Validation of finite element modeling approach for a rubber sealed structure by performing experimental modal analysis

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ABSTRACT

Adhesive bonding usage like rubber seal to assemble the glass in automotive sector is very common to use. However, creating finite element (FE) model of the structures assembled using such rubber seal is still quite difficult due to non-linear behavior of the bonding material.

To build a reliable FE model of a vehicle such as Concept V1, which has a unique glazed roof structure, is became necessary. This paper aims to determine whether an acceptable linear FE model can be built for correlating and adjusting the FE models using the measured modal properties of structures.

Verification of the FE modeling technique is performed progressively in three step to secure FE models in every steps forward. First of all, it is done by validating the properties of a simple welded structure. Secondly, experimental modal analyses are performed on test body without glass. Finally whole test body is tested to obtain the modal properties. The measured modal data are also used to adjust the properties of effective rubber seal model of the glass in order to minimize the error between the predicted and the measured natural frequencies and mode shapes.

Keywords: Finite element modeling, experimental modal analysis, roof structure

1. INTRODUCTION

Choosing a proper type of joint for bonding body and glass is important in a vehicle like Concept V1 in which the roof is completely glazed as shown in Figure 1. Rubber seal, which is a type of adhesive bonding, has been chosen because its advantages are that an even stress distribution is obtained over the whole of the joint; there is a low melting point and so flammable components can be joined; corrosion is reduced since the adhesives are gap filling; weld marks are eliminated and vibration-absorbing adhesives can be used to damp the motion in thin shell joints [1]. However, modeling rubber seal is a tricky and complex process. In this point of view, obtaining a validated theoretical model of structures is very important for design and optimization purposes. Such models can be used to optimize the structure for maximum efficiency, maximum life, minimum vibration and noise. However, model validation and updating is usually a difficult task, not only due to the cost

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involved during this process, but also due to the fact that a unique and physical model may not be available at the end of the model updating process. The problem is further complicated when the structure has joints and associated nonlinearities [2]. The joints, used in automotive, are rubber seal, as mentioned before, which is complicated since modeling rubber seal is surrounded with unknown parameters. Ahmedian et al. approach [3] to the modeling of the rubber seal recommends to use springs spaced at regular intervals around the edge of the window glass. The physical parameters of an equivalently distributed stiffness of these springs are identified from measured natural frequencies to provide a full model of the rubber seal and window together. In another study, Calvert and others [4] present an overview of the experimental and analytical work conducted in order to produce a reliable method of predicting seal performance which was verified by means of comparison with the seal in normal working conditions. Wagner et al. [5] dealt with a seal cross section that is analyzed for compression load deflection behavior, contact pressure distribution and aspiration due to a pressure differential across the seal using nonlinear finite element analysis. Stenti et al. [6] research concerns the structure-borne vibration transmission of a car door weather seal by considering a detailed non-linear FE model of the seal on component level and focusing on the extraction of the seal linearized equivalent model, which accounts for the amount of pre-stress of the seal. Likewise, Friswell et al. [7] consider two critical issues in model updating which are how to parameterize a finite-element model and how to regularize the resulting estimation equations to obtain a well-conditioned solution by exemplifying a rubber seal which provides the connection between a vehicle window and the car body structure.

Figure 1 – Concept V1

When performing experimental modal analysis, uncertainties and complexities have to be faced. Durant et al. [8] dealt with both model uncertainties and data uncertainties which must be taken into account to improve robustness of computational models which analyze structural vibrations and internal acoustic levels. Two main problems in this research are the experimental identification of the parameters controlling the uncertainty levels and the experimental validation. As an addition, Çakar and Şanlıtürk [9] dealt with the elimination of mass loading effects of transducers from measured frequency response functions (FRFs), which is adversely affected by many factors, most significant sources being noise and systematic errors. It is also known that the accuracy and the reliability of various analyses using the measured FRFs depend strongly on the quality of measured data. In other words, one of the major systematic errors in measured FRFs, namely the mass loading effects of transducers, is tried to remove. Çakar and Şanlıtürk [10] also handled the quality of measured FRFs which is adversely affected by noise originated from test environment as well as electronic devices with measured signals are contaminated with noise when a data acquisition system is used for an experimental measurement.

In this paper, a linear FE model of structures, which is constructed with welding and rubber seal, will be built for the determination of the natural frequencies and mode shapes with acceptable accuracy. First of all, this is done by validating a structure without any welding and rubber seal. Then it is done on two welded structures with different complexities. One of them is test body without rubber seal and glass. Finally the experimental modal analysis are performed on the test part, which includes all complexity mentioned. The experimentally determined modal data are then used to adjust the effective modeling method of structure with rubber seal in order to minimize the error between the analyzed model and the measured test body's natural frequencies and mode shapes. Results confirm that the procedure explained could be performed to obtain relatively representative linear FE models like these structures.

2. DEFINITION OF THE PROBLEM AND PROPOSED APPROACH

The need to model the rubber seal in an efficient way is one of the fundamental matters for such a vehicle like Concept V1 of which roof is completely glass. This is also important for the determination of the natural frequencies and mode shapes with satisfactory accuracy and to monitor the effects of the design immediately after modifications. However the exact rubber seal model involves complex non-linearity. Therefore, there is a need to develop approaches that will be applicable model of structures with rubber seal and this will also have a suitable accuracy by using simple, approximate rubber seal models.

In this study, four different approaches are used to model rubber seal and simulate its non-linear behavior accurately. In the first approach, rubber seal is assumed to be a solid (CHEXA) element [11] with appropriate material properties. This solid element is connected to glass and body with rigid elements of which connection nodes have interpolated weightings (RBE3) [12]. In the second approach, a bushing element (CBUSH) with an appropriate stiffness value, is used as rubber seal and connection between the bushing element and glass/body. Bushing elements are also connected with rigid elements of which connection nodes have interpolated weightings (RBE3). As an another approach, the same bushing element is used again but this time it also connects body and glass without any other element. The final approach is to use a simple rigid element (RBE2) to both model rubber seal to connect body and glass.

The proposed approaches, in this paper, are applied to determine the accurate FE modeling technique by utilizing the predicted and measured modal properties of sample structures. The flow chart of the proposed approach is given in Figure 2.
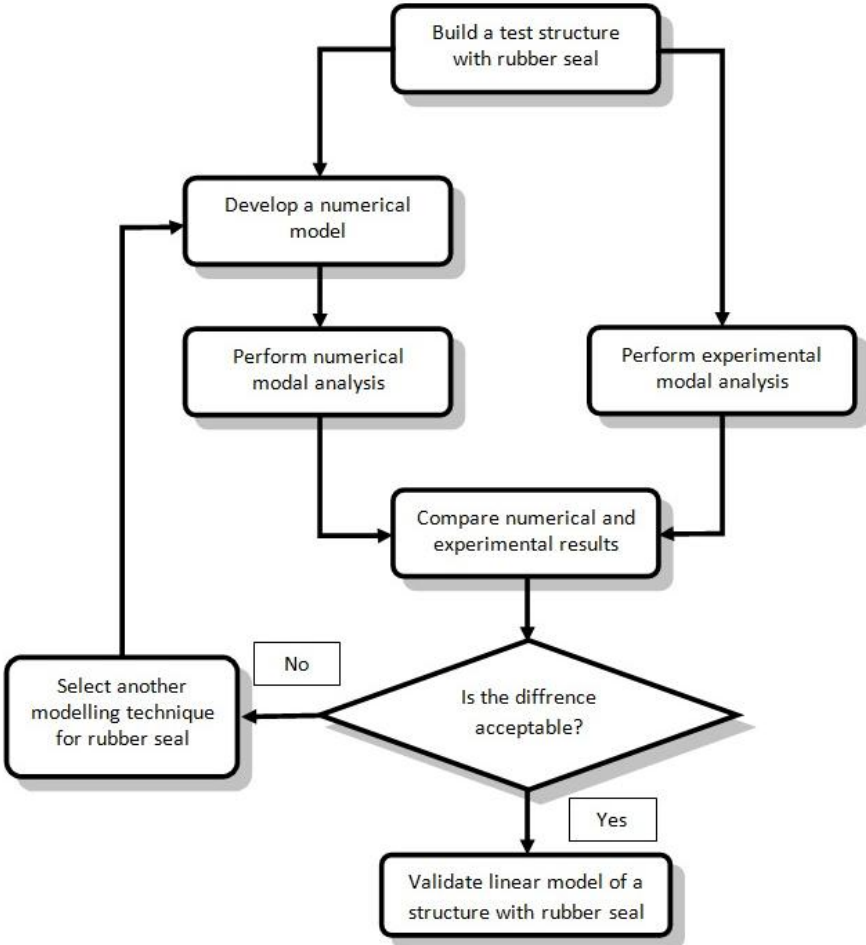

Figure 2 – Flow chart to obtain the accurate modeling technique

3. CASE STUDIES

3.1 Model Validation of a Basic Structure

Prior to validate the structure model with rubber seal, the FE model without rubber seal is examined as a first step of all three. In first two steps, the main aim is to be confident for modeling the welding structures and its discontinuities in a linear FE model. In Figure 3, the structure and FE model is shown.
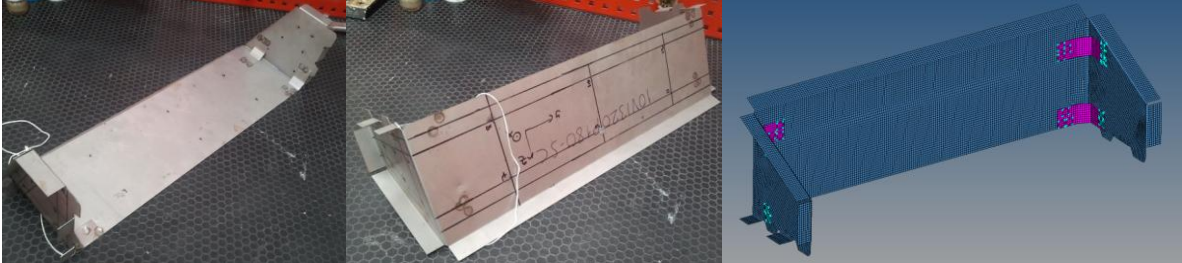

Figure 3 – Welded structure and its FE model (Model 01)

Experimental modal analysis was performed on the structure in free-free conditions using the FRFs. The FRFs were measured by impact excitation with an instrumented modal hammer while the responses were recorded by accelerometers. Excitation points were carefully selected to avoid missing any mode within the frequencies of interest. During tests, imaginary part of the point FRF and its force spectrum are checked to be confident for sufficient excitation as shown in Figure 4. If the energy distribution is proper while excitation, the imaginary part of point FRF is either entirely negative or positive in all interested frequencies. Being negative or positive is affected by the direction of the impact according to the response direction. Force spectrum is one of key properties for confident FRF measurement with impact hammer. For confident FRF measurement, the decrease of the level should not be more than 20dB in interested frequency band. [13, 14, 15]

In this case, it is considered that the FE analysis is a steady-state problem in which main characteristics is that the response of the system does not change with time. [16] On the other hand, more than 97% of finite element models are built by quadrilateral shell elements to avoid any significant errors within the frequency range of interest. To get FE model more realistic and obtain more accurate results, all details are modeled, a fine element size is used and connections are made properly.
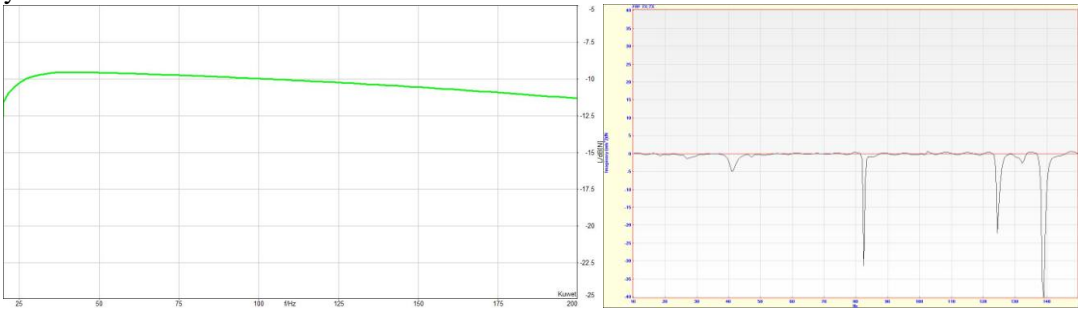

Figure 4 – Force spectrum and imaginary part of point FRF for Model 01

Table 1 – The predicted and the measured natural frequencies of Model 01

Mode	Mode Frequencies		Differences
	FEA Results (Hz.)	Test Results (Hz.)	
1	33,09	33,30	-0,63%
2	48,54	47,70	1,76%
3	52,56	54,70	-3,91%
4	88,37	91,10	3,00%
5	135,73	135,00	0,54%

The predicted and the measured natural frequencies are in agreement quite well. As an example first five modes are shown in Table1. The measured and the predicted FRFs are also compared in the mean of mode shapes which are also in agreement well as shown in Figure 5. Small differences in natural frequencies are believed to be due to the mass loading effect of the transducers. [9]

	Mode 1	Mode 2	Mode 3
FEA Results			
Test Results			

Figure 5 – The predicted and the measured mode shapes

3.2 Model Validation of Main Test Structure without Rubber Sealed Glass

The first test part, mentioned above, was fairly simple in terms of its construction and welded areas. Therefore, it is decided to test the main test structure before the assembly of the glass by rubber seal to ensure FE modeling approaches. That was used in first test part.

Test Photo	
Model Picture	
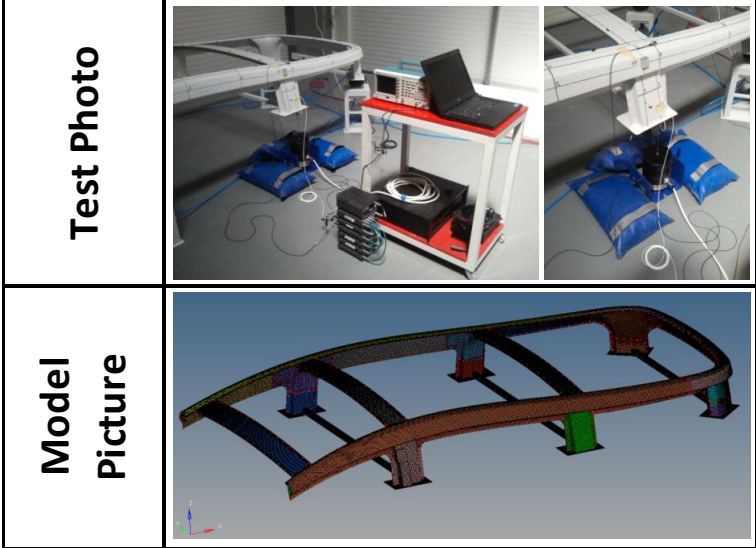
Figure 6 – Main test structure (MTSwoRSG) without rubber sealed glass and its FE Model

Main test structure without rubber sealed glass and its FE model are given in Figure 6. The same approach as in the previous section was followed to model and test this structure. Similarly, natural frequencies were predicted and they were identified using measured FRFs via experimental modal analysis.

During tests, imaginary part of the point FRF and its force spectrum are checked again due to be confident for excitation as shown in Figure 7. According to Maxwell's reciprocity principle, additional

measurements are also performed to see whether impact energy distribution is enough to excite all interested frequency ranges. Maxwell's reciprocity principle means that the response in a point p due to an input in point q equals the response in point q due to the same input at point p [17]. As shown Figure 8, impact energy distribution, according to Maxwell's reciprocity principle, is quite well to ensure the measurements.
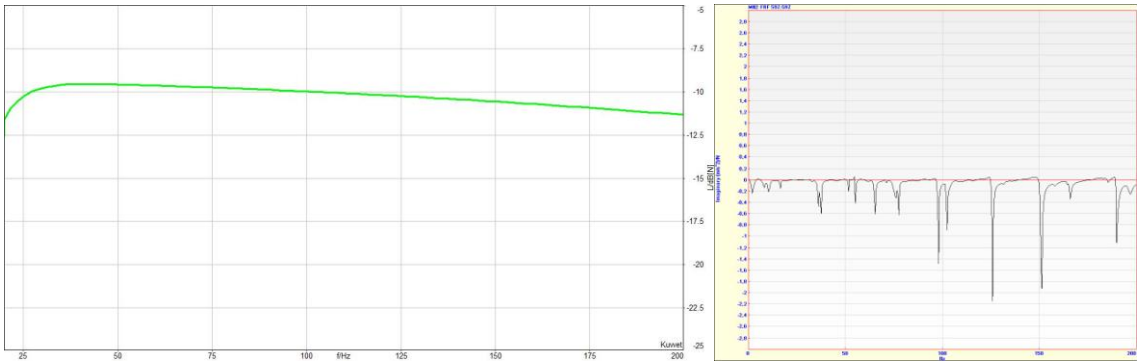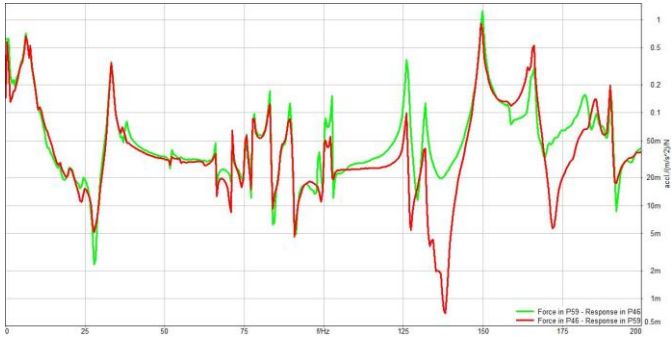

Figure 7 – Force spectrum and imaginary part of point FRF

Figure 8 – One of the performed measurements of Maxwell's Reciprocity Principle

As an example of the first ten modes shown in Table 2, the predicted and the measured natural frequencies are in good agreement except first mode. It is because of the dynamics of the experimental rig which is supposed to provide a rigid boundary constraint. However, it behaves like a rigid body mode at very low frequencies because of its large mass [3] and the energy of rigid body modes shifts first natural modes. The measured and the predicted FRFs are also compared in the mean of mode shapes which are also in good agreement as shown in Figure 9.

Table 2 – The predicted and the measured natural frequencies

Mode	Mode Frequencies		Differences
	FEA Results (Hz.)	Test Results (Hz.)	
1	12,08	16,66	27,49%
2	31,34	33,20	5,60%
3	34,28	36,15	5,17%
4	35,25	37,58	6,20%
5	51,66	51,85	0,37%
6	54,55	55,05	0,91%
7	61,62	61,79	0,28%
8	65,25	65,44	0,29%
9	65,78	65,62	0,24%
10	72,34	71,22	1,84%
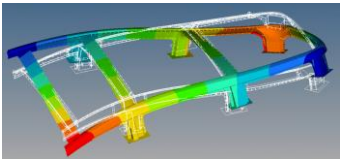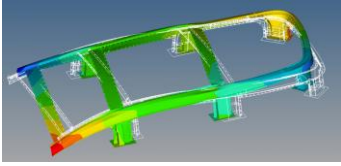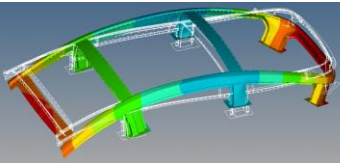
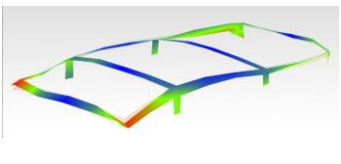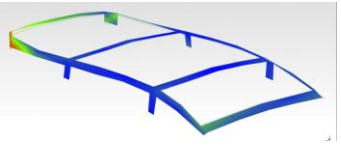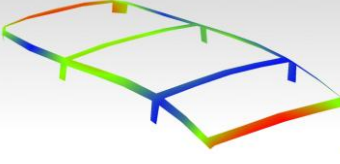|	Mode 1	Mode 2	Mode 3
FEA Results			
Test Results			

Figure 9 – The predicted and the measured mode shapes

3.3 Model Validation of Main Test Structure with Rubber Sealed Glass

The first two steps are performed to ensure FE modeling approaches of the welded structures. In this case, excitation is performed by both modal hammer and modal shaker to see if the energy distribution is high enough to excite in all interested frequency ranges by the modal hammer for the structure shown in Figure 10. Also four different approaches, described above, are compared to choose an accurate rubber seal model.

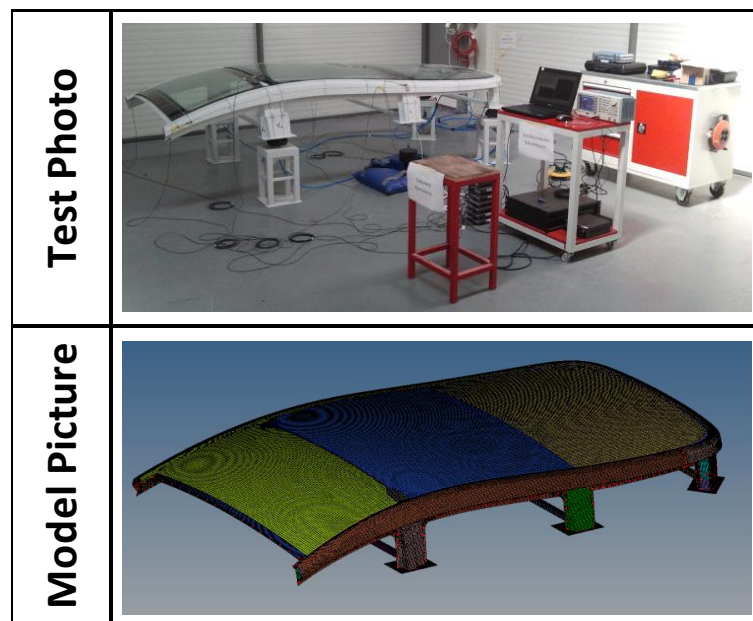|	
Test Photo	
Model Picture	

Figure 10 – Main test structure with rubber sealed glass (MTSwRSG) and its FE Model

Test structure with rubber sealed glass test setup with shaker is shown in Figure 10. In this step, modal shaker and modal hammer's excitation characteristics are compared.

To compare modal shaker and modal hammer's excitation, force spectrum and cross FRFs are examined as shown in Figure 11 and Figure 12. For this structure, modal hammer excitation is also reliable.
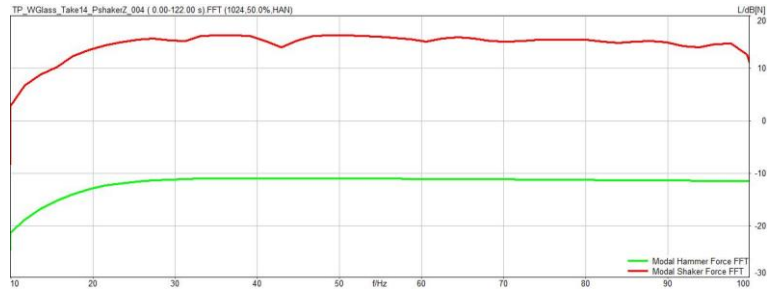
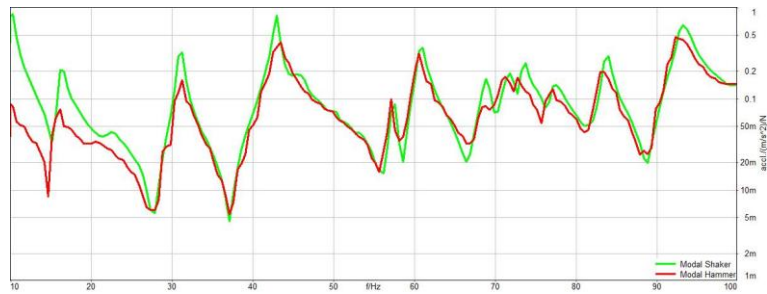Figure 11 – Force spectrum comparison of modal shaker and modal hammer

Figure 12 – An example of cross FRF comparison of modal shaker and modal hammer

To ensure the reliability of the hammer excitation, imaginary part of the point FRFs and Maxwell's Reciprocity Principle are also checked as shown in Figure 13 and Figure 14.
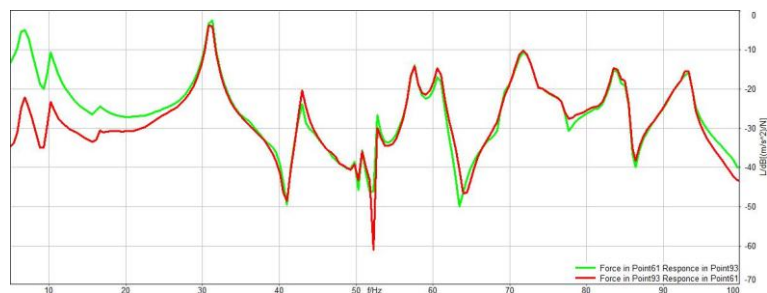

Figure 13 – Imaginary part of point FRFs

Figure 14 – One of the performed measurements of Maxwell's Reciprocity Principle

Four different approaches of rubber seal modeling are examined and compared with test results as presented in Table 3 and Figure 15. Connection Type1 is the most reliable and confident one to simulate rubber seal modeling.

As it is mentioned above, the differences in first mode is due to the dynamics of the experimental rig which is supposed to provide a rigid boundary constraint. However, it behaves like a rigid body mode at very low frequencies because of its large mass [3] and the energy of rigid body modes shifts first natural modes.

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Table 3 – The predicted and the measured natural frequencies

| Mode	Test (Hz.)	Type1 Freq.(Hz.)	Type1 Diff.	Type2 Freq.(Hz.)	Type2 Diff.	Type3 Freq. (Hz.)	Type3 Diff.	Type4 Freq.(Hz.)	Type4 Diff.
1	16,09	11,30	29,79%	11,12	30,91%	11,07	31,22%	12,39	23,02%
2	30,96	29,79	3,79%	27,46	11,31%	19,04	38,51%	39,61	27,93%
3	42,83	41,49	3,12%	29,29	31,61%	27,89	34,87%	46,38	8,30%
4	50,56	48,27	4,53%	38,83	23,20%	30,72	39,24%	51,58	2,02%
5	52,71	50,58	4,05%	44,45	15,68%	32,11	39,09%	54,39	3,18%
6	57,39	53,74	6,63%	48,99	14,63%	39,22	31,66%	64,63	12,62%
7	60,67	60,30	0,61%	50,44	16,86%	43,63	28,08%	68,93	13,62%
8	68,98	70,39	2,04%	53,31	22,72%	45,06	34,68%	72,61	5,26%
9	71,60	72,12	0,73%	57,62	19,52%	45,73	36,13%	79,82	11,49%
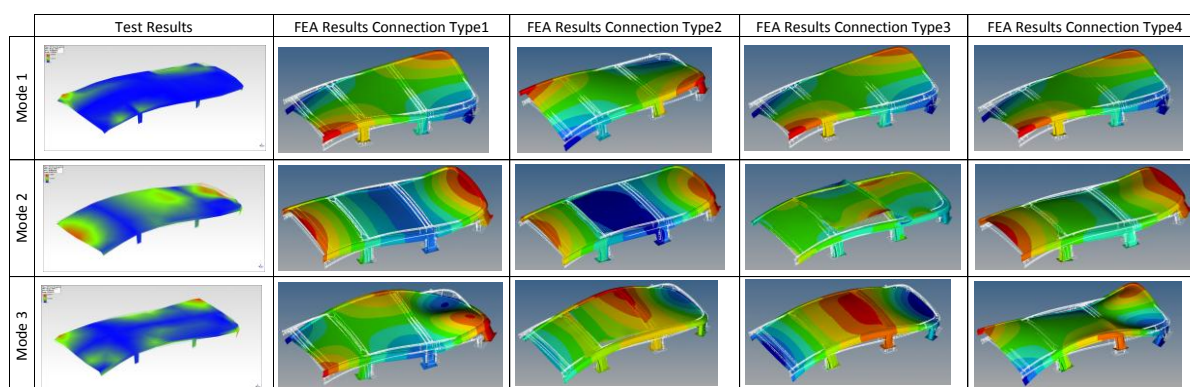
Figure 15 – The predicted and the measured mode shapes of MTSwRSG

4. CONCLUSIONS

In this study, a suitable linear FE model of structures with rubber sealed glass is built. This modeling technique, adjusted by the measured modal properties of this structure, has been achieved by comparing natural frequencies and their mode shapes. However, these modal parameters may depend on types of sealing method and properties. This study also gives an opportunity to examine the effects of the sealing properties to select.

The study also reveals that a simple rigid element (RBE2) can be used to represent rubber seal in quick prepared models to obtain less reliable but faster and/or preliminary results. However, the analyses, performed with this type of connection, is to be repeated with a new model having the suggested connection type in this study to obtain reliable results.

It is worth to mention that, to avoid the differences in first mode, the design of the rig is to provide more elastic boundary constraint due to the dynamics of the experimental rig and material characterization. Modeling techniques of rubber seal will also be studied and optimized due to observe more appropriate results in FE analysis.

ACKNOWLEDGEMENTS

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