

	Green & Innovative Lubrification Devices	<u>Society</u> : GDTech
	<i>Project Periodic Report Final (years 2013 to 2016)</i>	<u>Study ref</u> : 267617-GILD <u>Indices</u> : -


GDTech

Final Project Report

(years 2013 to 2016)



<p><u>Written by:</u> Th. WALTREGNY (LEAR)</p> <p><u>Date:</u> 30th March 2017 (update)</p>	 Visa
<p><u>Approved by:</u> M. Bruyneel (Scientific Director)</p> <p><u>Date:</u> 30th March 2017 (update)</p>	 Visa

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<u>Status of the issues</u>	
Issues and references	Incorporated modifications
.....
.....
.....
.....

<u>Recipient</u>	
Society	Name
Safran Aero Boosters (Safran group)	Laurent ARNAUD (Topic Manager)
.....
.....
.....



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1. Global project objectives

Preliminary notes:

- the project is issued from the "JTI-CleanSky-2009-2-SAGE-02-003" Call for Proposal (CFP)
- initial project starting date was set around may 2010
- however, due to significant lack of availability for the inputs (from Safran-Snecma), the Topic Manager (Safran Techspace aero) has decided to postpone the project's starting date until January 2013 (see Amendment #1 of the Grant Agreement)
- moreover, according with Topic Manager (TM), in order to realize the updated work scope of the 3rd period, the project has been extended to 31/12/2016 (see Amendment #2 of the Grant Agreement)

1st period:

According with the previous planning shift and as defined in the update Description of Work (DoW), the objective of 1st period was to achieve analytical and numerical engineering assessment on the <TRL2 lubrication devices.

note: the selected equipment was the heat exchanger

At the beginning of the project, GDTEch has been invited to be present on the Topic Manager (Techspace aero) site of Milmort (Liège – Belgium).

At that time, Techspace aero had not many experiences about the heat exchanger, it has so been decided to start from "scratch".

2nd period:

According with the update Description of Work (DoW), the objective of 2nd period was to achieve CAD design and numerical engineering assessment on the <TRL4 lubrication devices.

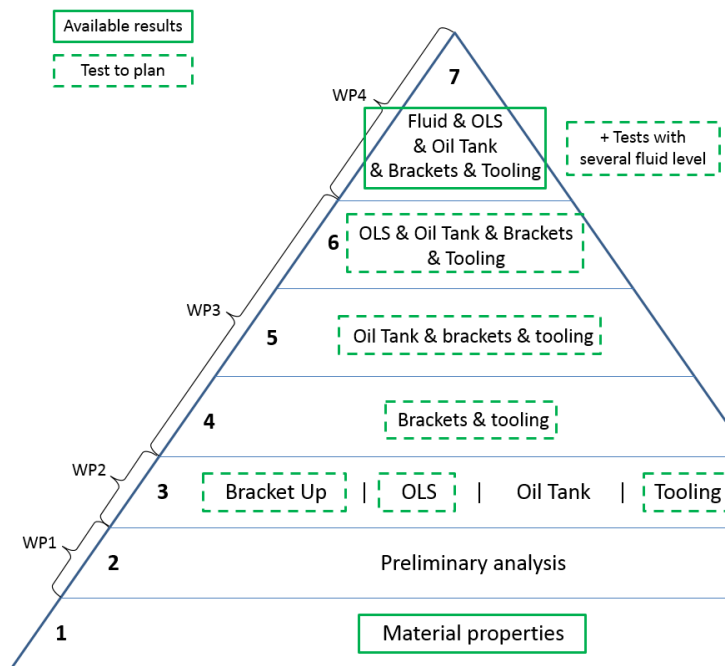
note: the selected equipment was the oil tank


In order to achieve TRL4 stage, a vibration test tooling must be designed, assessed, reviewed and calibrated.

3rd period:

According to the update Description of Work (DoW), the objective of 3rd period was to lead FEM analysis to the TRL5 on lubrication devices. The 3rd period is focused on FEA and test correlations (pyramid bottom-up approach).

note: the selected equipment was the oil tank




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2. Detailed project objectives per period

1st period:

The following activities have been realized:

- Kick-off-Meeting
- Acquisition and analysis of the SAGE-2 specifications
- Research about the existing technologies and concepts (cf. from standard → State-of-the-Art → advanced proof of concept)
 - note: at that time, additive manufacturing processes have not been investigated
- Definition of the Heat Exchanger Development Plan → the goal was to determine the roadmap to follow in order to design and validate the equipment
 - note: according with TM, the associated deliverable is the "[D3 - Heat Exchanger Development Plan](#)"
- Thermal sizing → to goal was to determine a realistic thermal sizing
- Design activities → the goal was to have a realistic overview of the volume to be further integrated in the SAGE-2 engine environment.
 - note: according with TM, the associated deliverable is the "[D4 – 3D Models](#)"
- Numerical assessment → the goal was to develop numerical model in order to help the thermal sizing
 - notes:
 - at that stage, only the Computational Fluid Dynamics (CFD) approach has been selected by the TM
 - according with TM, the associated deliverable is the "[D5 – FEA/analytical results](#)"
- Concept Design Review (CDR)
 - notes:
 - according with TM, the associated deliverable is the "[D6 – TRL2 Review](#)"
 - intermediate technical meetings and reviews have been realized

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2nd period:


The following activities have been realized:

- a) Kick-off-Meeting of the WP3
- b) Definition of the FEA roadmap and test tooling specifications
note: according with TM, the associated deliverable is the "*D7 - Oil Tank Development Plan*"
- c) Iterative design activities of the oil tank HCF test tooling
note: associated deliverable is the "*D8 - Oil Tank Vibration Test 3D Models*"
- d) Numerical assessment of the HCF test set-up → the goal was to validate the dynamic behavior of the tooling. Moreover, the HCF living has been assessed (e.g. bolted assembly, welded joints...)
note: associated deliverable is the "*D9 - Oil Tank Vibration Test FEA report*"
- e) Concept Design Review (CDR)
notes:
 - intermediate technical meetings and reviews have been realized
 - associated deliverable is the "*D10 - Oil Tank Vibration Tool Concept Design Review*"

3rd period:

The following activities have been realized:

- f) Kick-off-Meeting of the WP4: definition of the FEA roadmap and required tests (with specification test)
note: according with TM, the associated deliverable is the "*D11 - KOM*"
- g) Vibration tests on single parts and on assembled parts in a laboratory. Comparison between numerical results and the experimental measurements. The best model procedure is identified at each step of the pyramid approach.
note: the associated deliverable is the "*D12 – Test Correlation*"
- h) Numerical assessment on the oil tank (single and assembled parts). Study of assembly conditions: implementation of gluing, bolted connection, clamping, ... Mesh study on each part. The more suitable model is identified for each studied assembly condition.
note: the associated deliverable is the "*D13 – FEA Optimisation*"
- i) Presentation of the main results
notes:
 - intermediate technical meetings and reviews have been realized
 - the associated deliverable is the "*D14 – TRL5 Review*"

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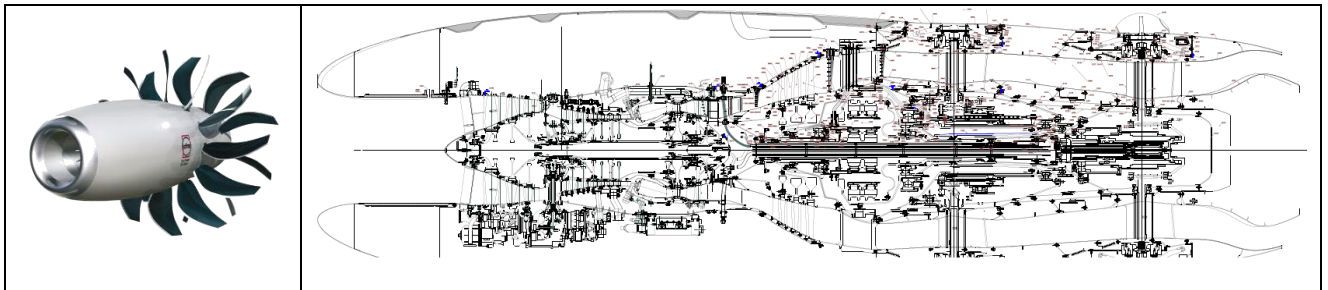
3. Work achievements per period

1st period:

During the 1st period (year 2013), activities have been achieved on 2 WPs:

- a) Management → planning (D1) & management plan (D2)
- b) TRL<2 engineering & assessments

Presentation of the Sage-2 engine demonstrator:



The SAGE-2 demonstrator is an Open Rotor engine. Basically, the gas generator is a modified M88 engine core, the counter rotating fan blades are located at the rear of the engine. Those are connected to the rotors through a gearbox located in the MSF section (downstream of the turbine and upstream of the fan stages). As well for GTF commercial engines (or even turboprop), the gearbox efficiency (even high) lead to have thermal loss that heat the oil that have to be cooled down through the heat exchanger (Air Cooled Oil Cooler). The design (compactness) and thermal efficiency of the ACOC are the main concerns to assess. Moreover, the heat exchanger will be located in the engine nacelle and is subject to high temperature and high vibration (HCF) environment.

Important note:


During the project, Safran-Snecma has informed Safran-Techspace aero (=TM) that the equipment (that had to be initially manufactured and delivered by Techspace) will not be directly installed in the nacelle (=engine environment) but in a slave position (away from the engine) → based on this decision, Techspace aero has inform GDTEch that the activity will remain on a study level.

a) Management tasks (WP1):

Because the GILD project is a single member one, the management activities have been focused on the setup of communication with the Topic Manager focal point (initially not identified). During the period, the level-1 planning (D1) and the Project Management Plan (D2) have been defined and submitted.

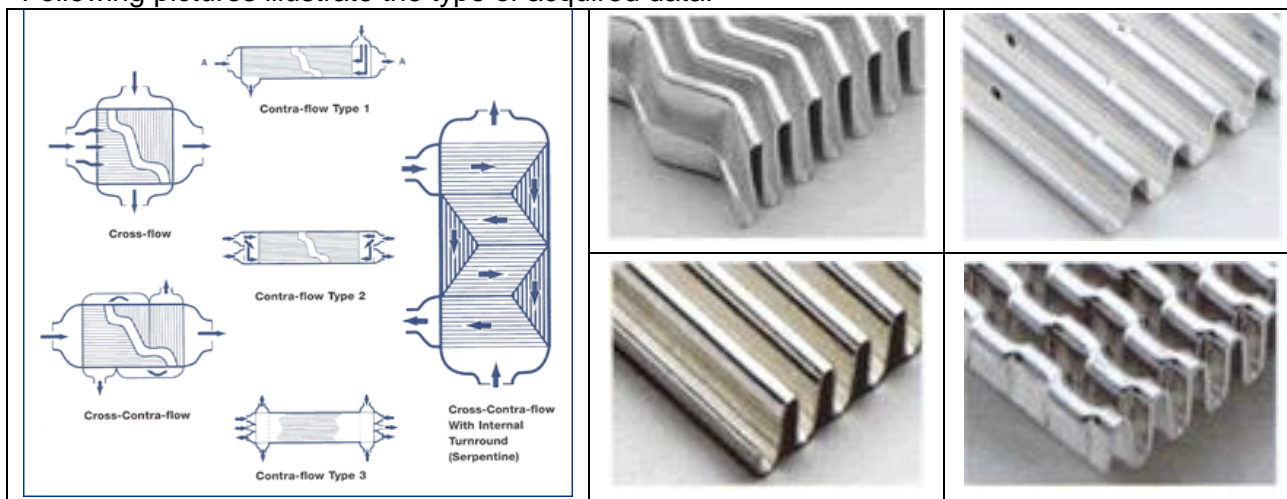
b) TRL <2 engineering tasks (WP2):

From an historical point of view, the Topic Manager (Techspace Aero Safran group) has not a strong experience legacy about heat exchanger.

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Documents research of the existing state-of-the-art has been realized covering technological aspects (e.g. materials, assembly, design, assessment...).

Following pictures illustrate the type of acquired data:



2nd period:

During the 2nd period (year 2014), activities have been achieved on 2 WPs:

- a) Management (coordination with TM)
- b) TRL<4 engineering & assessments

On Sage-2 demonstrator, the gearbox requires significant amount of oil flow rate. In addition, with the heat exchanger (see 1st period), **the oil tank is a major equipment in the oil system**. It has several functions, the 1st one is of course to manage the oil quantity but it also realizes the function of air-oil separator and oil monitoring.


The design (compactness) and mechanical behavior (including HCF) of the oil tank are the main concerns to assess.

The standard assessed mechanical conditions are:

- HCF behavior to correlate with tests {
- **Dynamic assessment** (frequencies & modes identification)
 - **Dynamic Response** (harmonic approach) assessment (with notching)
 - Thermal expansion load cases
 - Pressure load cases (including proof and burst pressures)
 - Fan Blade Out (FBO) load cases (_g static approach)
 - Assembly Pre-loads analysis

a) Management tasks (WP1):

Because the GILD project is a single member one, the management activities have been focused on communication activities with the Topic Manager focal point.

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b) TRL <4 engineering tasks (WP3):

From an historical point of view, the Topic Manager (Techspace Aero Safran group) has a strong experience legacy about oil tank but in commercial engines environment using damped solutions.

The oil tank design and validation require to:

- manage the "hard mounting" configuration
- manage the Finite Element Assessment correlation vs. tests

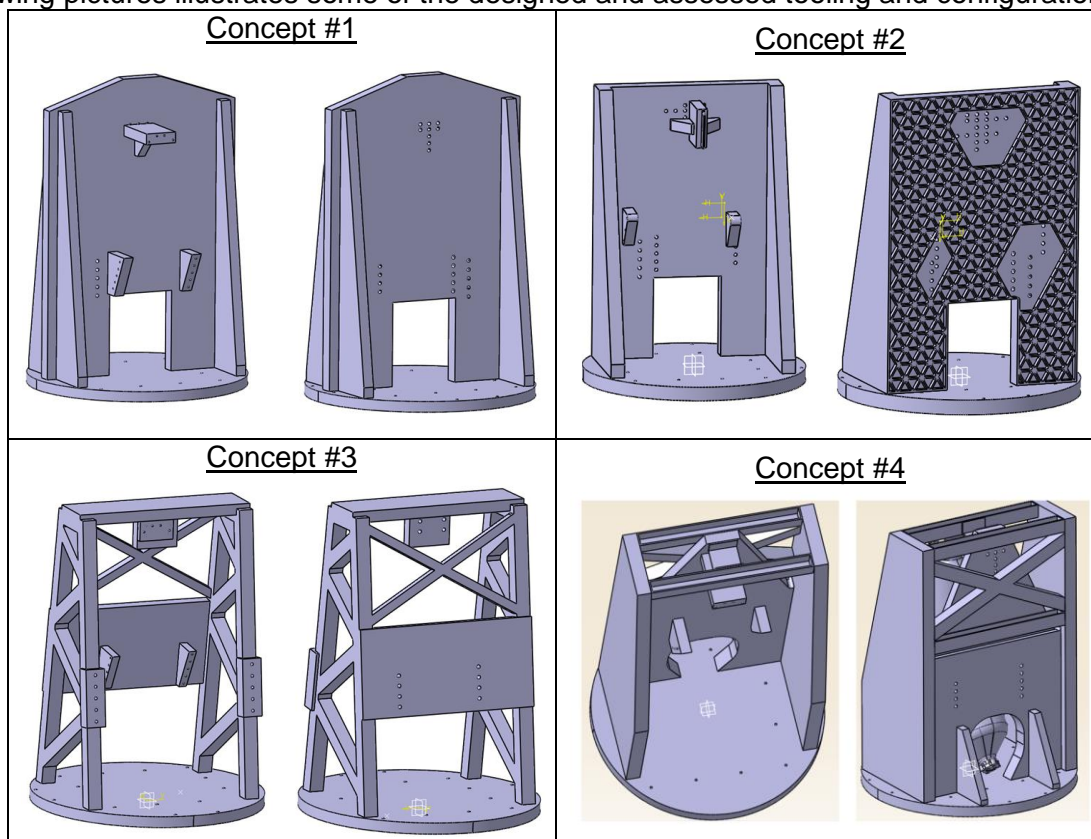
→ in order to achieve those challenges, it was mandatory to design and validate HCF dynamic test tooling.

b.1) Engineering & design:

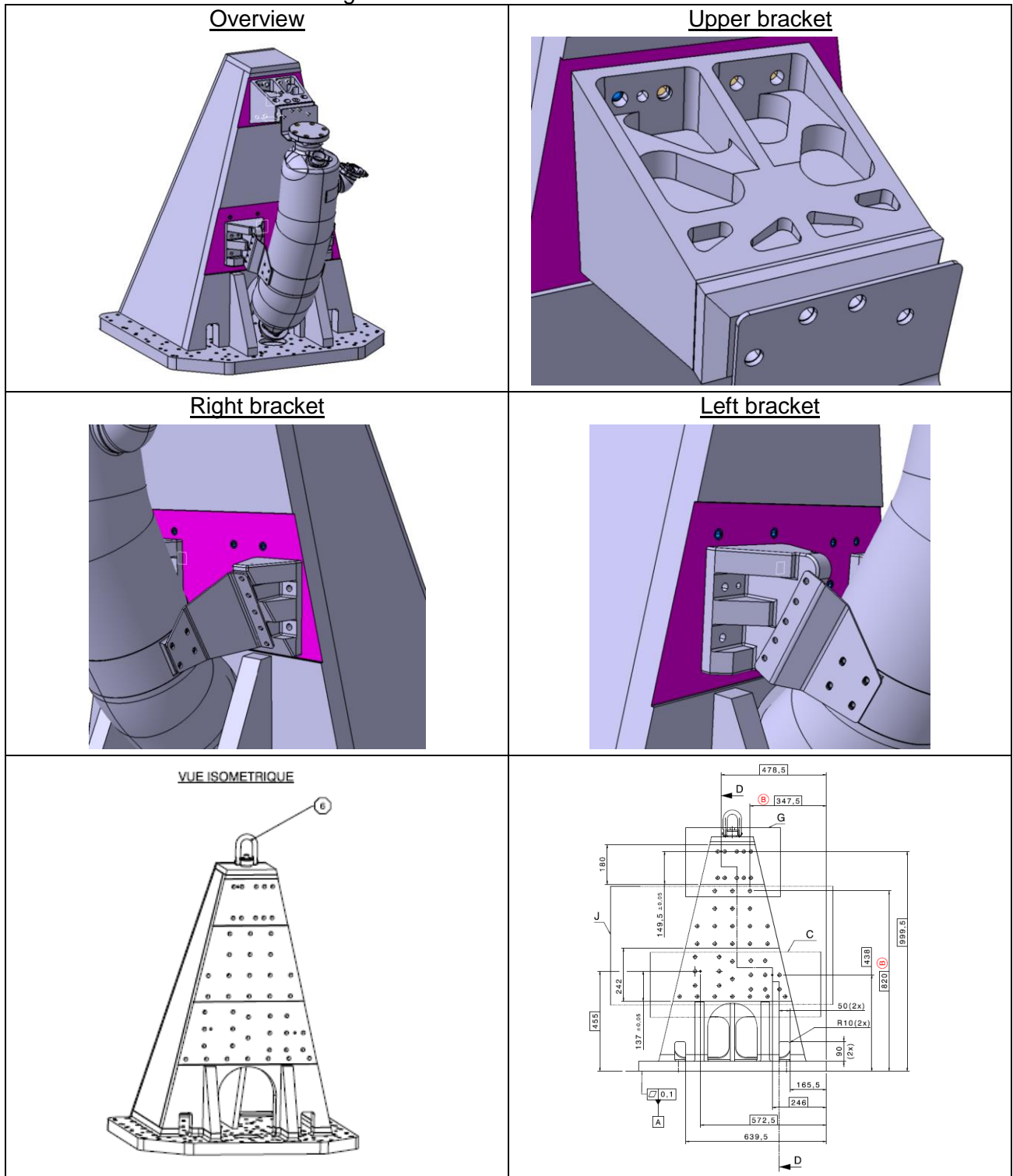
The main design constrain was set but the dynamic response of the tooling. Indeed, in order to interact with the genuine oil tank modes (especially the 1st ones at lower frequency), it was important to raise the tooling modes at higher frequency. This induces to have the stiffest tooling but also lightest one (cf. also limited by the shaker weight capability).

Several designs have been engineered and presented to the TM as trade-off review.


Following pictures illustrates some of the designed and assessed tooling and configurations:



Finally, the design has converged toward the "pyramid solution". The following pictures illustrate CAD model and drawings extracts:



note: see "[D8 - Oil Tank Vibration Test 3D Models](#)" for more details about CAD model and drawings.

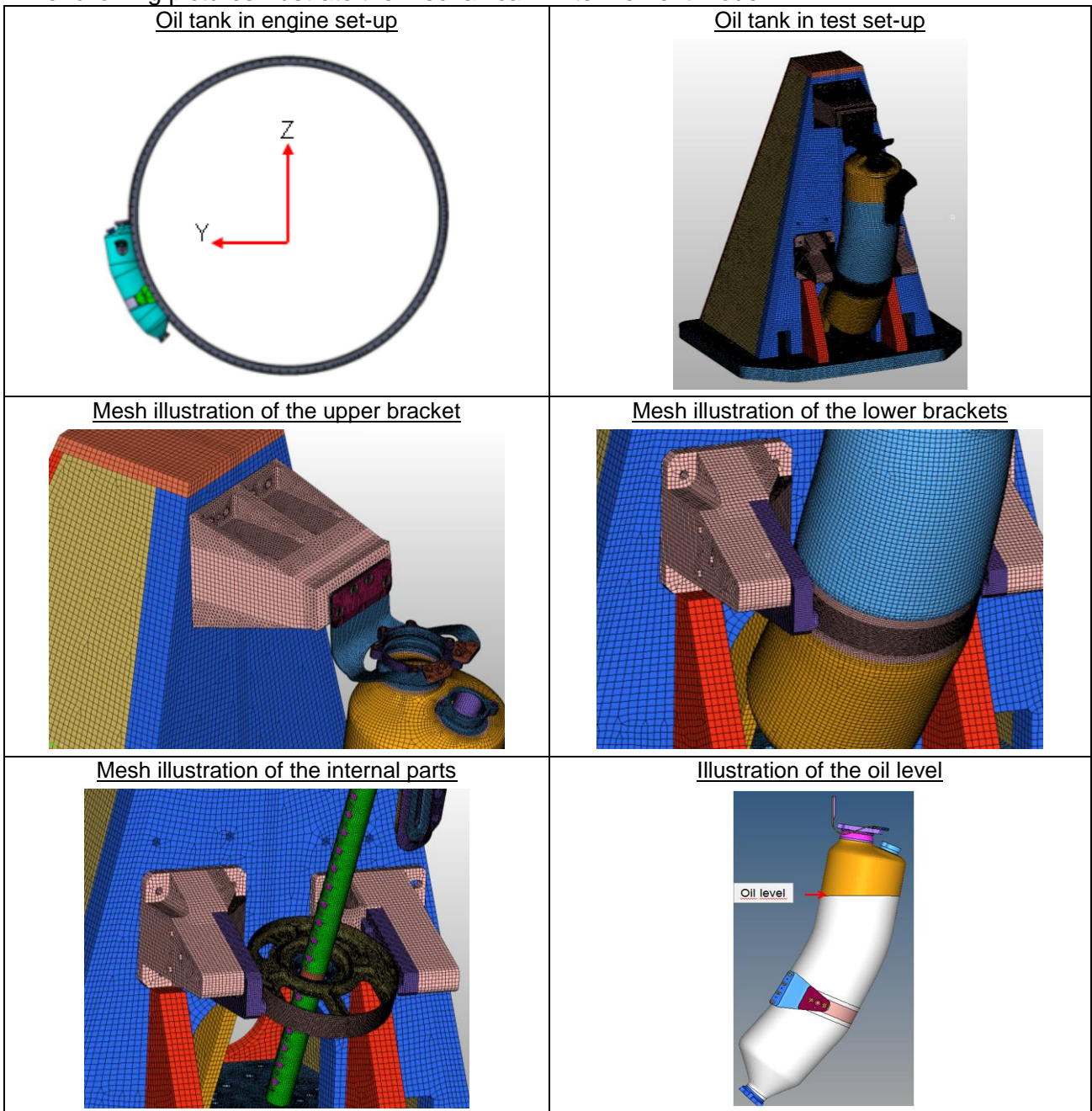
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b.2) FE numerical assessments:

During design process, numerical models have been achieved in order to optimize the 1st tooling frequency and to avoid anti-resonance behavior with the oil tank.

Moreover, the last converged "pyramid" design has been assessed in order to validate the tooling assembly (including bolted & welded joints) and the test set-up (including the oil tank).

The following pictures illustrate the mechanical Finite Element Model:



note: see "[D9 - Oil Tank Vibration Test FEA report](#)" for more details about model and boundary conditions.

b.3) Bolted joints validation:

M10		
vis métrique	10,00	mm
nombre de vis	25,00	
ampli	1,5	
Fe(force extérieur)	65138,4	N
Fe/syteme d'attache	2605,536	N
effort tranchant	44,92805038	Mpa
sigma traction	384	Mpa
Vomises Vis	391,8055502	Mpa
Marge 10,9	0,5646605	
Forcede frottement /boulon	3219,73085	N
Force de frottement	80493,27124	N
décollement	15354,87124	N

CSL
*1,5 car lors des essais le niveau de g au niveau du Rh sera limité à 30g = 20*1,5*
masse de l'outillage = 275 kg
Masse du Rh = 38 kg (leapx voir 57ST184261.9002)
par rapport à Re(640 Mpa)
si >=0 alors pas de décollement de la bride

Conclusion: 25 vis M10 (class 10.9) are sufficient and prevent the gap opening between the tooling and the shaker table.

b.4) Welded joints validation:

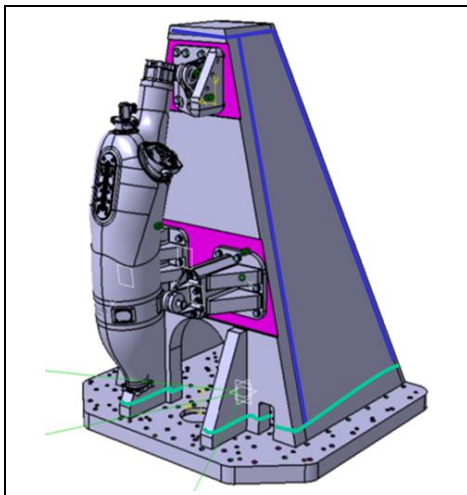


Table J.8 – Numerical values of $\Delta\sigma-N$ (N/mm²) for butt welded joints between members – detail categories as in Table J.7

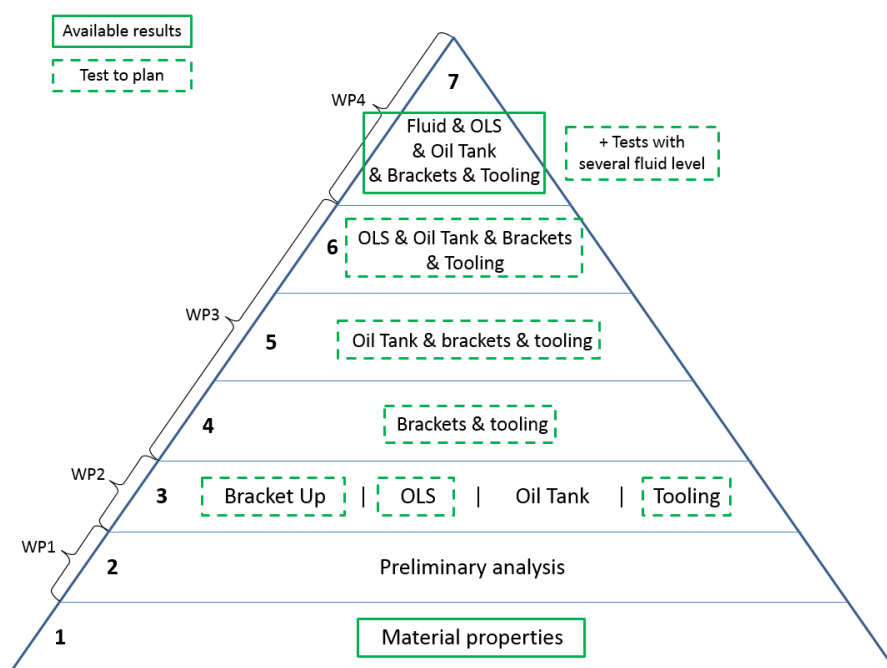
Slope		Cycles N						
m ₁	m ₂	1E+05	1E+06	2E+06	5E+06	1E+07	1E+08	1E+09
7	9	85,9	61,8	56,0	49,1	45,5	35,2	35,2
7	9	69,0	49,7	45,0	39,5	36,6	28,3	28,3
4,3	6,3	100,4	58,7	50,0	40,4	36,2	25,1	25,1
4,3	6,3	90,3	52,9	45,0	36,4	32,6	22,6	22,6
3,4	5,4	96,5	49,0	40,0	30,6	26,9	17,5	17,5
4,3	6,3	80,3	47,0	40,0	32,3	29,0	20,1	20,1
3,4	5,4	86,9	44,1	36,0	27,5	24,2	15,8	15,8
3,4	5,4	77,2	39,2	32,0	24,4	21,5	14,0	14,0
3,4	5,4	43,4	22,1	18,0	13,7	12,1	7,9	7,9

Conclusion: test limits have been defined to guarantee welding living integrity

3rd period:

During the 3rd period (years 2015 & 2016), the activities carried out for each WP are summarized:

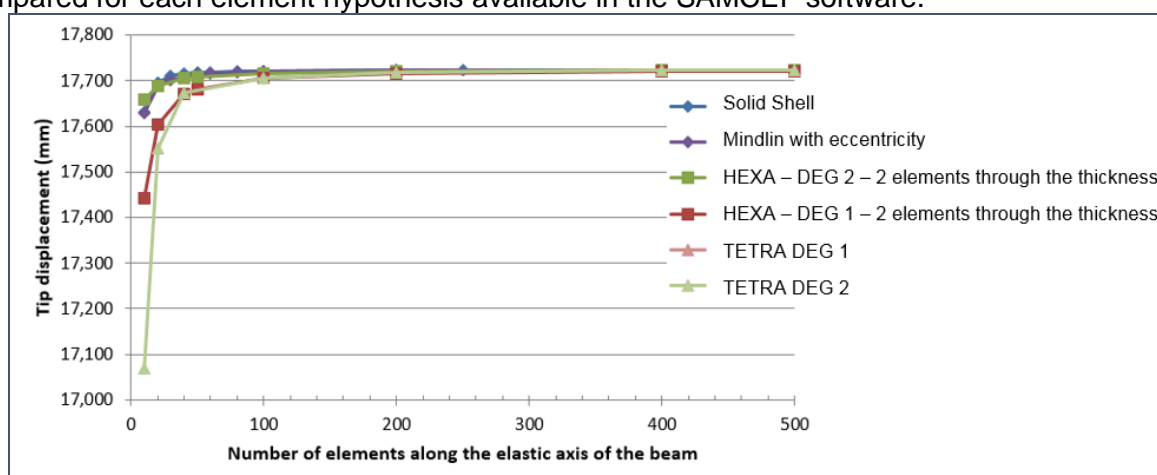
- a) WP1: Numerical investigation on academic cases → good-practice rules
- b) WP2: FEA on the single parts and vibration tests → correlation & optimisation
- c) WP3: FEA on assembled parts and vibration tests → correlation & optimisation
- d) WP4: vibration tests of the oil tank with the fluid



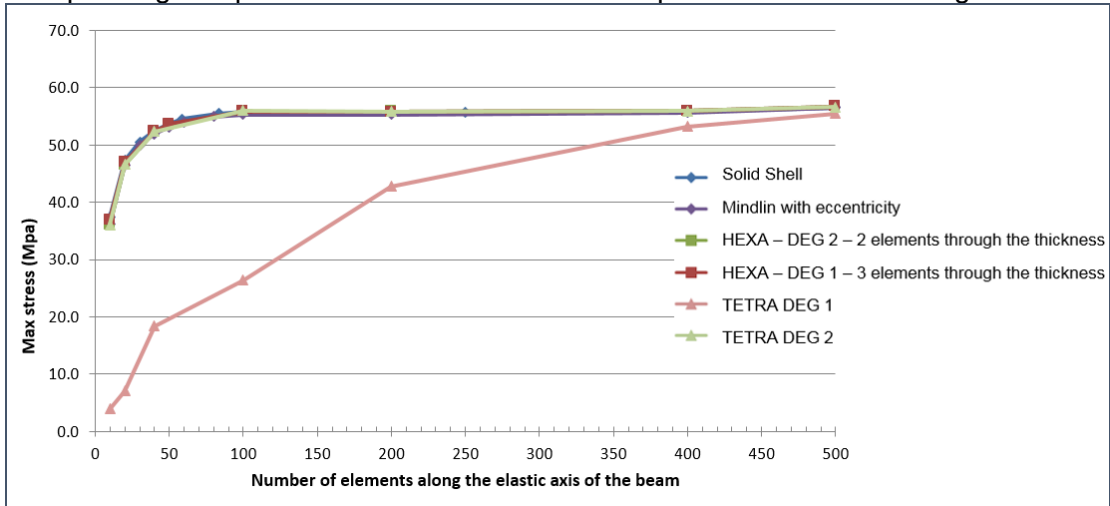
a) Numerical investigation on academic cases (WP1)

The considered academic case is a cantilever beam with a concentrated tip load.

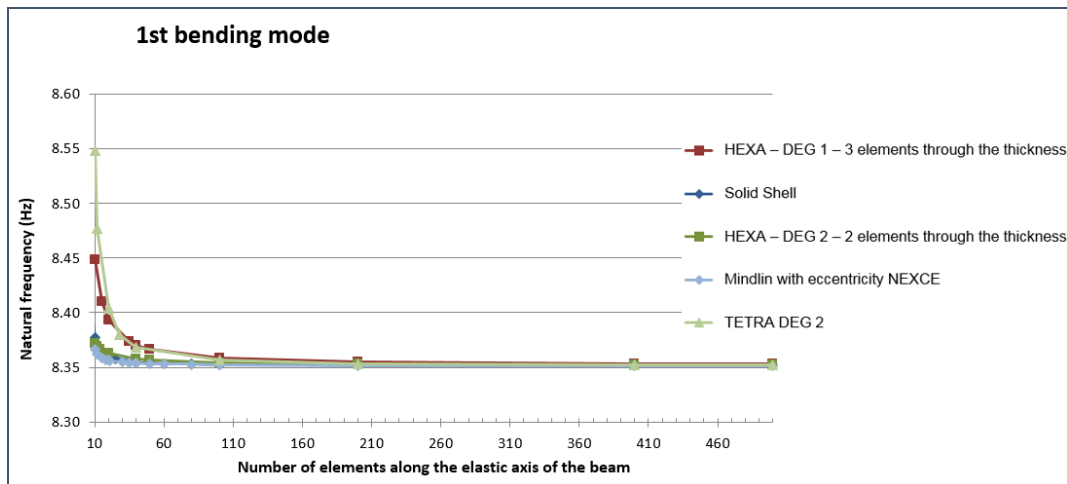
First, a convergence analysis is carried out to validate the mesh type. The tip displacement is compared for each element hypothesis available in the SAMCEF software.

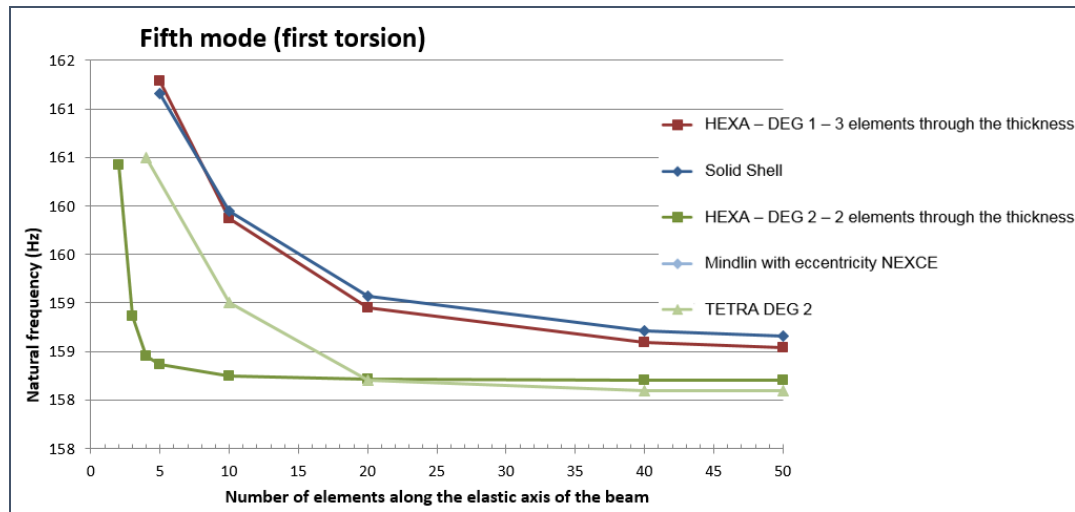


The corresponding comparison in terms of stresses is represented in the next figure.



The comparison in terms of natural frequencies is grouped according to the main deformation of the mode shape: the first bending and torsion modes are represented in the following:





Conclusions of WP1:


The mesh with HEXA DEG 1 must be created with a minimum of 3 elements through the thickness.

The mesh with HEXA DEG 2 must be created with a minimum of 2 elements through the thickness.

The use of the mesh with TETRA DEG 1 should be avoided: displacement, stress and natural frequencies are wrong.

The convergence of the TETRA DEG 2 mesh is slower than that of the HEXA DEG 2 mesh.

When the thickness of the beam is very small, “Solid Shell” and “Mindlin” hypothesis can be favourably used.

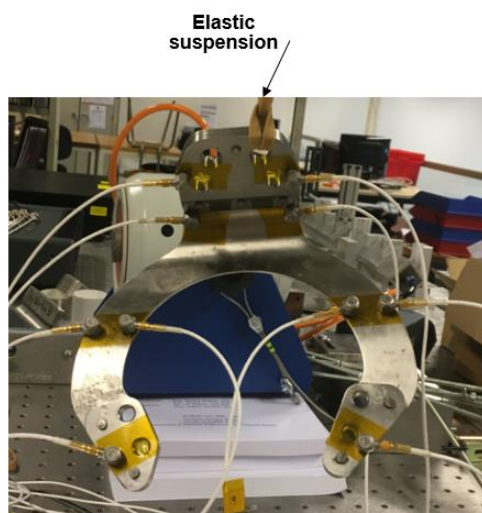
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b) FEA and vibration tests on single parts (WP2)

All the details about the experimental tests are in the deliverable "*D12 – Test Correlation*".
The numerical investigations for each part is detailed in "*D13 – FEA Optimisation*".

b.1) Upper bracket

Reference test for the correlation: modal analysis of an impact test in "free-free" conditions.



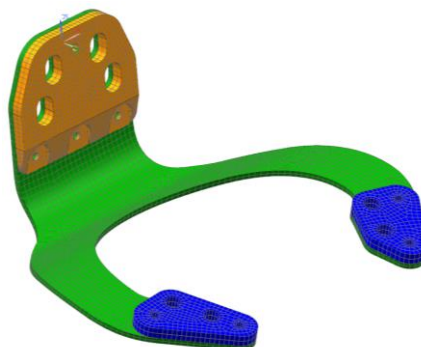
Test description:

- 10 single-axis accelerometers
- Frequency range up to ~1500Hz

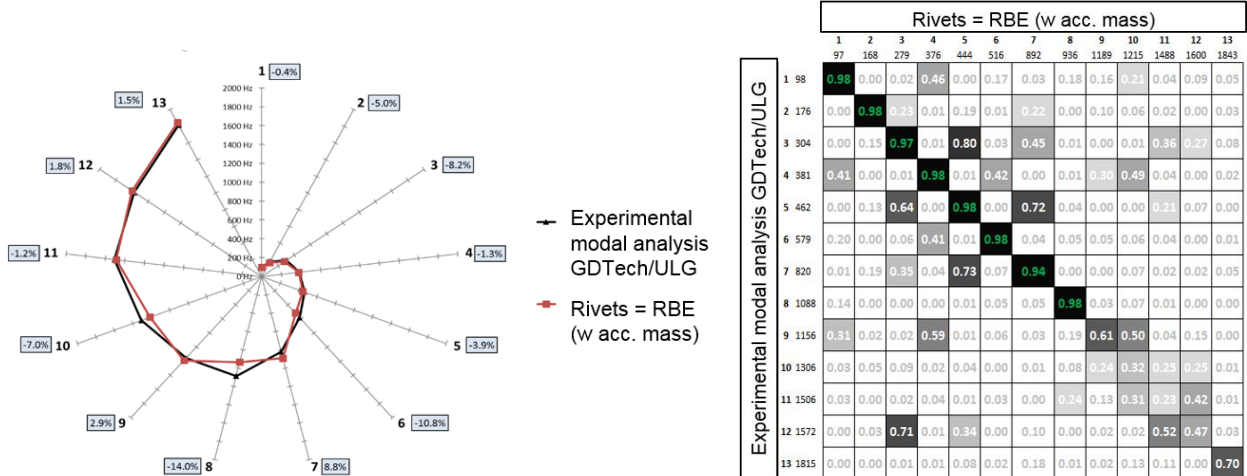
The model of the rivets should be realized with a rigid body link between a central node and the hole surfaces of the components.



The mesh study show that a mesh size of 2 mm is sufficient, with 2 elements through the thickness.



The results of the correlation are summarized in these 2 figures, in terms of a radar plot with the experimental and the numerical natural frequencies and a MAC matrix evaluated with the related mode shapes.

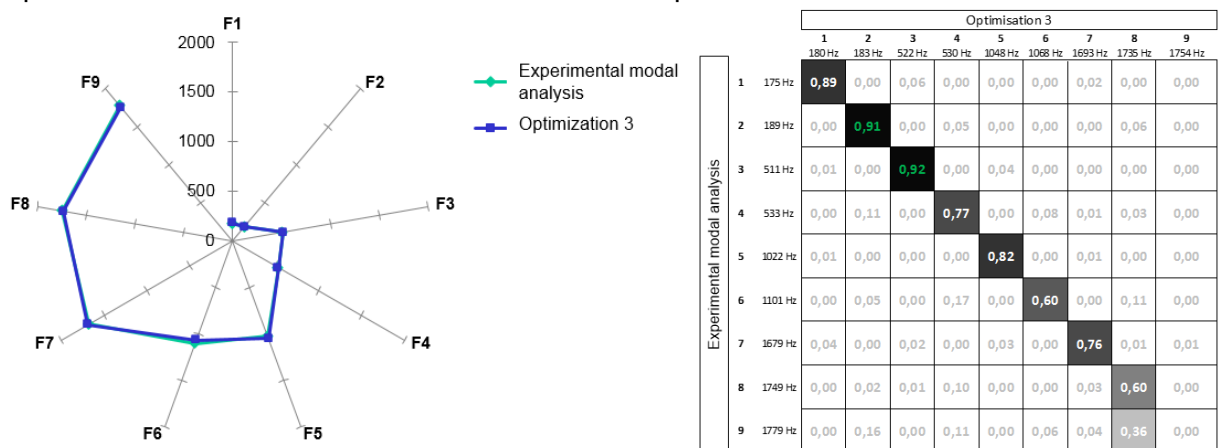



b.2) OLS (Oil Level Sensor)

The exact geometry of the OLS is unknown and complicated due to the presence of electronic components and stiffeners.

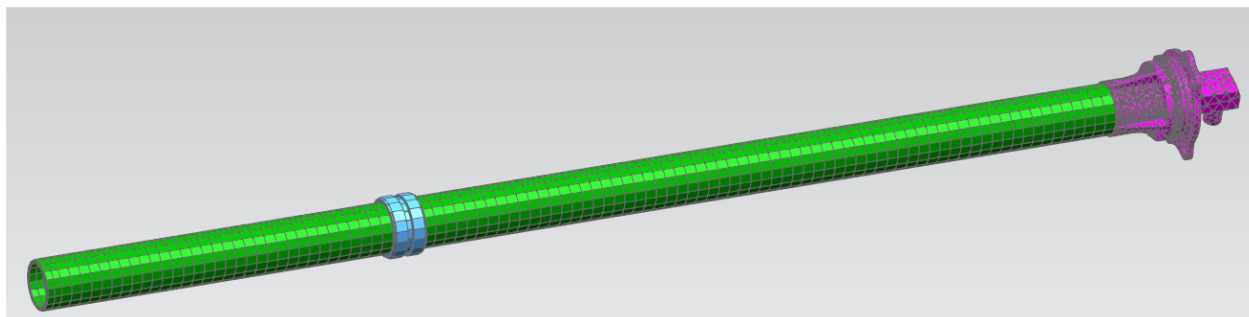


To approximate the vibration behaviour, an optimisation study was carried out on the material properties of the OLS to match the 9 firsts natural frequencies.



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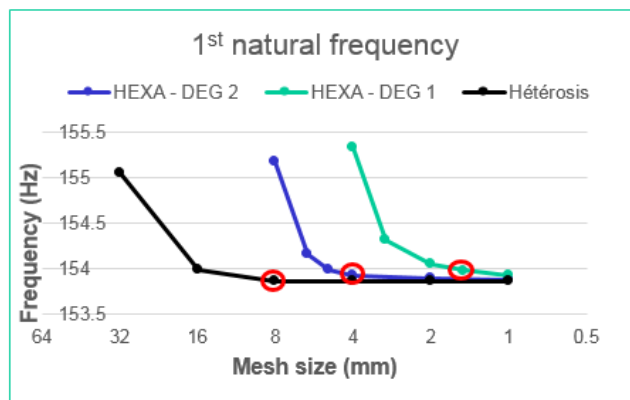
The mesh associates with this optimisation process is illustrated in the next figure (without stiffeners).



b.3) Oil Tank

As the geometry of the oil tank is very simple, no experimental tests have been performed. Only a numerical study was conducted.

Regarding the model of the oil tank skin, 3 element hypotheses were envisaged: solid hypothesis, shell hypothesis and solid shell hypothesis. The mesh convergence study is represented in the following to highlight the performance of each element type.



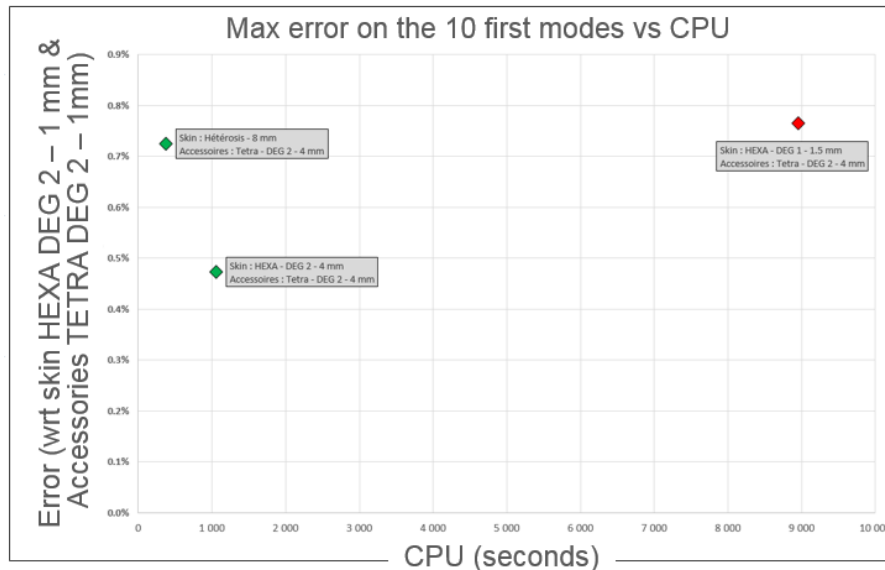
Hexa DEG 2 (4mm) // Hexa DEG 1 (1.5mm) // Heterosis (8 mm)

Based on the achieved numerical results, the « Solid Shell » and « Mindlin » hypotheses should be avoided.

A mesh study was carried out on the welded accessories (servicing panel, belt, interfaces, ...). A mesh size of 4 mm is selected to obtain reliable results.

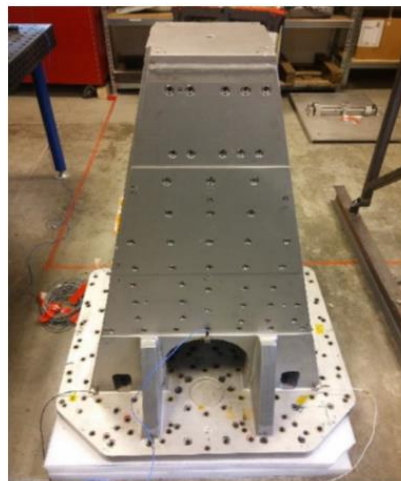


The choice of the most relevant modelling must be made based on the « Error – CPU » graph below.

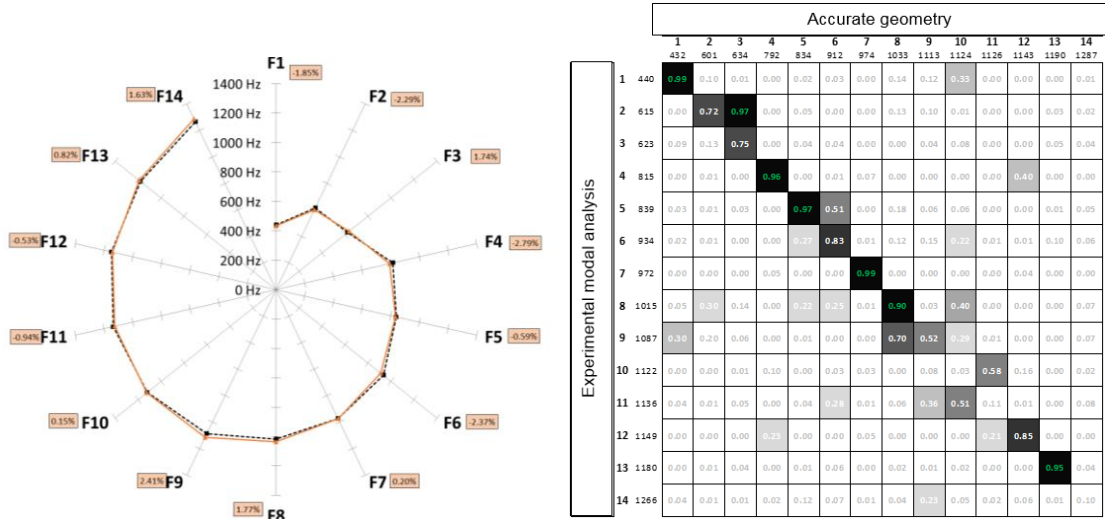


b.4) Tooling

The reference experimental results for the correlation are given by the modal analysis of the measurements collected during an impact test in “free-free” boundary conditions.



The correlation with the best numerical model is represented in the next figures. It is worth noting that the 3D geometry should not be simplified, otherwise the natural frequencies would be influenced.

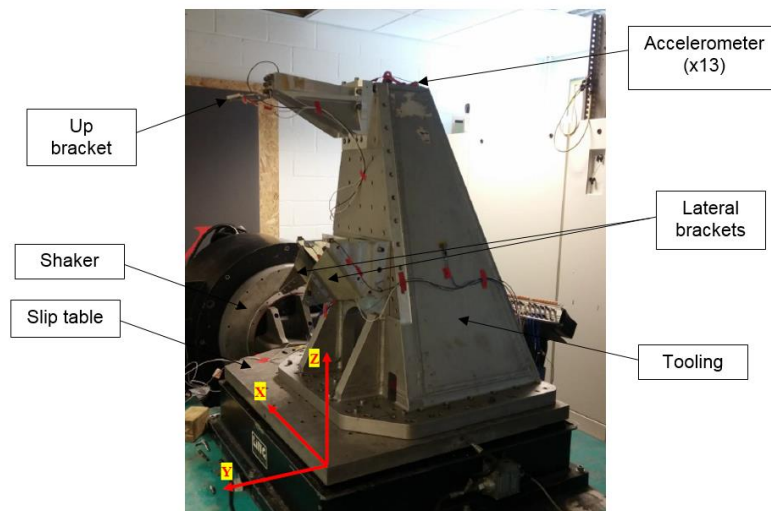


c) FEA and vibration tests on assembled parts (WP3)

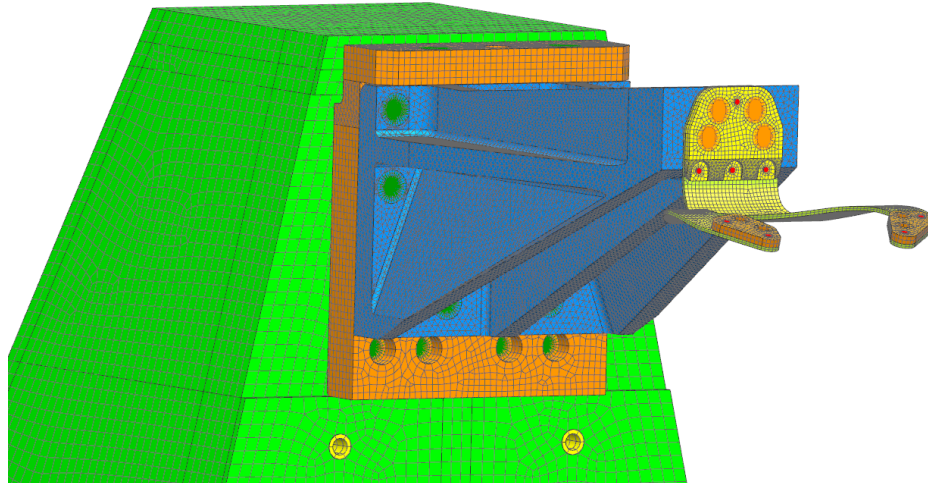
All the details about the experimental tests are in the deliverable "*D12 – Test Correlation*". The numerical investigations for each assembly are detailed in "*D13 – FEA Optimisation*".

c.1) Brackets & Tooling

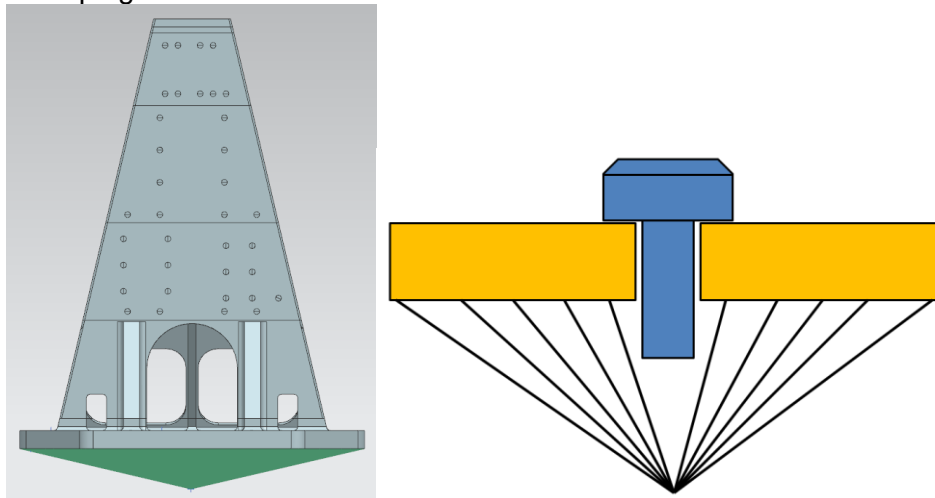
The reference tests for the correlation are three logarithmic sine-sweeps at low level (0.1g) along the 3 structural directions.



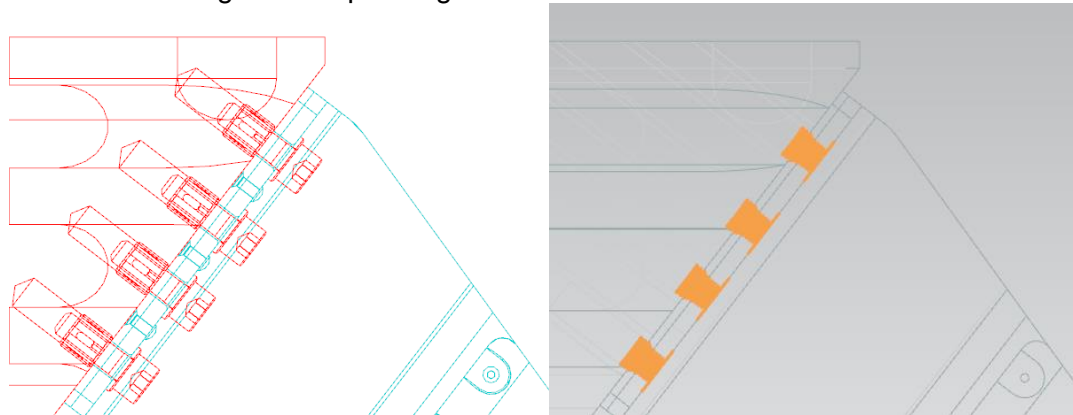
A mesh study of the bracket supports was realized. A mesh size of 4 mm was sufficient to give reliable results. The mesh of the assembly is illustrated in the next figure (tooling, support and bracket).



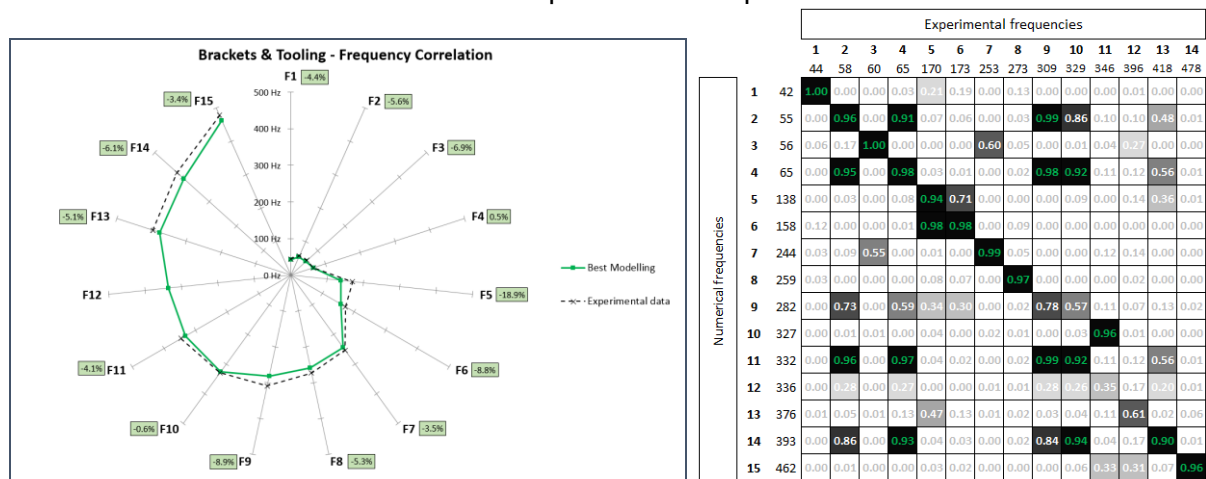
The model of the clamping of the tooling to the shaker interface was studied. The most relevant model is the clamping of the whole contact surface:



The way to connect the brackets and the supports to the tooling was also studied. The screw/insert assembly must be modelled with rigid links between the screw head surface and the drilling surface for a length corresponding to the first 3 threads of the inserts.

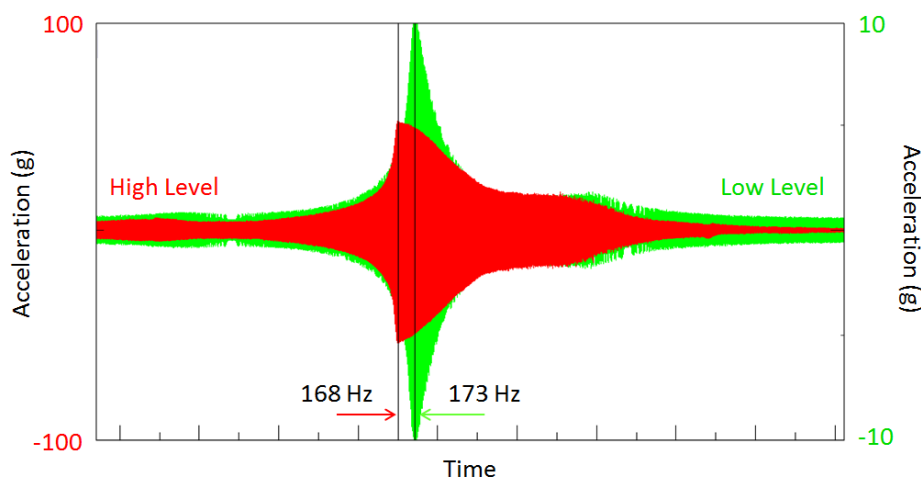


The numerical modes of this model are compared to the experimental data.



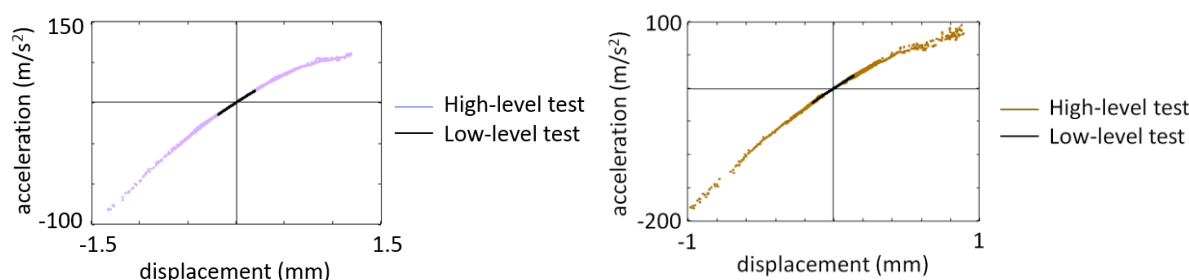
When multiple sine-sweep tests at different amplitudes of the base acceleration are carried out, the effects of the nonlinearities on the dynamic response of the assembly can be estimated. In presence of nonlinearities the modal parameters (natural frequencies, damping ratios and mode shapes) change with the amplitude of vibrations, and the correlation of the numerical finite element model becomes a challenging task. To this end, the effects of the nonlinearities introduced by the bolted connections are investigated to identify the dependency of the modal parameters on the vibration energy.

The identification of the nonlinear connections between each bracket and the tooling is carried out. The time responses of the accelerometers placed at the tip of each bracket recorded during the high-level vibration test are compared with the corresponding measures at low level. The following figure the time response at resonance measured by the accelerometer placed on the upper bracket is represented to compare the difference in frequency and in amplitude between the high-level and the low-level testes.



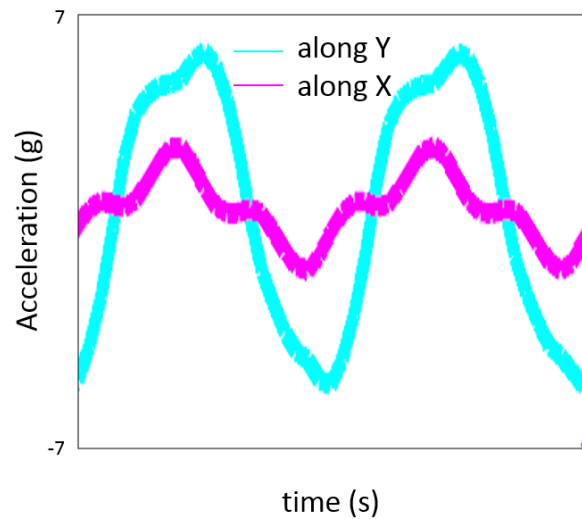
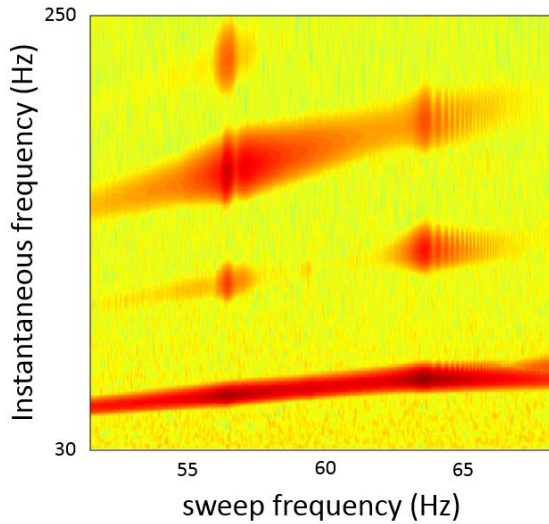
Although the vertical axes are properly scaled to compensate for the different excitation amplitudes, the difference of the peak values is remarkable: when the base acceleration is 1g the amplitude of oscillation reaches 45g, instead of the expected 100g (ten times 10g, which is the response when the base acceleration is equal to 0.1g). Moreover, the envelopes of the time responses follow two different trends around the resonance, whose frequency shifts of about 3%, from 173Hz to 168 Hz.

The Acceleration Surface Method is implemented between the sensor on the bracket and the sensor placed on the tooling in correspondence of the bracket to evaluate the nonlinear restoring force due to the bolt connections. Substantially, the results obtained from the data collected on the left and right brackets are similar and also coherent with the physical behavior of bolted connections.



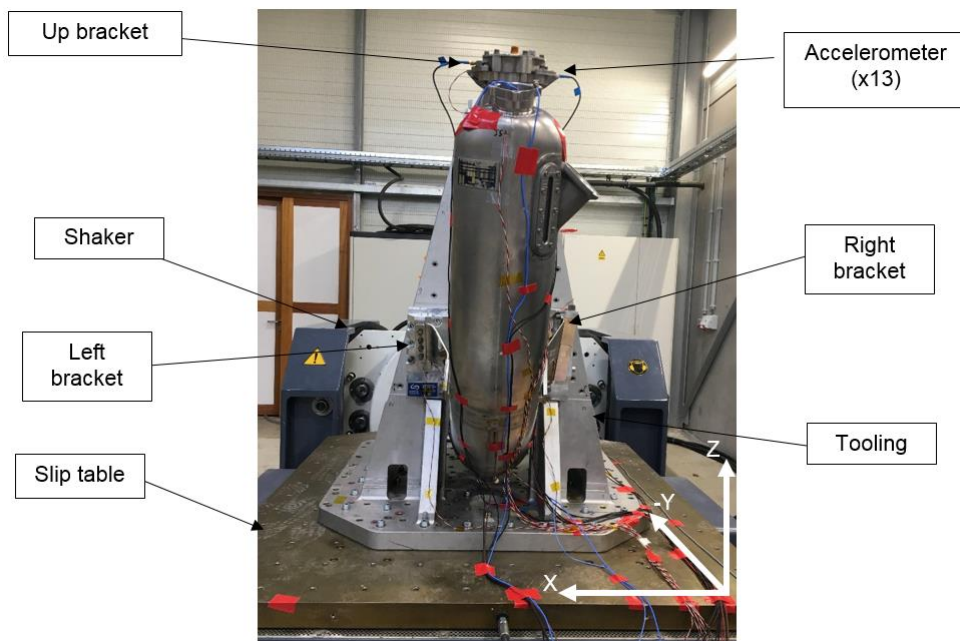
The figures highlight a softening behavior typical of loose connections along the in-plane directions.


Moreover, the response of the upper bracket shows an additional resonance between the first two modes at about 56Hz (as represented in the following figures). It is clearly visible that in this case the amplitude of the third harmonic is as high as the main harmonic. This means that a not negligible part of the energy is transferred to higher frequencies. Moreover, at 3 times the forcing frequency (that is about 168Hz) another mode of the bracket can be isolated, as already shown. It can be concluded that a modal interaction is occurring.



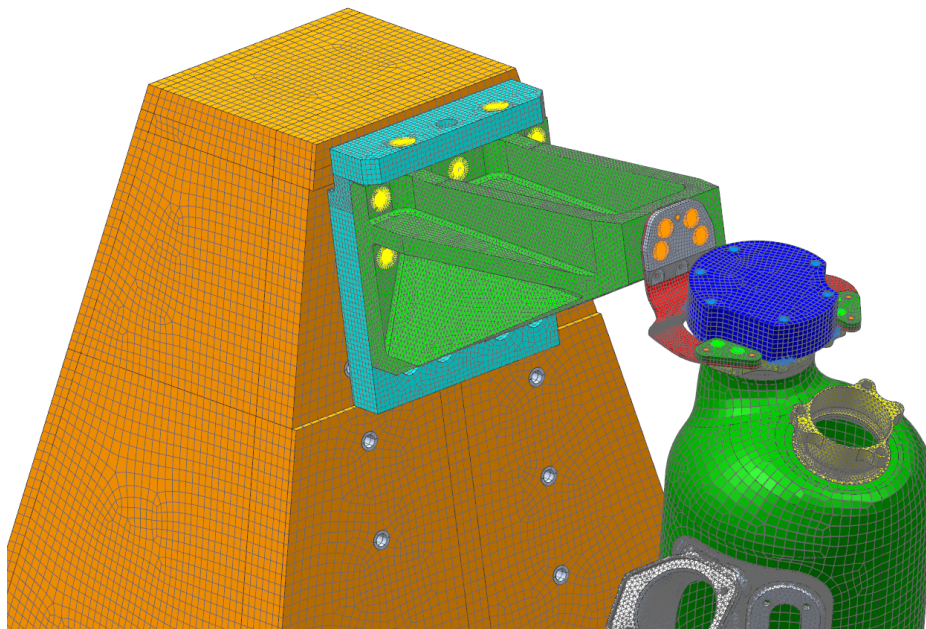
c.2) Oil Tank & Brackets & Tooling

The reference tests for the correlation are the logarithmic sine sweeps at low level (0.1g) along the 3 structural directions.



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The oil tank is meshed using the Heterosis elements, as advised after the numerical investigations of the oil tank skin. The mesh of the tooling, the support and the brackets is already described in the “Brackets & Tooling” assembly.



The first experimental natural frequency is higher than its numerical counterpart (about 10%). The model is more flexible than the real structure (according to the test). The experimental estimates suggest that the tooling and the supports do not participate to this mode, whereas the 3 brackets and the oil tank contribute to the mode shape. Conversely, in the numerical model also the supports of the brackets participate to the mode. Therefore, in the numerical model, the assembly between the brackets and the oil tank is not enough rigid. Similar remarks hold for the higher modes. It is recommended to further investigate to improve the model correlation with the experimental results. A possible solution could be related to the selection of a different way of assembling the parts in function of the stiffness of the single parts and of the connecting elements (rigid bodies, thin bodies, tightening torque, ...). Another recommendation is to study the influence of the nonlinearities as in the previous case (without tank).

c.3) OLS & Brackets & Tooling

The reference tests for the correlation are the logarithmic sine sweeps at low level (0.1g) along the 3 structural directions.

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The presence of the OLS during the test shifts the natural frequencies. When the mode shape is global (low oil tank elastic deformation), only the OLS mass influences the natural frequencies: they shift towards lower values. When the oil tank elastic deformation is not negligible, the OLS contributes to the stiffness of the oil tank and the natural frequencies shift towards higher values. The presence of accelerometers on the OLS during the test allowed the identification of some local modes of the OLS.

d) Vibration tests of the oil tank with the fluid (WP4)

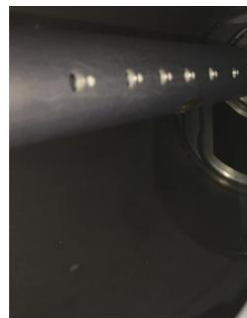
All the details about the experimental tests are in the deliverable "[D12 – Test Correlation](#)".

d.1) Fluid

The reference tests for the correlation are the logarithmic sine sweeps at low level (0.1g) along the 3 structural directions. Three fluid levels were tested (4 litres, 9.9 litres and 18.8 litres).



Level 1: 4.0 liters




Level 2: 9.9 liters



Level 3: 18.8 liters

The influence of the fluid level on the measured frequencies is reported in the next table.

	Level 1	Level 2	Level 3	Main contribution
F1	71.5 Hz	59.5 Hz	56 Hz 65.5 Hz	Bottom Oil Tank
F2	119 Hz	113 Hz	106 Hz 110 Hz	Oil Tank + Up Bracket
F3	137 Hz 141 Hz	133 Hz	118 Hz 120 Hz	Global (test X) Global (test Y)
F4	156 Hz	154 Hz	146 Hz 151 Hz	Global (test X) Global (test Y)
F5	160 Hz	158 Hz		Oil Tank + Up Bracket
F6	167 Hz	163 Hz		Oil Tank + Up Bracket (test Y)
F7	171 Hz	168 Hz	162 Hz	Oil Tank + Brackets (test X)
F8	181 Hz	174 Hz		Oil Tank + Left Bracket (test Y)

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F9		178 Hz	170 Hz	Oil Tank + Brackets (test X)
F10	202 Hz	183 Hz	182 Hz	Global (test Y)
F11	225 Hz	198 Hz	186 Hz	Global (test Y)
F12	233 Hz	209 Hz	194 Hz	Global (test X)
F13	280 Hz	266 Hz	266 Hz	Oil Tank + Brackets (test Y)
F14	287 Hz	287 Hz	287 Hz	Tooling + Brackets
F15	298 Hz	300 Hz	283 Hz	Tooling + Brackets
F16	301 Hz	281 Hz	272 Hz	Global (test Y)
F17	306 Hz	292 Hz	275 Hz	Global (test X)
F18	320 Hz	316 Hz	304 Hz	Tooling + Brackets
F19	333 Hz	326 Hz	323 Hz	Tooling + Brackets
F20	348 Hz	348 Hz	348 Hz	Global
F21	363 Hz	363 Hz	363 Hz	Global (test Y)
F22	376 Hz	376 Hz	376 Hz	Tooling
F23	421 Hz	413 Hz	397 Hz	Global

As the structure mass increases with the fluid level, the natural frequencies shift towards lower values.

Some peaks in the Frequency Response Functions are doubled for some fluid levels and the modal density changes with fluid level. These phenomena could be explained by the sloshing (moving liquid) and the interaction between the fluid and the structure.

d.2) Fluid & OLS


The test setup was the same of the test with “Fluid” without OLS.

The influence of the fluid level on the structure with the OLS is resumed in the next table. In this case the OLS is instrumented with only one accelerometer placed at its top tip (outside the tank).

	Level 1	Level 2	Level 3	Main contribution
F1	68.0 Hz	58.5 Hz	54.5 Hz 63.0 Hz	Bottom Oil Tank
F2	111 Hz	107 Hz	101 Hz	Oil Tank + Up Bracket
F3	124 Hz	112 Hz	105 Hz	Oil Tank + Up Bracket
F4	132 Hz	126 Hz	118 Hz 120 Hz	Global (test X) Global (test Y)
F5	157 Hz	153 Hz	145 Hz 148 Hz	Global (test X) Global (test Y)
F6	165 Hz	158 Hz	158 Hz	Oil Tank + Up Bracket

F7	169 Hz	169 Hz		Oil Tank + Up Bracket (test X)
F8	180 Hz	183 Hz	176 Hz	Global (test Y)
F9	196 Hz	179Hz	170 Hz	Oil Tank + Brackets (test X)
F10	216 Hz	191 Hz		Global (test Y)
F11	227 Hz	203 Hz	182 Hz	Global (test Y)
F12	265 Hz	266 Hz	266 Hz	Oil Tank + Brackets (test Y)
F13	280 Hz	281 Hz		Global (Test Y)
F14	286 Hz	285 Hz	283 Hz	Tooling + Brackets
F15	297 Hz	301 Hz	287 Hz	Global
F16	300 Hz 305 Hz	291 Hz	272 Hz	Global (test Y) Global (test X)
F17	317 Hz	317 Hz	303 Hz	Tooling + Brackets (test X)
F18	324 Hz	325 Hz	323 Hz	Tooling + Brackets (test Y)
F19	340 Hz 348 Hz	338 Hz 348 Hz	348 Hz	Global
F20	363 Hz	363 Hz	363 Hz	Global (test Y)
F21	377 Hz	377 Hz	375 Hz	Tooling
F22	423 Hz 434 Hz	414 Hz 420 Hz	392 Hz	Global

As the structure mass increases with the fluid level, the natural frequencies decrease. Some peaks in the FRFs are doubled for some fluid level and the modal density changes with the fluid level. These phenomena are exacerbated with respect to the tests without OLS: there could be an interaction between the fluid, the OLS and the oil tank.


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4. Deliverables & Milestones during the project:

Extract of the update Description of Work (DoW):

	Deliverable no.	Deliverable name	WP no.
	D1	schedule/planning	WP0
	D2	Management Plan KOM	WP0
Critical techno	D3	Development Plan KOM	WP1
	D4	3D Models	
	D5	FEA/analytical results	
	D6	TRL2 review material	
TRL4 techno	D7	Development Plan KOM	WP2
	D8	3D Models	
	D9	FEA results	
	D10	TRL4 review material	
TRL5 techno	D11	Development Plan KOM	WP3
	D12	Test correlation	
	D13	FEA optimisation	
	D14	TRL5 review material	
	D15	Project's archive	WP4

Deliverable n°	Title	Corresponding WP	Status
D1	Schedule/planning Version	WP1 (Mgt)	Submitted
D2	Management Plan	WP1 (Mgt)	Submitted
D3	Heat exchanger Development Plan	WP2 (year 2013)	Submitted
D4	3D Models	WP2 (year 2013)	Submitted
D5	FEA/analytical results	WP2 (year 2013)	Submitted
D6	TRL2 review material	WP2 (year 2013)	Submitted
D7	Oil Tank FEA Development Plan	WP3 (year 2014)	Submitted
D8	3D Models	WP3 (year 2014)	Submitted
D9	FEA/analytical results	WP3 (year 2014)	Submitted
D10	TRL4 review material	WP3 (year 2014)	Submitted
D11	Development Plan KOM	WP4 (year 2015)	Submitted
D12	Test Correlation	WP4 (year 2016)	Submitted
D13	FEA Optimisation	WP4 (year 2016)	Submitted
D14	TRL5 review material	WP4 (year 2016)	Submitted

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Milestones n°	Title	Corresponding WP	Status
M1	KOM–WP2 (year 2013)	WP2 (year 2013)	Passed
M2	TRL2 review	WP2 (year 2013)	Passed
M3	KOM–WP3 (year 2014)	WP3 (year 2014)	Passed
M4	TRL4 review	WP3 (year 2014)	Passed
M5	KOM	WP4 (year 2015)	Passed
M6	TRL5 review	WP4 (year 2016)	Passed

5. Project management during the period

Because the GILD project is a single member one, the management activities have been focused on the setup of communication with the Topic Manager focal point.

6. Dissemination

According with the DoW, several dissemination intents have been achieved and will be (at least during about 12 months after project end).

However, according with IPR restrictions (see Topic Manager), those have been limited to (public) generic concerns and not detailed with figures.

Nevertheless, 2 main dissemination processes have been realized:

- the project and objectives have been presented to the **ENOVAL (FP7) executive board** (MTU, Safran, RR, Avio, ITP,...)
- the project and objectives have been presented during technical & commercial meeting towards several companies (majors, SME) in different activity domains (including automotive,...)

Moreover, our intent is to publish and present a scientific paper presenting the GILD project benefits and to present to a Siemens software user conference.

Important note: in order to cover the final multi-physic correlation, the topic will be achieved through 2 further national R&D projects.

7. Encountered issues

At the end of the GILD project (December 2016), all the experimental tests were carried out. However, at that stage, the correlation study is limited to “Brackets & Tooling” step of the pyramidal approach.