Damping – Oscillation Elimination after Rupture

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1 Introduction

The subject of this article is using of the Damping for Oscillation elimination after the Rupture. The problem appears in explicit simulations of the plastic parts with computational models of material damage. After the rupture or erosion of finite elements, the model of the structure oscillates. The oscillation of the structure may be significant and cause additional erosion of the structure, which is non-physical. All real structures have any damping and this damping should be considered in numerical simulation. We have a lot of possibilities of the damping options, but we need to know the correct value of the damping. So, in this contribution, very simple method for damping factor estimation for plastics is described, concretely TSCP (Typical Semi-Crystal Polymer). This method is a combination of numerical simulations and experimental estimation of the damping factor for specific eigen-frequency. Repeating this method with different boundary conditions, we can cover some frequency spectrum. The contribution shows a lot of the damping options in LS-DYNA with interesting consequences. Using different damping schemes requests dissimilar changes of the stiffness and model properties. The purpose of our work is the preservation of the same physical, material and stiffness properties of the model and the real physical structure. The physical structure of FSM (Fuel Supply Module) is used for numerical validation of damping options, used in computational simulation.

2 Damping coefficient – experimental determination

The main motivation is non-realistic oscillation of the stress waves (for TSCP material) after the cracking, if we use material model with damage. This problem can be present in general domain. The basic experiment was made for simple test specimen of the TSCP material. The specimen was fixed in fixation device and then pre-loaded. The pre-tension was l=10mm from vertical axis, but it was possible to setup different pre-tension displacement too. The accelerometer was glued on the specimen with prescribed position and acceleration signal was used for evaluation of the dynamic response. The second step was releasing of the pre-tension deformation and record of the damped specimen oscillation. The important result of measurement was acceleration signal from accelerometer. We parallelly made numerical simulations in LS-DYNA software.

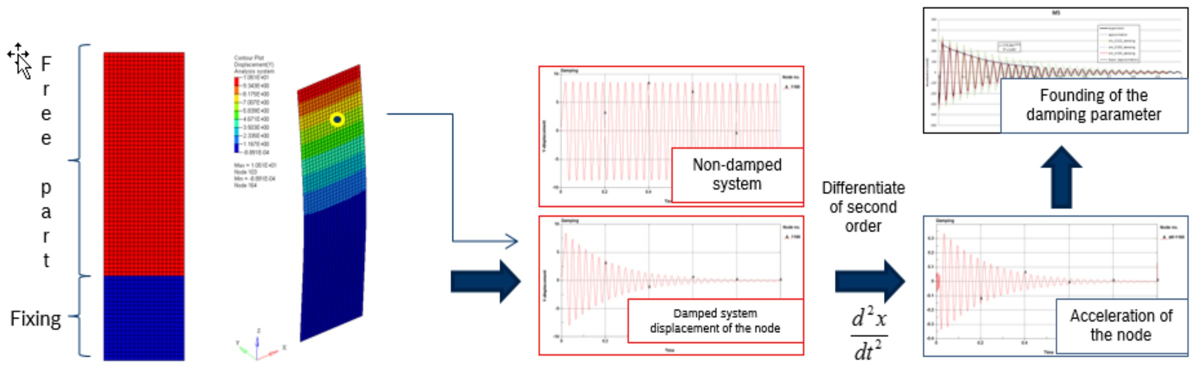


Fig.1: Analytical reverse analysis and figure of the damping coefficient verification in LS-DYNA

The estimation of the damping ratio (coefficient) was calculated based on the analytical damping theory. We have to show necessary fundamental rules and equations, which were used for better understanding. The damping ratio ζ characterizes frequency response of the Second Order Ordinary differential equation and relates with logarithmic decrement δ. The logarithmic decrement δ establishes state of the damping system (un-damped, under-damped, over-damped and critically-damped). The logarithmic decrement δ is used to find the damping ratio of underdamped system in the time domain.

The physical definition is that logarithmic decrement δ means natural logarithm of the amplitudes of any two successive peaks, see equation (1).

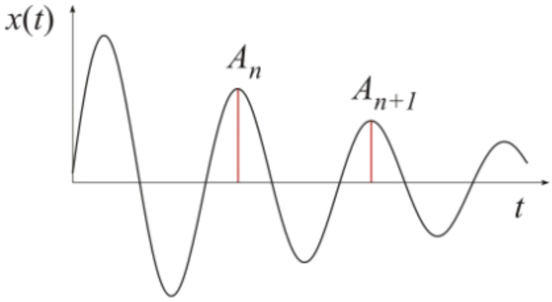


Fig.2: Scheme of the of the amplitudes for underdamped system - oscillator

$$\delta = \frac{1}{n}\ln\lambda = \frac{1}{n}\ln\frac{x_0}{x_n} \qquad (1)$$

$$\zeta = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}} \qquad (2)$$

, where X_0 is the greater of the two amplitudes and X_n is the amplitude of last range peak. The damping ratio we can estimate from definition logarithmic decrement, see equation (2). The one way of the experimental damping ratio determination is using of logarithm decrement. We can show the process of the determination final equation (2). We have equation (3) for basic definition of the logarithmic damping decrement. X_1 is first higher amplitude and X_2 is following amplitude.

$$\delta = \ln\frac{X_1}{X_2} \qquad (3)$$

The definition of the underdamped system $\zeta < 1$ is possible define following equations:

$$X(t) = Xe^{-\omega_n\zeta t}\ \sin(\omega_d t + \phi),\ t_2 = t_1 + T_d,\ T_d = \frac{2\pi}{\omega_d},\ \omega_d = \omega_n\sqrt{1-\zeta^2} \qquad (4.1 - 4.4)$$

, we can write following equation for each amplitudes:

$$X_1 = Xe^{-\omega_n\zeta t_1}\sin(\omega_d t_1 + \phi),\ X_2 = Xe^{-\omega_n\zeta t_2}\sin(\omega_d t_1 + T_d + \phi) \qquad (5.1 - 5.2)$$

, substituting equations (5.1, 5.2 and 4.3) to (3) and mathematical simplifications give equation (6):

$$\delta = \ln\frac{1}{e^{-\omega_n\zeta T_d}} = \ln(e^{\omega_n\zeta T_d}) = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \Rightarrow \zeta = \frac{\delta}{\sqrt{4\pi^2+\delta^2}} \quad \text{...damping ratio} \qquad (6)$$

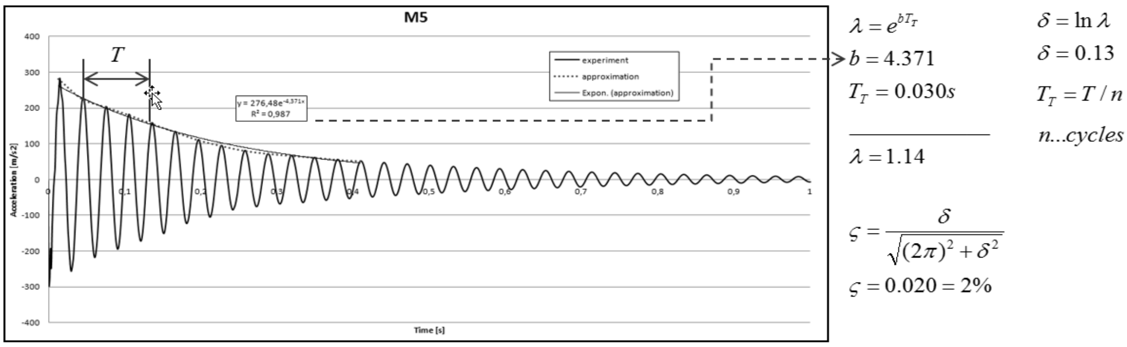


Fig.3: The determination process of the damping ratio from logarithmic decrement

3 Sensitive analysis of the damping setup in LS-DYNA

Now, we have correct value of the damping coefficient, but a lot of damping options are possible to be chosen in LS-DYNA. The differences between each damping option (damping keyword) are very significant. We tested a lot of damping keywords with different setups. The selection of damping keyword crucially depends on the know-how about solved problem or structure, which is subject of calculation. If we know some own –frequency characteristics of the structure then we can use particular specifics of the damping. Usually, we know some frequency range, where we can wait oscillations. So, we describe each solved damping keywords based on this variety.

3.1 *DAMPING_FREQUENCY_RANGE (*DFR)

This option suggests constant damping ratio in request frequency range. It is very common and universal damping card in LS-DYNA. The using of this card is directly connected with material stiffness change. If you use this option, then you have to change elastic stiffness of the material, due to increasing/decreasing of the material stiffness by damping in moving equation.

```
$
*DAMPING_FREQUENCY_RANGE
$---+----1----+----2----+----3----+----4----+----5----+----6----+----7----+----8
      0.025           1        3000           1
```

CDAMP…	*Damping in fraction of critical*
FLOW…	*Lowest frequency in range of interest*
FHIGH…	*Highest frequency in range of interest*
PSID…	*Part set ID*
PIDREL…	*Optional part ID of rigid body, motion is damped to relative to rigid body motion*

*Fig.4: Character of the keyword *DAMPING_FREQUENCY_RANGE*

The sensitivity effect of the damping to material stiffness is dependent on the value of damping ratio and frequency range, which is interest. The recommending about option of back change material elastic stiffness is possible find in LS-DYNA manuals, see below *Fig.5*.

Damping Ratio	% error for F_{high}/F_{low} =		
	3 to 30	30 to 300	300 to 3000
0.01	3%	4.5%	6%
0.02	6%	9%	12%
0.04	12%	18%	24%

Fig.5: Average error across the frequency range (change of the own frequency)

We can see comparison of experiment and different setups of this damping card. A lot of variants were tested, but for better explanation we show only some interesting variants. It is possible to see, that the best agreement with experiment is achieved when recommended value of the change stiffness and estimated damping ratio are used. If we setup lower and higher elasticity of our computational model then we obtain different amplitude period from experiment, see *Fig.6*. The elasticity change option is very sensitive on percentage increasing/decreasing. The general recommendation is using of the very narrow range of the damped frequency. Then the change of the elasticity will be very small and non-significant. The damping ratio value is given by structural and material real damping, so it is depending on the solved problem.

Next step was behavior of this card for different frequency ranges (different width too) and damping ratio which is used for real frequency of our test sample $f_0=33$ *Hz*. The difference of upper/lower frequency from range *(f)* and *(f_0)* oscillation frequency is showed in the X axis of Fig.6. You can see, that choice of the correct frequency range for our simulation is very important. If we use incorrect frequency range then we have different damping ratio for oscillation frequency which is in interest from our overview. The detailed behavior of these different damping options you can see in *Fig.7*.

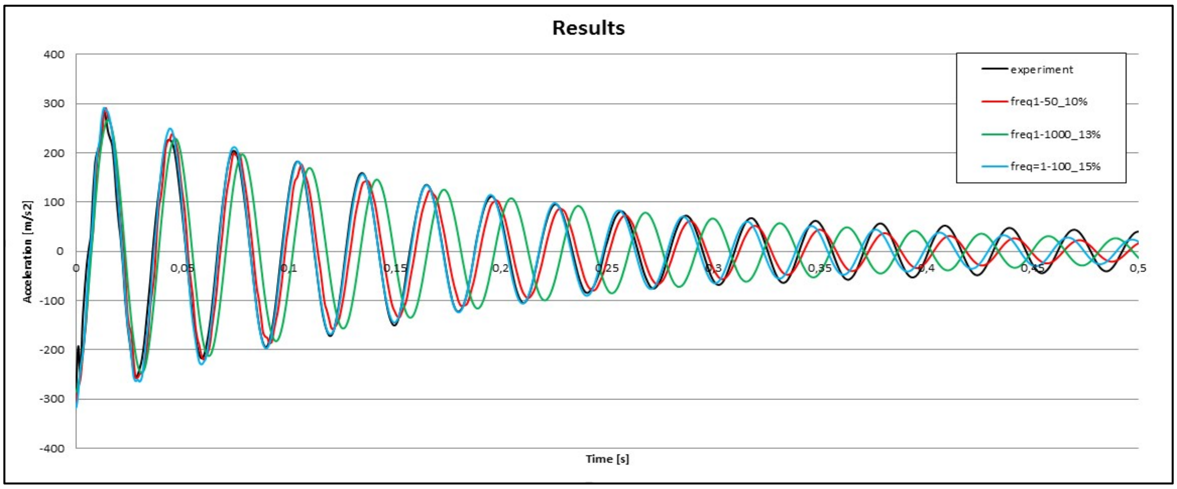
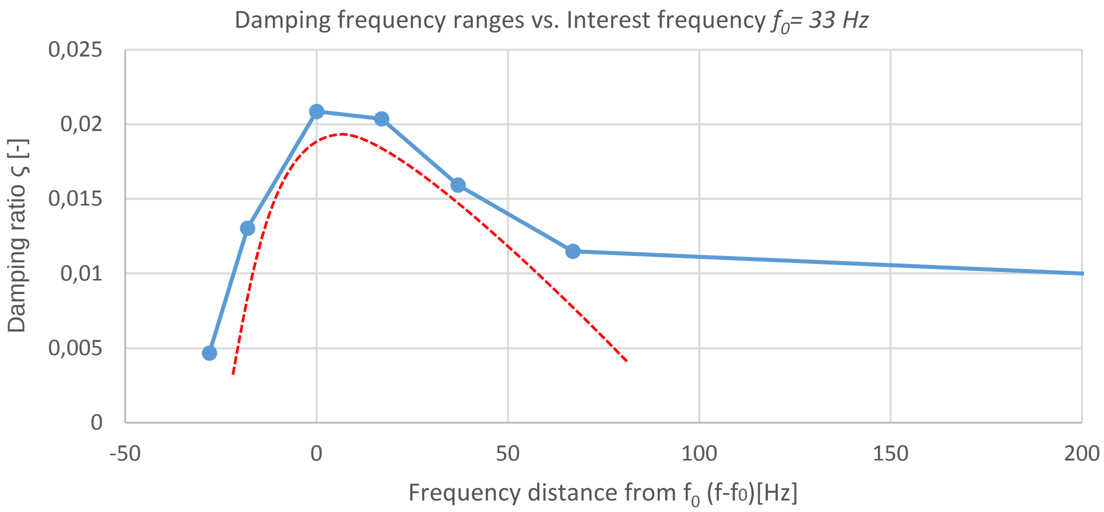


*Fig.6: Different setup of the *DAMPING_FREQUENCY_RANGE (*DFR) and percentage elasticity change*

Following study of the frequency range and reality damping ratio for interest oscillation is focused on different frequency ranges with different frequency width. The damping ratio ζ=0.02 was used for all study variants in this chapter. It is possible to find very interesting results and dependency, if we use very different frequency range of our interest then our oscillation will be damped with very small damping, but it is not non-damped problem. More study variants should be carried out to better understand this dependency. Generally, we can see there is approx. parabolic shape dependency with global maximum of function *f(x).* The example of this dependency for our case, see below *Fig. 7*.



Freq. range [Hz]	*f-f0 [Hz]*	*b*	*TT [s]*	*λ*	*δ*	*ξ*	*ξ [%]*
1-5	-28	1,245	0,0236	1,02982	0,02938	0,004676	0,47
1-15	-18	3,559	0,0230	1,08530	0,08186	0,013027	1,30
33	0	4,371	0,0300	1,14012	0,13113	0,020865	2,09
50-100	17	4,924	0,0260	1,13658	0,12802	0,020371	2,04
70-120	37	4,038	0,0248	1,10533	0,10014	0,015936	1,59
100-200	67	2,889	0,0250	1,07490	0,07222	0,011494	1,15
1000-2000	967	0,354	0,0250	1,00889	0,00885	0,001409	0,14

*Fig.7: Detailed results of the frequency range study *DAMPING_FREQUENCY_RANGE*

3.2 *DAMPING_GLOBAL

This damping option defines mass weighted nodal damping which is applied to the nodes of the deformable body or to the center of mass of the rigid body. This damping is directly applied in motion equation as additional node forces. The recommend damping constant for *DAMPING_GLOBAL card is calculated such as:

$$D_S = 2\omega_{\min}\zeta$$
$$\omega_{\min} = 2\pi f_{\min} \qquad (7.1 – 7.2)$$

, where D_S is damping constant (input in keyword card), ζ request damping ratio and $\omega_{\min}$ is angle frequency (minimal of the frequency of interest). We calculated this option for following configuration with *f=33Hz and* damping ratio ζ =0.025. We don't have to change the elasticity of the solved model such as in previous case. You can find comparison between different option D_S the correct value is D_S =10.03, see *Fig.8*.

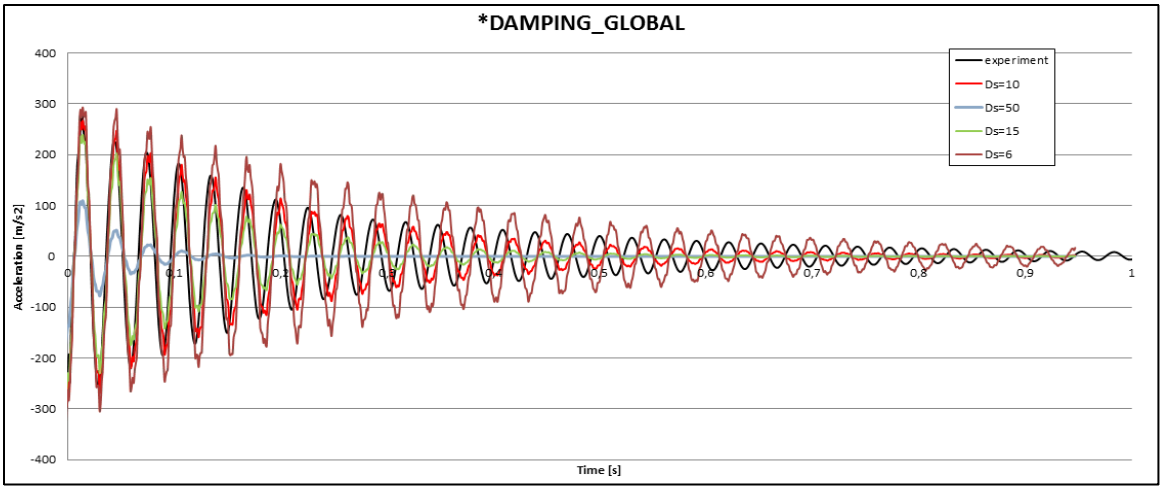


*Fig.8: The comparison analysis of the *DG for different options*

Sometimes, it is possible to state that we use double definition of the damping in practice task. We have to be careful in this case, due to non-reality over-damped simulation task. If we use combination of the *DFR and *DG with the same request damping ratio and for the same interest oscillation frequency then we obtained totally different frequency response than is requested. It is necessary to use only one of damping options.

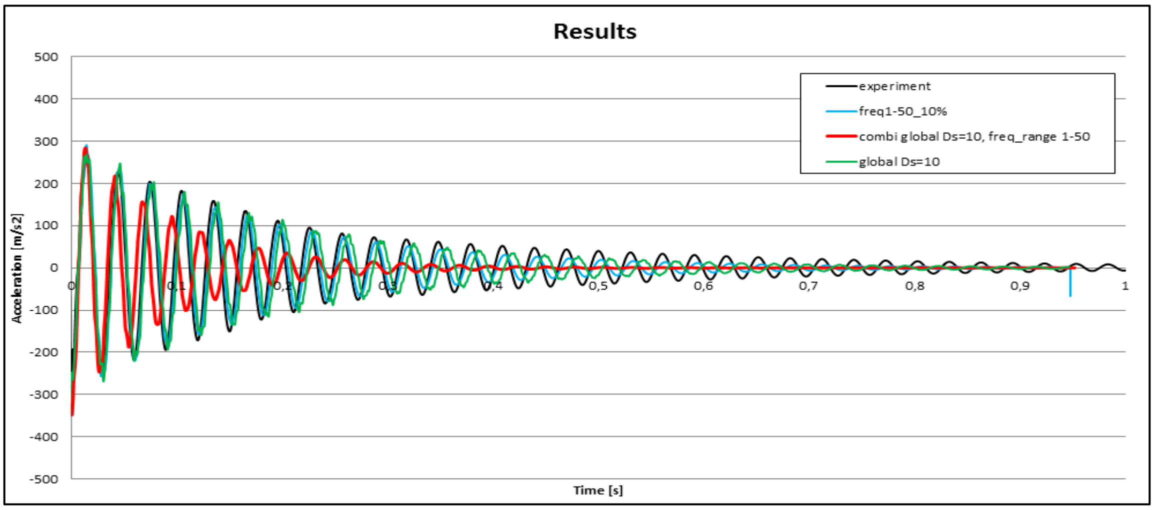


*Fig.9: The comparison analysis of the *DFR, *DG and combination of both setups together*

4 Safety criterion of Fuel Supply Module

Our product must survive a lot of different conditions (loads). It must remain safe in conditions even if a car is totally wrecked in the crash. Fuel supply module can be damaged, but fuel must stay in the tank (tight flange). Flange is designed to be broken in predefined areas where no leakage is possible. Correct crack zone is a main factor of the passive safety of FSM. This mechanism is simulated and tested. Crash process is a very fast process. Damping is one of many parameters which can influence final result.

5 Bosch Impact test

A Bosch Impact Test is used for an initial design evaluation from a breakaway point of view. This is a quick, cheap, destructive test which always shows up a crack zone on the flange when guiding rods are bent. The Impact test is a simplification of a real crash situation in the vehicle when the tank is loaded by the acceleration and the pot assembly bends the guiding rods in the flange.

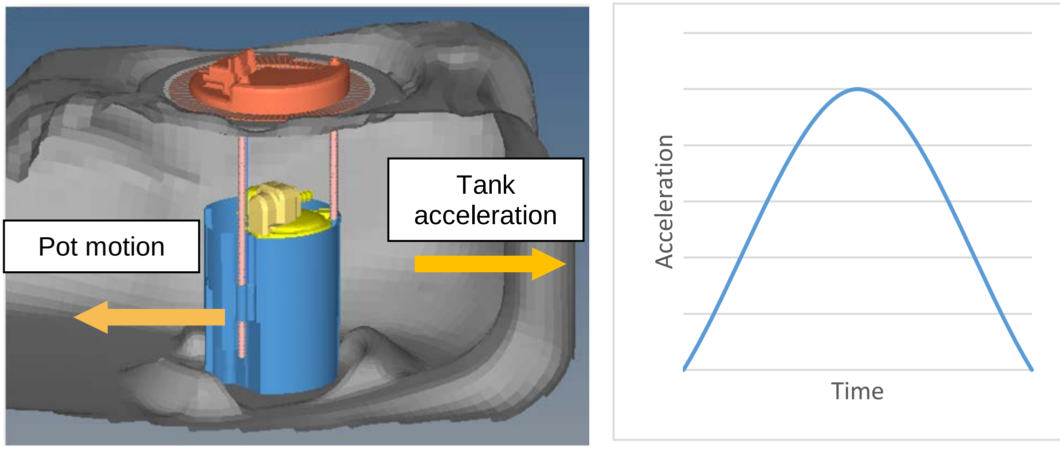


Fig.10: Crash test

The crash test simulation consists of more parts, what extends a calculation time. The correct tank geometry and acceleration profile is not often known in the initial state of car development. The crash acceleration doesn't have to cause flange crack. The crash test is thus firstly substituted by the Bosch impact test. The crack is always reached and the flange tightness can be evaluated.

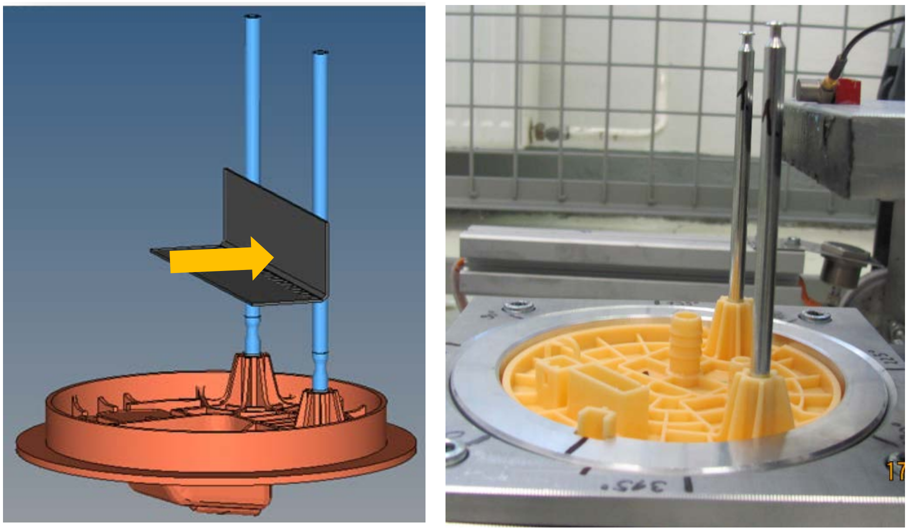


Fig.11: Bosch Impact Test

The Bosch impact test consists of the flange, guiding rods and steel hammer which hits guiding rods by a defined velocity and causes the rod bending till damage occurs. This situation is easy to calculate and easy to test.

6 Damping in real conditions

The damping is not needed in the most of simulations with material damage focused on fuel supply module. The oscillation of resting mass after the crack is not so critical to cause higher stress. The damage accumulation caused by the oscillation is irrelevant in our standard design concepts. But a new rod housing design shows up the opposite way. The correct model evaluation is not possible without the damping definition.

6.1 Original design of plastic guiding rods

The Bosch Impact Test was used for the first design verification, see *Fig.12* and *Fig.13*. The design proposed by the simulation passed the real tests without problem. Simple material card **`*MAT_ADD_EROSION`** was used for the crack evocation. The maximum principal stress and maximum principal strain criterion was set up to reach an element erosion.

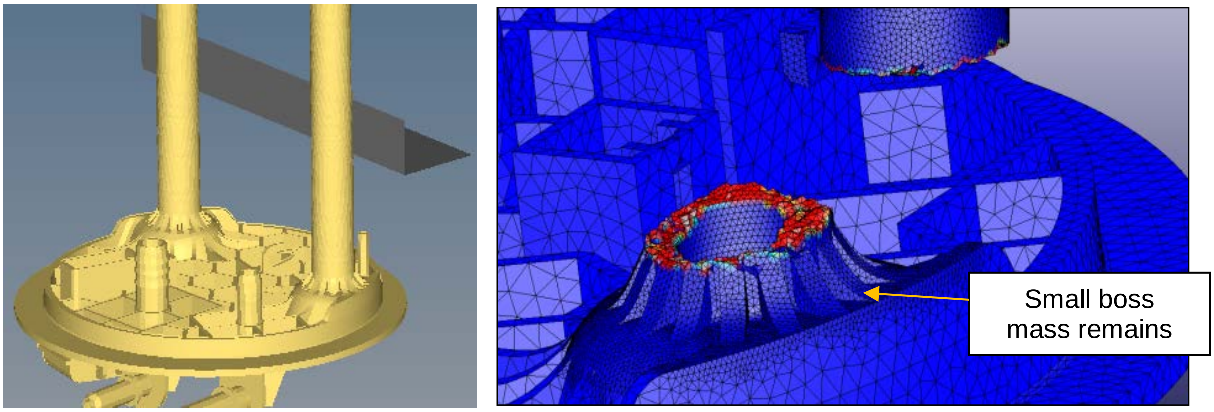


Fig.12: Simulation of Bosch Impact Test (original design)

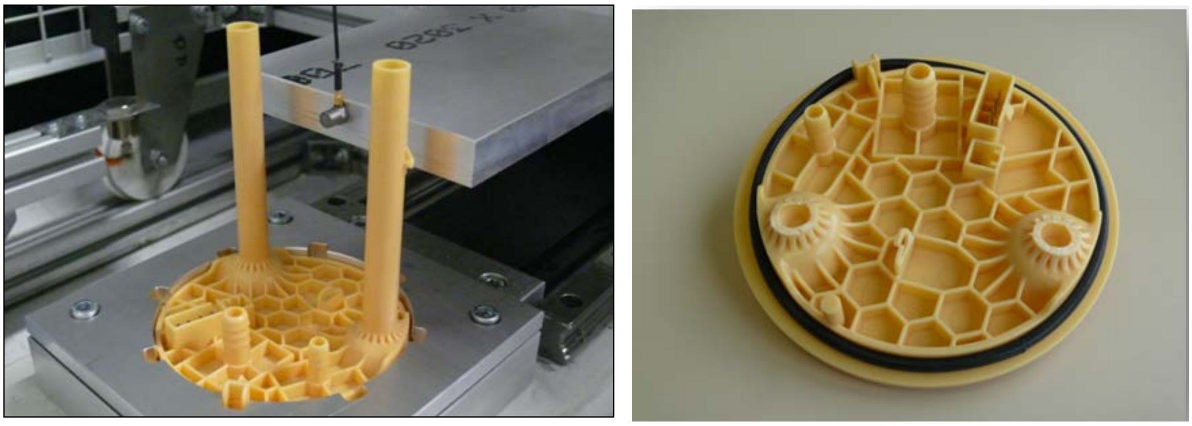


Fig.13: Bosch Impact Test (original design)

The plastic guiding rod is broken and only small boss remains on the flange. The oscillation of this boss causes very small stress which cannot cause additional problem. There is no need for the damping application.

6.2 New design of plastic guiding rods

The longer plastic guiding rods were demanded by the customer. The longer tubes cannot be moulded with needed quality. A higher rod pedestal is necessary. Several design variants were simulated to pass the mould flow process, vibration test, shock test and primarily Bosch impact test.

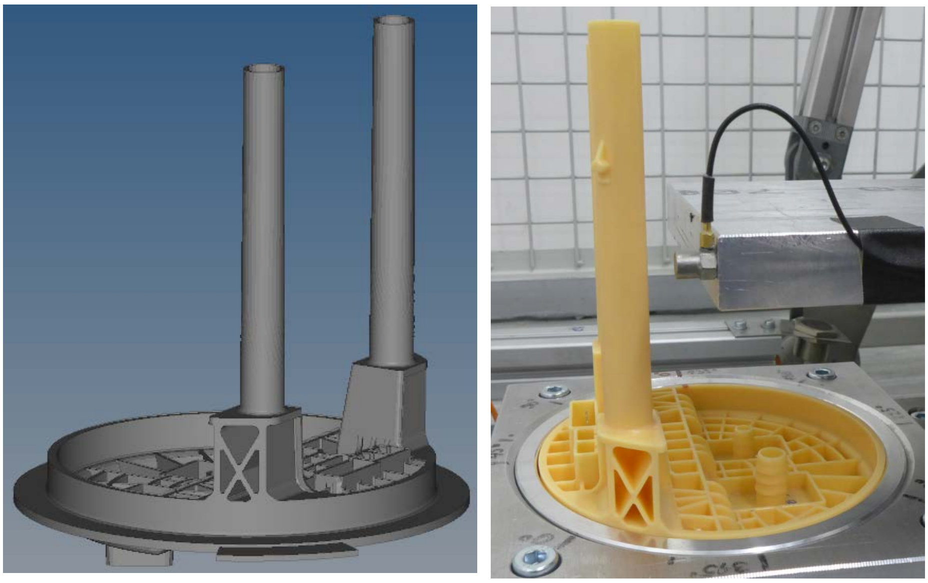


Fig.14: Prolonged guiding rods

The simulation of the impact test was standardly performed absolutely without the damping. The result shown that the additional pedestal creates mass with a high oscillation amplitude. This newly excited vibrations causes a stress peaks close to the ultimate stress *Fig. 16*. This stresses can lead to the damage accumulation and finally to the consequent, unwanted crack.

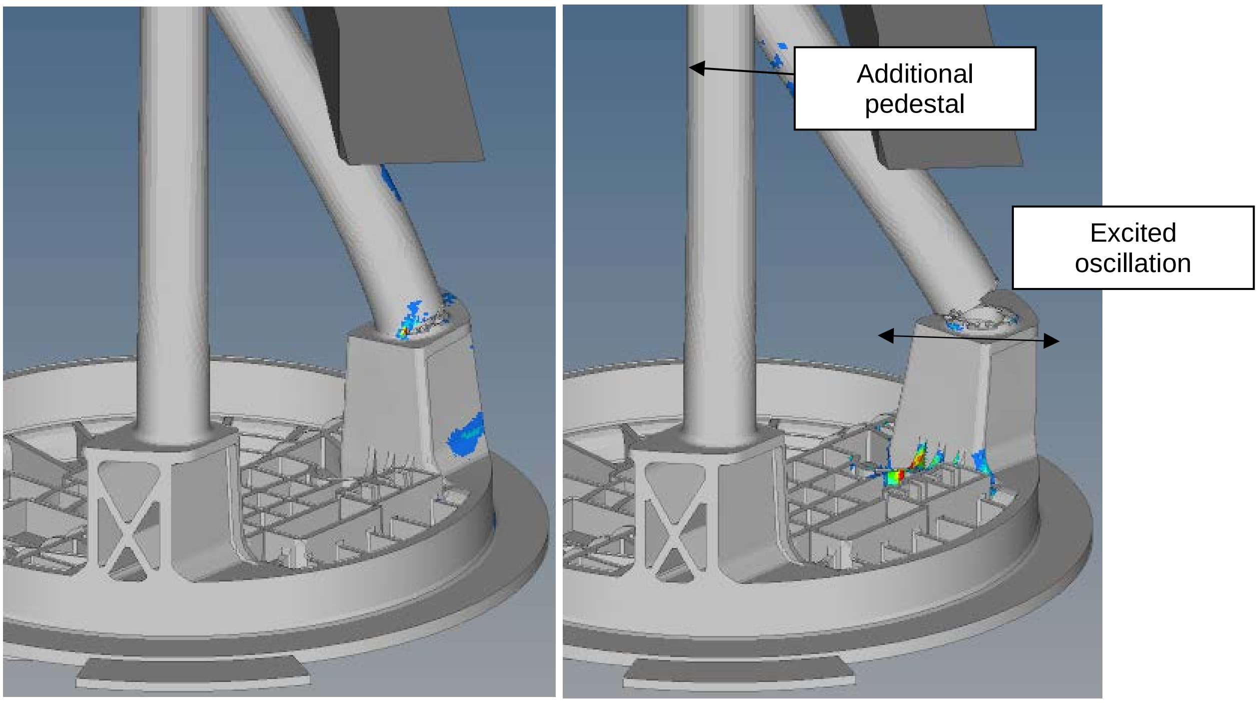


Fig. 15: Impact test simulation of prolonged guiding rods

Fig.15 shows high stress on the pedestal ribs. This non-damped behavior cannot be evaluated. The damping is necessary to obtain the correct stress behavior after the primary crack. The sophisticated damage model with the damage accumulation function is also demanded.

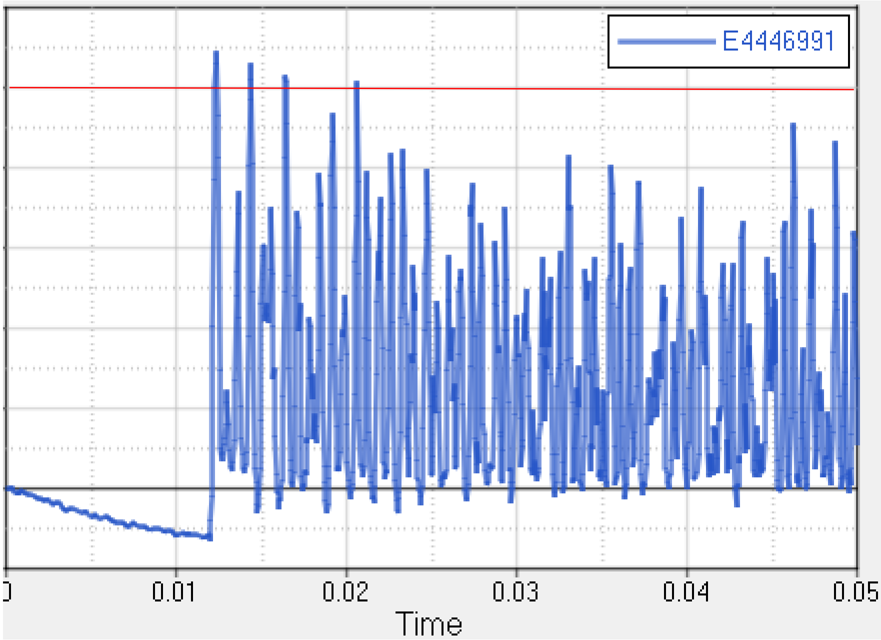


Fig. 16 Stress oscilation without damping

6.3 New design with damping

The same FEM model with totally identical setup was calculated with the *DAMPING FREQUENCY RANGE, which was the most optimal setup from the previous research on plastic sheet.

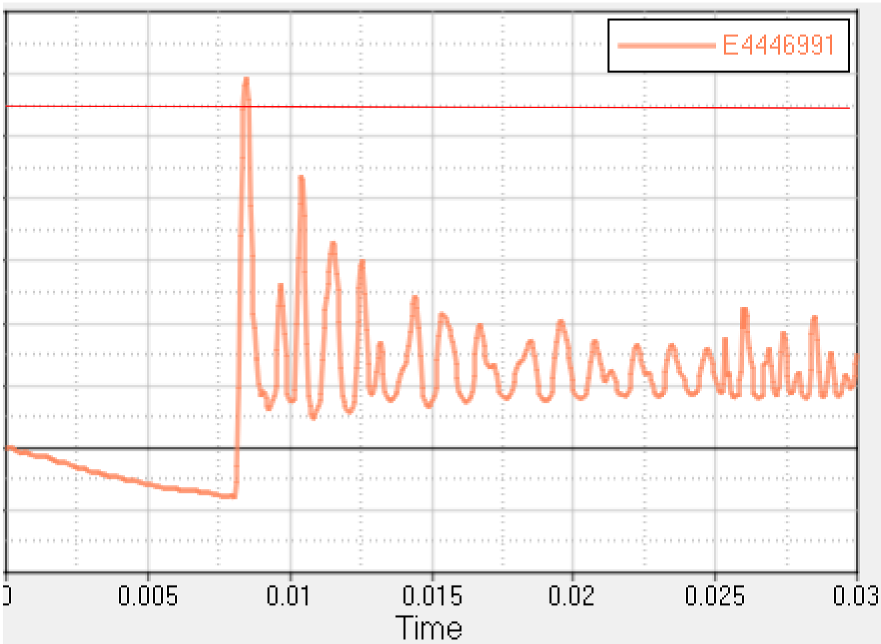


Fig. 17: Stress oscilation without damping

Stress amplitude has decreasing tendency. Only one peak is close to the ultimate stress. Design can be evaluated with usage of maximum principal stress and maximum principle strain criterion. Damage accumulation is very low. This behavior is comparable with real experiment result.

7 Summary

The work deals with the damping application for oscillation elimination after the crack. The basic experiments were necessary made for estimation of the damping ratio. The logarithm decrement approach was used to this evaluation. In next step we focused on the sensitivity analysis of the damping option and two keywords were solved *DFR and *DG. The different variants of the damping option were analysed and typical dependency were estimated based on this. The correct value of the damping ratio and other options were applied to the engineering practice problem. The main problem was additional non-physical move oscillation after cracking (FE erosion) in CAE analysis for plastic parts. In general domain we could discuss about step change of the structural stiffness. The FSM (Fuel Supply Module) or only FSM flange was used for detail analysis of the applied damping.

Several stress peaks close to the ultimate stress could occur in the simulation as a secondary load after the crack propagation in the case that damping was not involved. The evaluation of these results was not possible. The damping had to be used for the correct consequent behaviour description and a possible damage accumulation. The deformation behaviour of the simulation with damping and the real experiment was compared. Only a few higher amplitudes occurred in both cases. The experiment with the real part showed no additional damage. The stress oscillation in the simulation was decreased onto safe values. We can use very simple experimental method for damping ratio determination. The wide frequency spectrum (low-frequencies) can be solved by this method.

8 Literature

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