs using the force control method can be carried out for the selected modes. Two excitation levels, one low and one high, is usually enough to identify those modes with the highest frequency shifts due to non linearity.

All accelerometers included in the test planning for the linear modal analysis should be used for the force control measurement method. The analysis of all measured FRFs can help to identify those resonances responding more non-linearly than the others and so help to select a smaller number for subsequent amplitude control measurements. This will also help identify the best measurement positions for conducting an amplitude control.

The final phase of non linear modal testing is the measurement of FRFs using amplitude control. This technique is time consuming and so it is advisable to use it only for those modes which are definitely thought to be non linear. The FRFs resulting from amplitude controlled testing can be post-processed using currently available linear modal analysis tools.

Non linear behaviour was observed and measured for a mode which presents a strong out of plane vibration of the flanges.

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