SHOCK AND ELASTIC WAVES IN SPACE STRUCTURES: SIMULATION, ATTENUATION AND USAGE FOR MONITORING

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OVERVIEW AND BACKROUND

In the work presented, mechanical shocks, that means highly transient vibrations, also characterised by their high modal density are presented. These shocks may be generated by the spontaneous release of stored strain energy in hold down and release mechanisms or by the spontaneous transition of kinetic energy into potential energy as observed in end stops of actuators or cinematic devices. Due to the weak damping of space structures and the optimised mass/stiffness ratio this shocks may lead to a high vibration load to the subsystems mounted onto the primary structure. Subsystems mainly endangered are sensors, electronic, optics and also mechatronical systems.

A further aim is the simulation of shock loading in the early design phase to establish a subsystem placement within the satellite structure that takes into account the shock robustness of the S/Ss and the distribution of the shock loading within the structure. Mainly empirical or extrapolation procedures are used nowadays as a full transient FE-Simulation of complex structures is still inefficient. A numerical method is presented combining the advantages of numerical treatment, experimental tests, and statistical considerations. Using the shock response spectrum as the basis of the simulation the results can be directly interpreted and used for the subsystem qualification. A further extension w.r.t. the classical statistical energy analysis is the implementation of friction coupling and the normalisation of the subsystems coupling to their mode shapes, exactly spoken to the amplitude of each mode at the coupling point.

A common aim of the space industry is the reduction of the shock loads direct at the source, for example by low shock or even no shock release mechanisms [7].
A structure integrated shock absorption element as well a bumper and isolator combining visco-elastic and frictional damping are presented. Numerical simulations and experimental tests indicating the efficiency of the developed shock damping elements are presented.

Besides having an endangering potential shock waves can also be of advantage when using them for monitoring of mechanisms and structures. It will be shown that shock waves can be used to monitor the wear of coated friction couplings. Furthermore it will be shown that the elastic waves generated by mechanisms, e.g. by stick-slip phenomena, allow a damage pre-indication and a damage localisation. The time-of-travel of elastic waves to a spread sensor system is used in a cryo-vacuum friction test stand for monitoring two independent friction pads, also called narrow support elements (NSE).

The shock relevant components on a space structure and the methods for damping, monitoring and simulation are summarized in FIG 1.

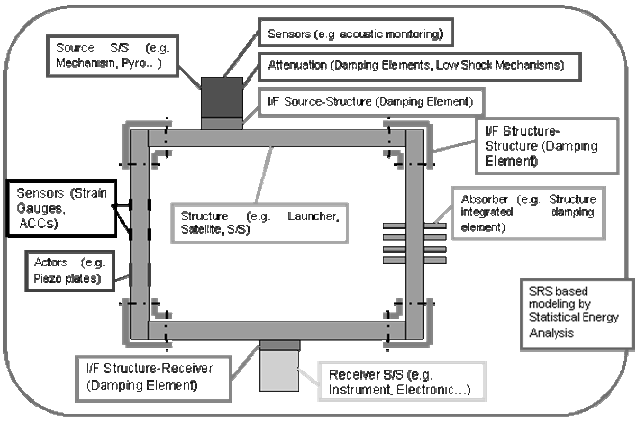


FIG 1. Shock relevant components in space structure

1. MOTIVATION

The motivation for the three aspects of shock and elastic waves in space structures can be summarised shortly:

1.1. Simulation

Dynamic simulation of space structures are performed for the three main load cases:

- low frequency harmonic and transient response
- high frequency harmonic response
- high frequency transient response (shock response, 100Hz to 1MHz)

The simulation methods used for the various frequency domains are mainly:

- Low/Mid Frequency: Hybrid FE-SEA Model
- High Frequency: SEA Model
- Low/Mid Frequency: FE-BEM Model

The following procedures are available:

- Empirical methods, e.g. extrapolation with scaling on shock energy, S/S distance and interfaces, used are based on experimental data of “similar” structures and may be false by a factor of 10. (Ref NASA).
- Finite Element Models of a complex space structure are inefficient for high frequent transient simulations, having also the problem of interface modelling and interpretation.
- Classical statistical energy analysis (CSEA) is a good compromise between numerical effort and detailed modelling. Still having the disadvantages of neglecting the transient character, by using the time

averaging Fourier transformation and the uncertainties of the models loss factors. Various modifications as, e.g. the transient SEA (TSEA) have been established to overcome this limitation.

A simulation method is proposed that can be easily used in the early design phase combining statistical modelling, transient character and practical interpretability of the results gained.

1.2. Isolation and attenuation

The need for shock attenuation is based on the following developments [1][2]:

- increasing pyrotechnic power of separation mechanisms
- increasing hold down forces
- increasing difficulties to place S/S in the necessary distance to the source
- increasing shock sensitivity due to miniaturisation
- increasing number of sensitive S/S as micro-electronics, optic and ceramics
- increasing lightweight design with low damping materials

The total shock generated e.g. by release mechanisms can be subdivided as follows:

- ~50% mechanical impacts and end stops
- ~40% spontaneous release of preload energy
- ~10% pyrotechnics

The current developments in shock attenuation are presented in the following.

The primary goal is the reduction of the shock generated. Therefore a huge effort is being made w.r.t. the development of low shock mechanisms, which are hold down and release mechanisms, separation rings and other gadgets using pyrotechnic devices [9].

- SMA (shape memory alloys) → QWKNUT (Starsys Research), Low Force Nut (Lockheed Martin) and the Two Stage Nut (Lockheed Martin) :200g – 500g
- Fuse wire → NEA Release unit (NEA Electronics) 350 g's @ 35,000 N preload
- „Slow“ transition from potential energy into kinetic energy by a planetary roller nut → Low Shock Release Unit (LSRU) (EADS Astrium GmbH): <500g at 16kN preload [7]

A visco-elastic end stop (bumper) has been developed for the use in 16kN LSRU (EADS Astrium). The mechanical impact of a magnetic actuator as well of a lever arm has been attenuated, so reducing the overall shock generated by 50%. Visco-eleastic and structural (friction) damping has been used by means of a specific topology and geometry optimisation.

In a later design phase the shock load distribution in the space structure has been determined by shock tests and the evaluation of the SRSs. If it is then necessary to increase the shock resistance of the S/S isolation may be the only solution.

Isolation w.r.t. shock load is limited as

- The stiffness and strength of the connection decreases.
- The S/S displacement due to shock excitation increases.
- The natural frequency of the S/S mounted to the structure is decreases.

A structure integrated shock isolation element has been developed for the isolation of actuators combining visco-elastic and structural (friction) damping and so minimising the limitations mentioned above.

1.3. Monitoring

Elastic body waves can be analytically described by the wave equation:

$$\frac{\partial^2 u}{\partial t^2} = c^2 \frac{\partial^2 u}{\partial x^2}$$

This equation, connecting the second derivates of the elongation/distortion *u* with location *x* and time *t* by the sound speed, gives the basis for various monitoring concepts based on elastic body waves.

Typical applications are impact detection and location in plate structures by lamb waves [3]. The elastic body wave is thereby generated externally by piezo plates in a controlled way.

An alternative monitoring concept is presented which uses the elastic body waves generated by the S/S to be monitored itself. Most mechanical processes are generating elastic waves. These waves can be acquired and used for the evaluation of the quality and quantity of the process of interest. In the work presented a shock based monitoring system is set up that can be used in space environment (ultra-high vacuum and 77K). Several sensor types and signal analysis procedures have been tested and evaluated using the data gained in a cryo-vacuum test stand for friction pads under very high normal load (up to 170to).

2. SIMULATION

2.1. Basics of Statistical Energy Analysis

Statistical Energy Analysis is mainly used in the high frequency range up to more then 10kHz, where other standard methods as for example the Finite Element Method (FEM) of the Boundary Element Method can not be applied reasonably. At high frequencies, the characteristic wavelength of the propagating vibration waves become much smaller than the overall dimensions of the structure and a very fine mesh is necessary in FEM, thus yielding very large models and very long computation times. Today the statistical energy analysis (SEA) is the most famous energy-based method. Although similar approaches were used before in room acoustics, the actual development of SEA started in the early 1960 with the application to vibro acoustic problems in aerospace engineering. ”Statistical” means that the variables are drawn from statistical population and all results are expected values. "Energy" denotes that energy variables are used and, according to Lyon [6], "Analysis" means here that SEA is more a general approach rather than a particular technique. The main idea in SEA is that a structure is partitioned into coupled "subsystems" and the stored and exchanged energies are analysed. The original theory is based on the study of interaction of groups of modes: no energy exchange takes place between different frequency ranges. Energy flows between "subsystems" which interact so that their vibrating energy tends to be the same as the one of the adjacent subsystem. - A comparable phenomenon occurs as for heat transmission where "subsystem" temperatures tend to equal -. Once computed, these energies are used to estimate the acceleration and stress levels on the parts of the structure.

In mechanical structures the energy transmitted between the S/S is strongly dependent on the vibration modes. Energy can be transferred from one subsystem to another in a certain frequency band only if both subsystems do have vibration modes in that frequency range. It is also important that the modes are acting in a common direction. Vibration modes do thus have energy storage capacity. A group of similar vibration modes are defined to be a SEA S/S. A selection of possible SEA S/Ss is shown in FIG 2 for a plate structure.

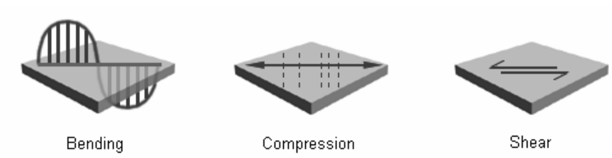


FIG 2. SEA subsystems in a plate

A complex structure can thus be represented by a limited number of S/S that can be treated with low numerical effort. The S/Ss as well as their coupling are defined by a couple of parameters that are shown in FIG 3. Power input P_i, Power output P_e and transferred Power P_{12}' /P_{21}' as well as stored Energy E are displayed.

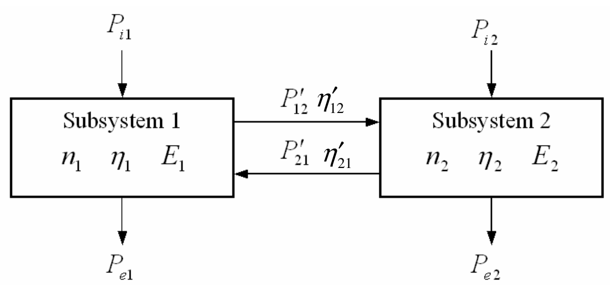


FIG 3. Two element SEA system with parameters

2.2. Parameters of SEA

The internal loss factor (ILF) is representing the structural damping and defines the power dissipation level of a subsystem in a frequency band, being defined as the ratio of energy dissipated per second to the average energy stored in the system:

$$\eta(\omega) = \frac{1}{\omega} \cdot \frac{P_e(\omega)}{E(\omega)}$$

The coupling loss factor is the SEA mechanism that characterizes the modal energy transfer between coupled SEA elements. The SEA assumptions that, between subsystems, response and excitation are proportional, and that the response changes in the same manner as the excitation provide the reciprocal relationship between η_{ij} and η_{ji}, n_i and n_j being the modal density of the coupled subsystems.

$$\eta_{ij} \cdot n_i = \eta_{ji} \cdot n_j$$

Modal density is the average density of modal frequencies in a subsystem and a frequency band:

$$n_i(f_k) = \frac{N_i(f_k)}{\Delta f_k}$$

The coupling loss factor η is depending on the type of the S/S junction. As can be seen in FIG 3 the SEA assumes that the coupling is energy conservative, with means that no energy is dissipated, when being transferred through a junction from one S/S to another.

The basic equations of the SEA are presented shortly in the following:

– energy balance equation for one S/S: $P_{i1} = P_{e1} + P_{12}$
– intrinsic loss: $P_{e1} = \omega \cdot \eta_1 \cdot E_1$
– power transmitted: $P'_{12} = \omega \cdot \eta_{12} \cdot E_1$ $P'_{21} = \omega \cdot \eta_{21} \cdot E_2$
– with $\eta_{12} \cdot n_1 = \eta_{21} \cdot n_2$.
– expressed in matrix form for a two S/S model:

$$\begin{bmatrix} P_{i1} \\ P_{i2} \end{bmatrix} = \omega \cdot \begin{bmatrix} \eta_1 + \eta_{12} & -\eta_{21} \\ -\eta_{12} & \eta_2 + \eta_{21} \end{bmatrix} \cdot \begin{bmatrix} E_1 \\ E_2 \end{bmatrix}.$$

2.3. Modification of the classical SEA

Summing up all the characteristics of the SEA it shows to be a good compromise between efficiency and accuracy and can be applied for space structures high frequency simulation [8].

In this paper three modifications are suggested to further increase the simulation and idealisation capacities of the SEA.

2.3.1. SRS based SEA

As the SEA in its classical formulation is also statistical w.r.t. time its application is limited to harmonic vibration. For the simulation of shock and elastic waves in structures the transient characteristic has to be implemented. It is proposed to use the Shock Response Spectraum (SRS) instead of the Power Spectral Density (PSD) as the frequency dependent simulation variable in SEA. The SRS [5] is the standard tool for the evaluation of the shock generation of pyros and other mechanisms and is also used for the S/S design and qualification test. On the basis of the acceleration time data e.g. from release shock measurements it calculates the maximum vibration response of a S/S mounted on the measurement location depending on the S/S natural frequency. Using the SRS also simplifies the evaluation of the energy input, as the

SRS is available for most shock generating units. As a result the distribution of the SRS within the structure can be used easily for further S/S design and test.

A typical SRS calculation for a measured shock response is shown in FIG 4 and FIG 5, using the following parameters: Quality factor Q=10, octave scale 6.

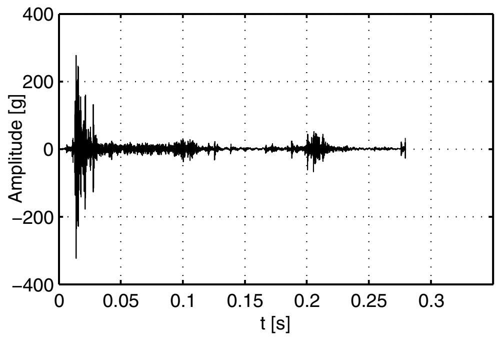


FIG 4. Acceleration time response of a shock source

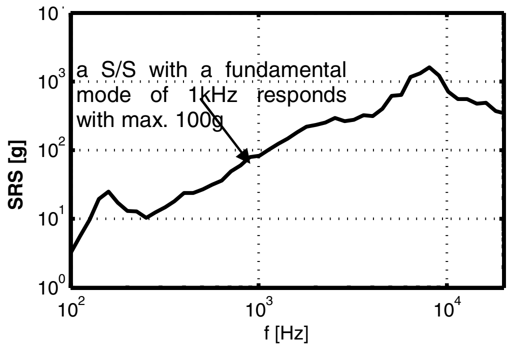


FIG 5. Corresponding Shock Response Spectrum

2.3.2. Mode shape normalisation of CLF

For the calculation of the coupling loss factor only the modal density is used. That implies that the energy transfer of modes is independent of the mode shape. Especially at very local junctions as point junctions the modal elongation of each S/S's mode at the location of the junction has a high influence. The energy transfer is maximal, if both S/Ss do have an antinode (max. elongation) and minimal if both S/Ss do have a vibration node (no elongation) at the coupling point. In between these two extremes a reduced energy transfer is observed. As in many cases the modal parameters of the independent structural subsystems are known from FE models or measurements or can even be calculated easily the additional numerical effort can be accepted.

A normalisation factor can be computed by

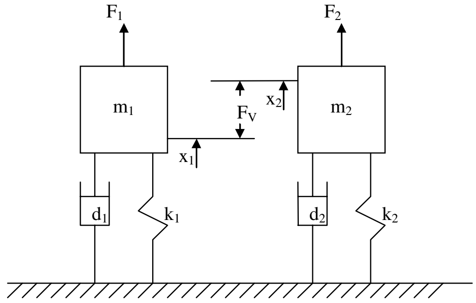
$$\mu_{n,ab} = \Phi_{ik,a} \cdot \Phi_{jk,b}$$

With Φ being the modal matrix, *i* and *j* being the nodes of the connected S/S *a* and *b* and *k* being the mode under investigation. Multiplying the mode k of each subsystem with $\mu_{n,ab}$ when calculating the modal density *n* gives a mode shape normalisation.

$$n_{n,a} = n_a \cdot \mu_{n,ab} \text{ and } n_{n,b} = n_b \cdot \mu_{n,ab}$$

2.3.3. Friction coupling and non-conservative coupling

One of the basic assumption of the classical SEA (CSEA) it is stated, that coupling of subsystems is conservative. In fact, most couplings are non-conservative, as for example in the high frequency range the damping observed in junctions is higher then material damping. A common practical assumption that is used, states that the SRS will be reduced by 40% if a shock load is transferred through a junction between two S/S. This assumption is not frequency dependent and does not respect the kind of the junction. Therefore a non-conservative (friction) coupling is implemented in the SEA. A way of implementing damping loss within the junction is presented in the following for a two mass oscillator.



- in the case of a spring: $F_V = k_V(x_1 - x_2)$
- in the case of viscous damper: $F_V = d_V(\dot{x}_1 - \dot{x}_2)$
- for friction (coulomb): $F_V = F_R \operatorname{sgn}(\dot{x}_1 - \dot{x}_2)$
- ILF for conservative damping: $\eta_i = \frac{P_{id}}{w_i E_i} = \frac{d_i}{m_i \varpi_i} = 2D_i$
- effective ILF (EILF) for a coupled S/S: $\eta'_i = \eta_i + \frac{\eta_V k_V}{\eta_j k_j}$
- for two similar oscillators: $\eta_{12} = \left(\frac{2}{\Delta_1 \varpi_1}\right)\left(\frac{\chi}{2\varpi_1}\right)^2$
- where $\Delta_i = \eta_i \varpi_i$; $\varpi_i = \sqrt{\frac{k_i + k_V}{m_i}}$,

$$\chi = \frac{k_V}{\sqrt{m_1 m_2}} \text{ and } \mu = \frac{d_V}{\sqrt{m_1 m_2}}$$

Exemplary the shock response of SMO1 and SMO2 for a half sine shock on SMO1 is calculated for various values of *X* is presented in FIG 6.

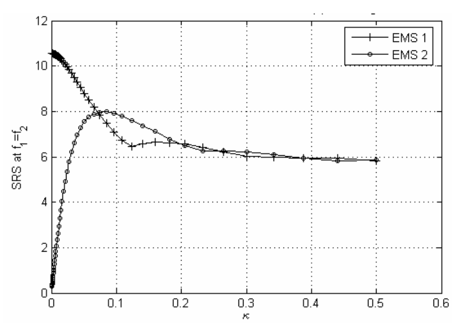


FIG 6. Response on two coupled EMS

2.4. Application of the modified SEA in a complex structure

An exemplary application of the modified SEA (MSEA) is presented in the following. The structure under investigation is a nuclear fusion experiment of the Institute of Plasma Physics in Greifswald, Germany. The experiment is equipped with 50 non planar coils and 20 planar coils, all being superconducting and, important for the application of the modified SEA, being coupled by friction junctions at several points. The shock source is a stick-slip phenomenon that may occur if the coating of the narrow support elements (NSE) wears. The aim of the simulation is to compute the vibration response within the coils and also in the complete coil system. The results can be used to quantify the risk of a Quench (loss of superconductivity) due to stick-slip excitation.

In FIG 7 an overview on the nuclear fusion experiment is given, showing the complex character of the mechanical structure. A numerical treatment of high frequency, transient effects by FEM is inefficient or even impossible.

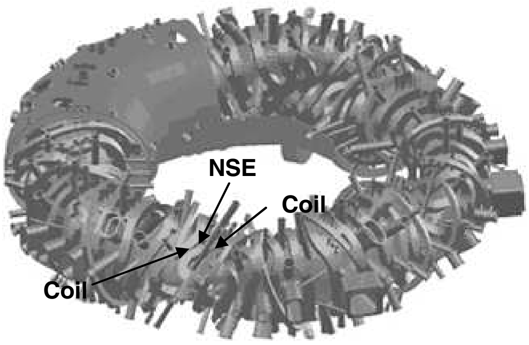


FIG 7. Layout of the Wendelstein 7-X with coils and NSE

In FIG 8 a detailed view on the NSE connecting two coils is given.

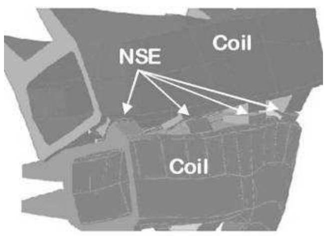


FIG 8. Detailed view on two coils connected with 3 NSEs

To apply the mode shape normalisation of the coupling loss factor as described in 2.3.3 finite element modal results of each coil, already available, have been used and a MSEA model has been set up. For verification reasons and a plausibility check a reduced model, consisting of 10 coils, connected to each other at one coupling point has been used. The system is loaded on two connections with a shock load coming from a stick-slip event. In FIG 9 the transmission of the shock load within one coil and from one coil to another can be seen. As the simulation is performed in the response frequency domain the results can be graphically only presented for one frequency band, here 2kHz.

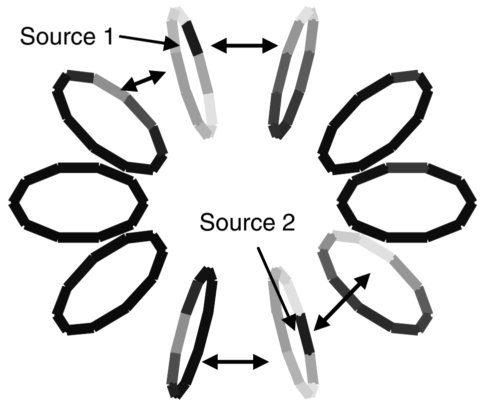


FIG 9. Shock response on coil system by MSEA

Another possibility of result presentation is to plot the SRS at one frequency band along a path of SEA elements. In FIG 10 a path plot along all 100 SEA elements (10 each coil) shows the reduction of the shock load when travelling through the coils and from one coil to another.

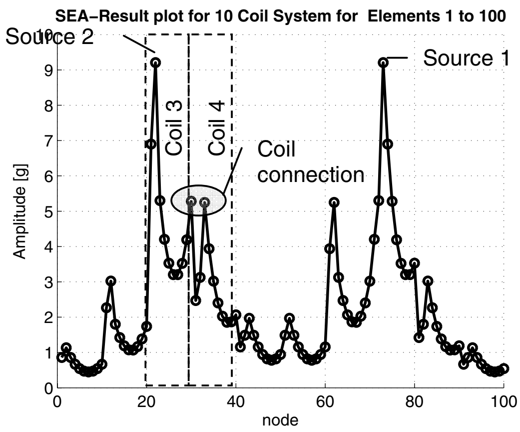


FIG 10. Path plot of SRS at 2000Hz

Summary:

It can be summarized that the modified SEA is a tool, well suited for the simulation of high frequency transient shock loads at complex structures. It has the advantage of a reasonable model condensation by transferring the model in the geometric view into statistical and in the dynamic view into the frequency response space. The setup of a complex model requires data mostly available and the implementation of friction coupling can be done, based on experimental or numerical modal parameters.

3. ISOLATION AND ATTENUATION

There are a number of isolators, passive and active, that have been developed for space structures [11][13]. One goal of them is to isolate the satellite structure from the vibration and shock excitation of the launcher and of separation mechanisms. Attenuation is normally connected to a increase of mass for damping materials or externally mounted vibration absorbers.

In the following isolators and end stops on the basis of visco-elastic and frictional damping as well as structurally integrated absorbers are presented.

3.1.1. Isolation

The isolation against shock waves in the high frequency range differs from the isolation against harmonic excitation. For example a high damping force is not always of interest. Nevertheless frequency and displacement requirements have to be met.

The development of a isolator for a magnetic actuator as it can be implemented in a Low-Shock-Release-Unit (LSRU) is presented. The aim is the isolation of the actuator from external shock that may lead to unintended release. An external shock of 700g half sine with 0,25ms shall be applied to the LSRU without failure. For the rigidly mounted actuator it showed that the retraction force of the magnet is to less to hold the actuator pin in position. Viscoelastic isolators have been developed with the aim of implementing frictional damping to further increase the efficiency at lowest possible mass.

The LSRU as well as the actuator to be isolated are presented in FIG 11.

FIG 11. LSRU (l) and actuator with actuation pin (r)

A couple of topologies, materials and parameter variations have been numerically investigated. In FIG 12 the topologies under investigation are presented.

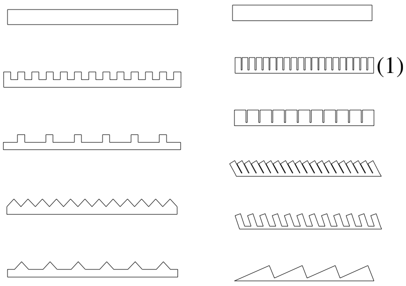


FIG 12. Isolator topologies under investigation

Finite element models have been build up and explicit time integration have been used to evaluate the different isolator concepts. LS-Dyna has been used to calculate the shock response of the actuator from the specified external shock input. The finite element model of the rippled isolator is presented FIG 13.

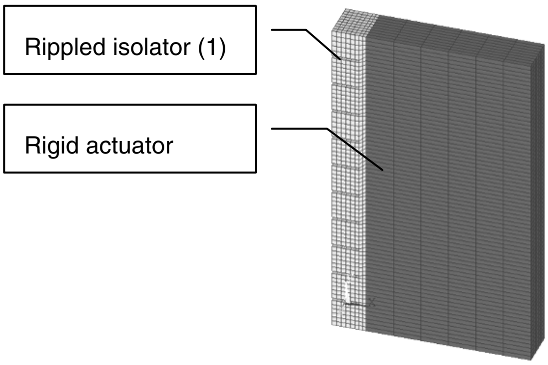


FIG 13. Finite Element Model of rippled isolator

The results for a selection of isolator configurations are presented in FIG 14.

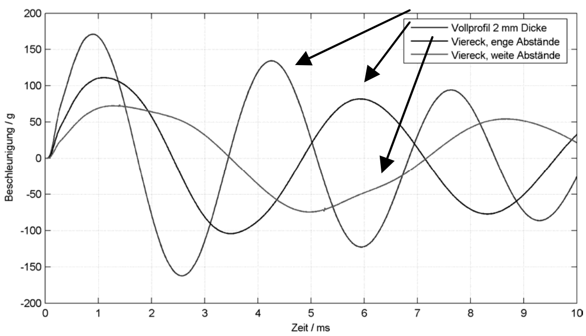


FIG 14. Shock response of various isolator types

The constrains for the isolator optimisation are:

– fundamental frequency > 200Hz
– maximal vibration amplitude < 0.8mm
– shock robustness (no release) > 640g

The visco-elastic material has been implemented by the Moony-Rivlin-Model, friction in between the ripples has been implemented with a friction coefficient of 0.6, see FIG 15.

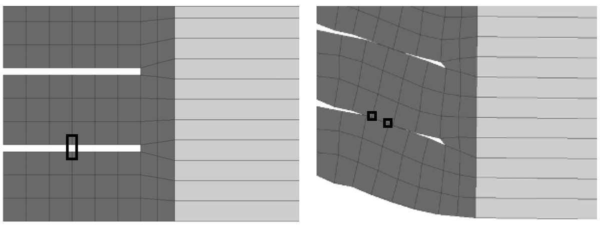


FIG 15. Friction in isolators modelled in FEM

Experimental tests have been used to verify the FE Model. Isolators with the optimized configuration have been built and experimental shock tests, see FIG 16, showed good correlation, taking into account the transient character and the complex material behaviour, with the numerical results with a mean deviation of 10% to 20%, the simulation being to stiff.

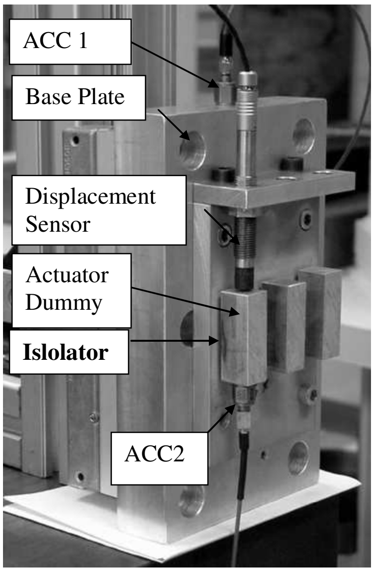


FIG 16. Experimental setup for isolator verification test

Result:

A final evaluation showed that the shock robustness increased from 300g to 1000g with less then 10g of additional mass.

3.1.2. Absorption

Absorbing wave energy in absorption elements and dissipating it is the way of reducing the shock wave amplitude when it travels through the structure. Externally mounted absorbers as a discrete vibration absorber (DVA) have the disadvantage of a high amount of additional mass and the problem of the connection to the structure being stiff also in the high frequency range. Therefore a structure integrated absorber has been developed. For a rod like structure as used for primary structure stiffeners “stuts”, various methods of energy absorption and also wave reflection have been numerically investigated by FEM explicit simulation and evaluated as can be see FIG 17.

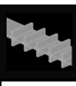
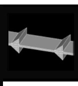
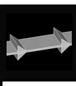
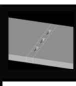
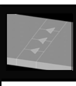
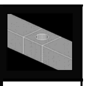
						
Reduced amplitude	36%	10%	10%	5%	5%	5%
Production	+	-	-	--	--	++
Weight	--	-	-	+	+	+
Solidity	++	++	++	--	--	--
Total cost	+	-	-	++	++	++
Variability	+	-	-	-	-	-

FIG 17. Absorber / Reflection elements

The most promising design has been numerically optimised and experimentally tested. It showed that a beam equipped with small beams, working as vibration absorbers in their natural frequency can be used to manipulate the shock wave energy transferred from one end to the other in a very defined way. A numerical model of the absorber integrated beam is shown in FIG 18.

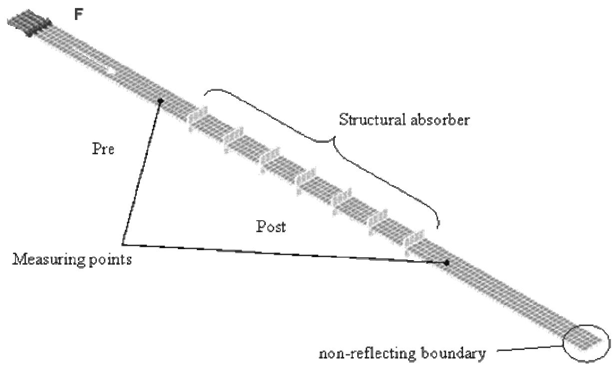


FIG 18. FE-Model of the integrated absorber

The absorbers are designed such, that their fundamental frequency matches the frequency range for which the SRS shall be reduced, see FIG 19.

FIG 19. Mode shape of the beam at the absorber frequency

A optimisation with the aim of maximal reduction, the constraint of <10% additional mass and the design parameters being number, thickness and length of the absorber beams has been performed. The optimal configuration has been manufactured and experimentally tested as can be see in FIG 20, which is comparable with the numerical simulation shown in FIG 18.

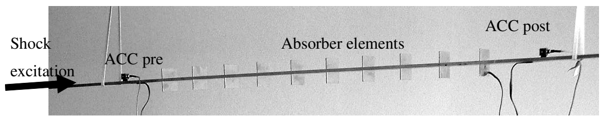


FIG 20. Test setup for absorber test

The results have been very promising, showing a reduction of 36% in the time domain, see FIG 21.

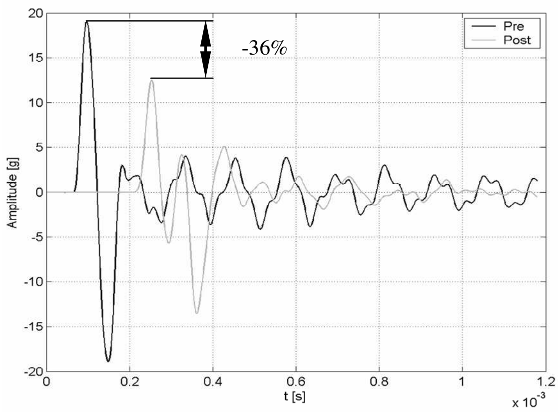


FIG 21. Results of absorber test in time domain

From the time data a calculation of the response data of the sensor before (pre) and after (post) the absorber has been performed, see FIG 22

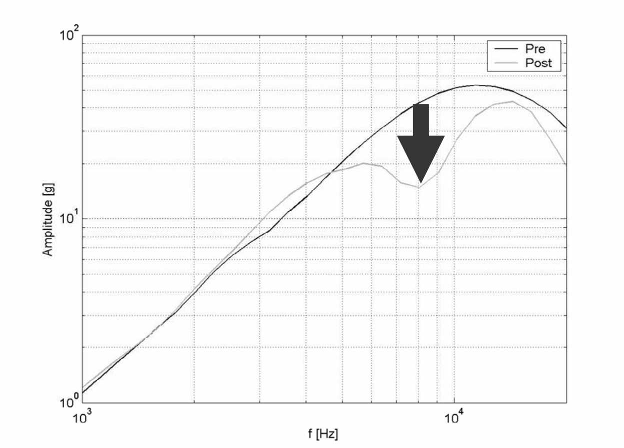


FIG 22. Results of absorber test in frequency domain

Result:

The results have been even more promising, showing a reduction of the SRS in the frequency range from 5kHz to 10kHz by a factor of up to 3.

4. MONITORING

Condition monitoring is of special interest at components that can not be inspected visually during operation or maintenance service. The reason may be limited or impossible access or the need to quantify the wear parameter by means of a sensor system. Monitoring gives information on the loading and wear situation of the structural component being inspected. This information is used for redesign, efficient use of the durability or for the adaptation of the load to the wear status.
Besides of static or low frequency physical properties as strain, displacement, acceleration and temperature also high frequency acoustic waves can be used for structural health monitoring.

As already mentioned, elastic waves are travelling through the structure with sound speed, so that the measured time data can be correlated with the location of the sound source. This enables not only the detection of events, but also the location of them. Many mechanical events are triggering a sound wave, so that acoustic monitoring is an adequate observer system.

In the following a monitoring concept is presented that has been developed for the inspection of friction pads in a friction test stand. The friction pads, also called narrow support elements (NSE) are being installed in the nuclear fusion experiment *Wendelstein 7-X* by the *Institute of Plasmaphysics* in Greifswald, Germany. The installed sensor system is used for an early indication of wear and for an assignment of the wear to the two test samples installed. The monitoring system has been improved by the evaluation of various sensor types and signal analysis procedures, so that the system can be installed in the complex structure in which the friction pads shall be installed.

The basis of the presented monitoring system is the use of a distributed sensor system and the difference in the delays of the elastic wave from the source to the sensors. At least two sensors are used to locate a sound source for one dimensional wave propagation. The knowledge of the sound speed and the distance of possible sound sources enable a comparison of experimental data with analytical results.

This relationship is shown using experimental data acquired in a friction test stand, after a short description of the test aim and test setup.

The event to be monitored is stick-slip. Stick-slip is caused by the surfaces harmonically alternating between sticking to each other and sliding over each other, with a corresponding change in the force of friction. This spontaneous change of the friction force triggers a sound wave that is used to detect the stick-slip phenomena, even in a very early phase before an increase of the coefficient of friction (COF) can not be detected by the load cell installed. Stick-slip is an indication of wear of the coating, used at the friction pads that are tested.

The friction test stand has been developed by KRP-Mechatec Engineering GbR, Garching and has the following properties:

– Environment: RT or cryo-vacuum (85K, 10e-6mbar)
– Normal load: up to 1.75MN (175to)
– Friction load: up to 2.0MN (200t) dep. on COF
– Sensors: mechanical, thermal and acoustical

The test stand has the main components:
- Tensile testing machine (2MN)
- normal pressure loading frame
- LN2-cryostat and internal cooling
- Vacuum chamber
- Symmetric fixation for two samples

An overview of the test stand is given in FIG 23, FIG 24.

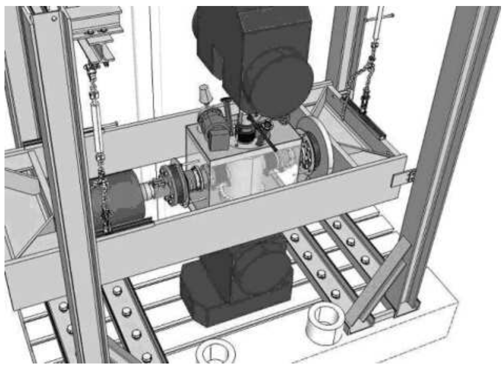


FIG 23. CAD of the cryo-vacuum friction test stand

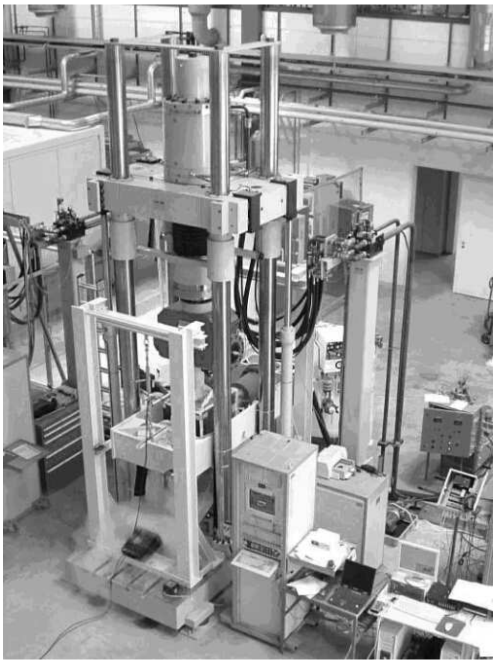


FIG 24. Test stand installed in a 2MN tensile testing machine

A friction sample after the lifetime of 4000 sliding cycles showing wear on the molybdenum disulfide coating is shown in FIG 25.

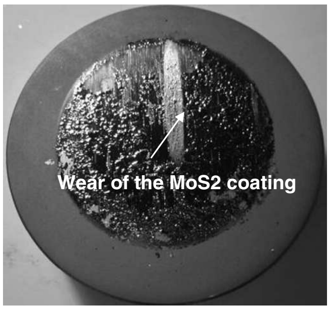


FIG 25. Wear pattern of the coated friction sample

A typical behaviour of friction pads in life time test is shown in FIG 26, giving also the start of stick-slip and the early detection by acoustic monitoring.

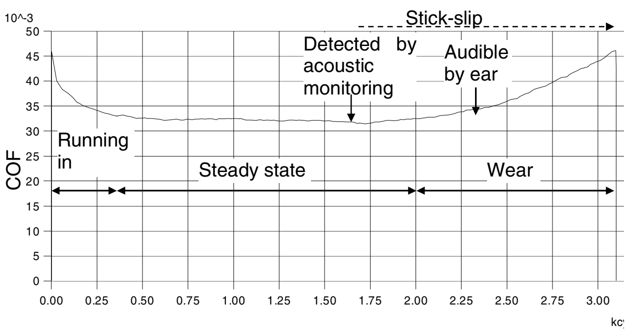


FIG 26. COF during life time test

The path of the sound wave from the source, which is the stick-slip at the friction pads sliding surface, to the acoustic sensors installed is given in FIG 27. Acoustic sensors are installed close to each of the two friction pads.

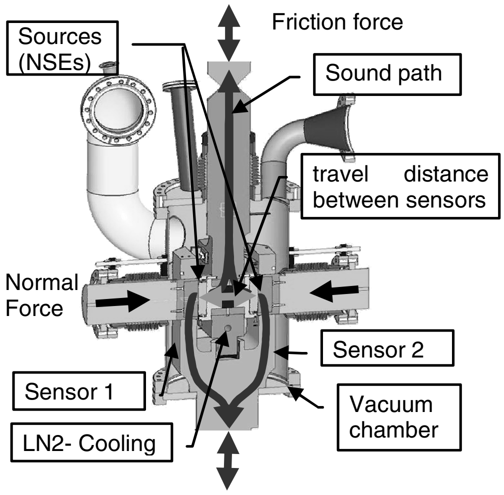


FIG 27. Path of the shock wave through the structure

The comparison of experimental and numerical time delay as mentioned above is presented below.

A measurement of the difference in the sound path for the two sensors gives 170mm. The sound speed can be calculated by assuming longitudinal propagation with:

$$c_{structure,longitudinal} = \sqrt{\frac{E(1-\mu)}{\rho(1-\mu-2\mu^2)}}$$

Using the following values,

- E (at 77K) = 210MPa
- ρ= 7980kg/m³
- μ=0,3

Gives a sound speed of c=5950m/s, what gives a time delay of

$$\Delta t = \frac{\Delta l}{c} = \frac{0.170m}{5950\frac{m}{s}} = 29\mu s$$

The acoustic signals monitored with 400kHz during stick slip are shown in FIG 28.

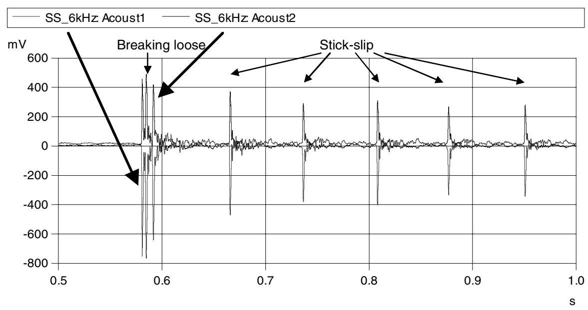


FIG 28. Acoustic signals during stick-slip

Zooming in a stick slip event enables a extraction of experimental time delay between acoustic sensor 1 and acoustic sensor 2 as shown in FIG 29. It shows a very good agreement of experimental (30μs) and analytical values (29μs).

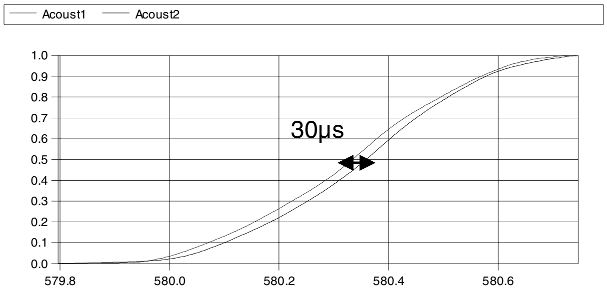


FIG 29. Experimental data for time delay

This verification of the sensor system and the time delay being detectable with good accuracy is the basis of the further work, comparing various sensor types and setting up efficient signal processing tools.

The sensors installed and evaluated w.r.t. sensitivity, dynamic and applicability are:

– hole mounted body sound sensor (ps/ks/11, marco GmbH)
– glued piezo plates (20mm x 20mm x 0.5mm, Sonox ® P53, Ceram Tec AG)
– strain gauges (CFLA-1-350-11, Tokyo Sokki Kenkyujo CO.,LTD)
– Accelerometer (500g, IMC Additive GmbH)

In TAB 1 a comparison with respect to noise level and sensitivity w.r.t. stick-slip events of the three sensor types for the application of acoustic monitoring is presented.

TAB 1. Comparison of sensor types for acoustic monitoring

	Noise		Slight stick-slip		Strong stick-slip	
	RMS	Max.	RMS	Max.	RMS	Max.
Body Sound [µV]	3,3	10	15	90	37	205
Piezo	6,3	25	38	150	64	280
Strain gauge [mProm]	0,36	0,71	2,6	3,5	3,1	4,1
Accelero-meter [g]	0,08	0,38	0,47	5,7	6,2	64

Besides of the quantitative comparison of measurement data a qualitative comparison of applicability and signal processing shall be given below:

– Body sound sensor: easy installation, no signal processing necessary, robust
– Piezo plate: installation needs experience also for cryo-vacuum environment, no signal processing necessary, needs no machining on the structure
– Stain gauge: installation as piezo plate, high signal processing effort to extract stick-slip phenomena from low frequency strain and noise
– Accelerometer, easy installation, no signal processing necessary

<u>**Summary:**</u>

– **Body sound sensors and piezo plates are very well suited for acoustic monitoring.**
– **For strain gauges a more enhanced signal processing is necessary.**
– **A clear understanding of the sound propagation is necessary for a location of the sound path**

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