Vibration, acoustic and fatigue solvers in LS-DYNA®

Presented at DYNAmore information day

Yun Huang, Zhe Cui

Livermore Software Technology Corporation

10 October, 2017

Stuttgart, Germany

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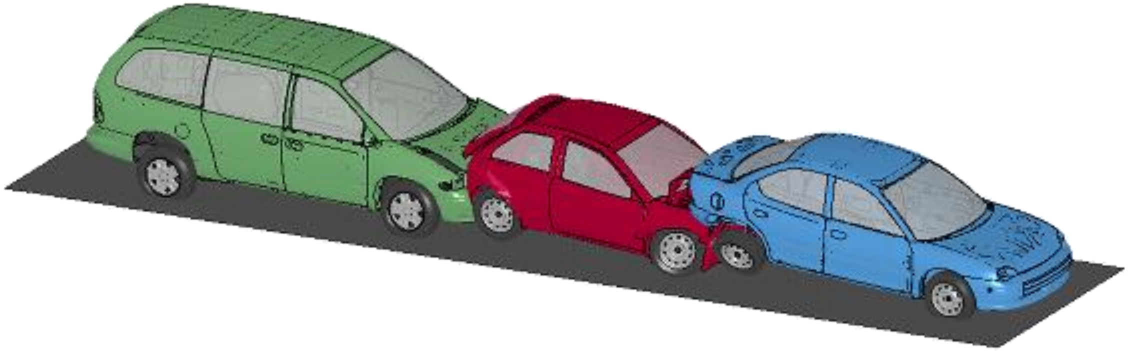
1) INTRODUCTION

Application of LS-DYNA in automotive industry



One code strategy

- In automotive, one model for crash, durability, NVH shared and maintained across analysis groups.
- Manufacturing simulation results from LS-DYNA used in crash, durability, and NVH modeling.

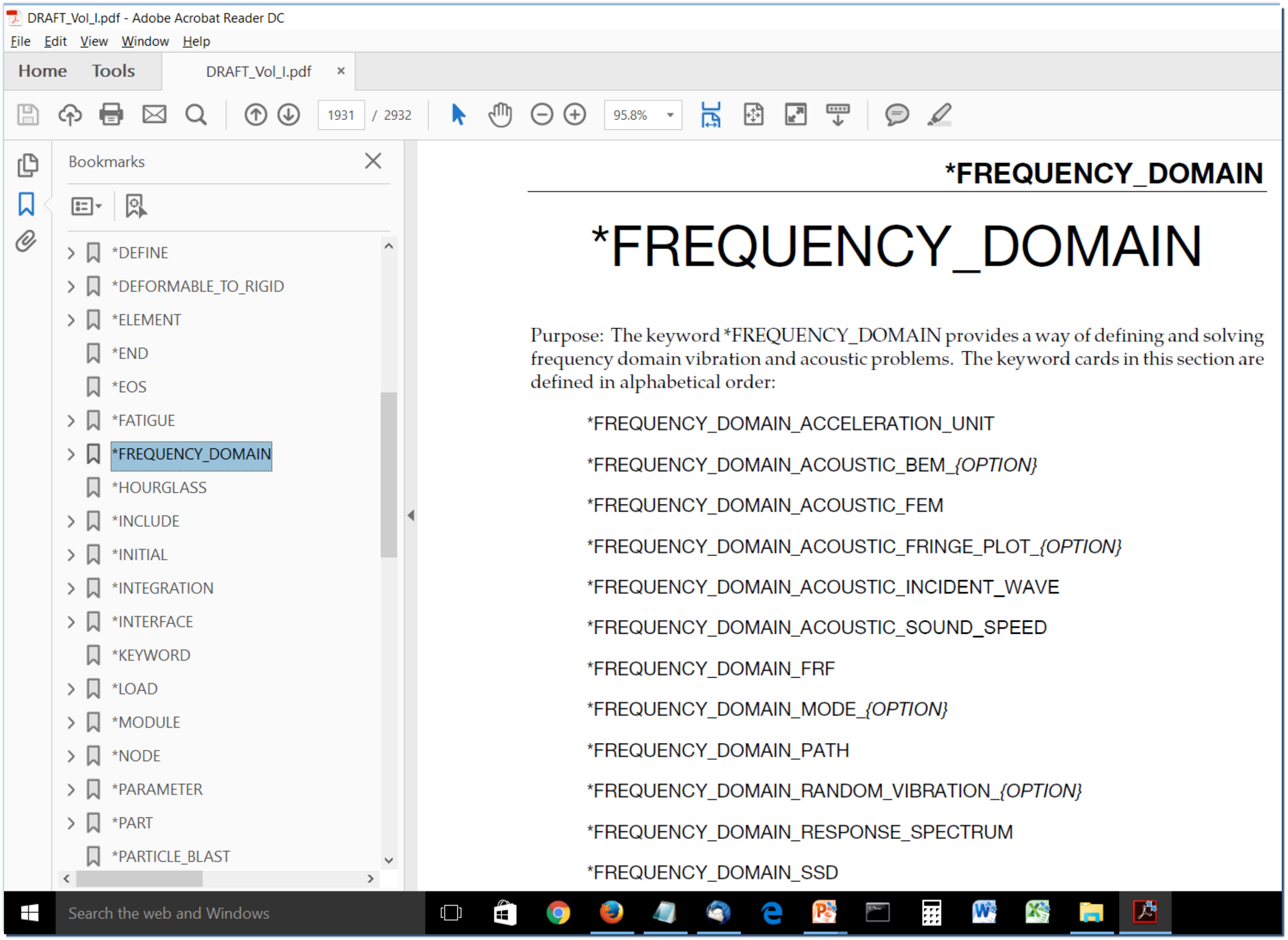
"All-in-one" package

Crashworthiness

Occupant Safety

NVH

Durability



*FREQUENCY_DOMAIN

*FREQUENCY_DOMAIN

Purpose: The keyword *FREQUENCY_DOMAIN provides a way of defining and solving frequency domain vibration and acoustic problems. The keyword cards in this section are defined in alphabetical order:

*FREQUENCY_DOMAIN_ACCELERATION_UNIT

FREQUENCY_DOMAIN_ACOUSTIC_BEM_*{OPTION}

*FREQUENCY_DOMAIN_ACOUSTIC_FEM

FREQUENCY_DOMAIN_ACOUSTIC_FRINGE_PLOT_*{OPTION}

*FREQUENCY_DOMAIN_ACOUSTIC_INCIDENT_WAVE

*FREQUENCY_DOMAIN_ACOUSTIC_SOUND_SPEED

*FREQUENCY_DOMAIN_FRF

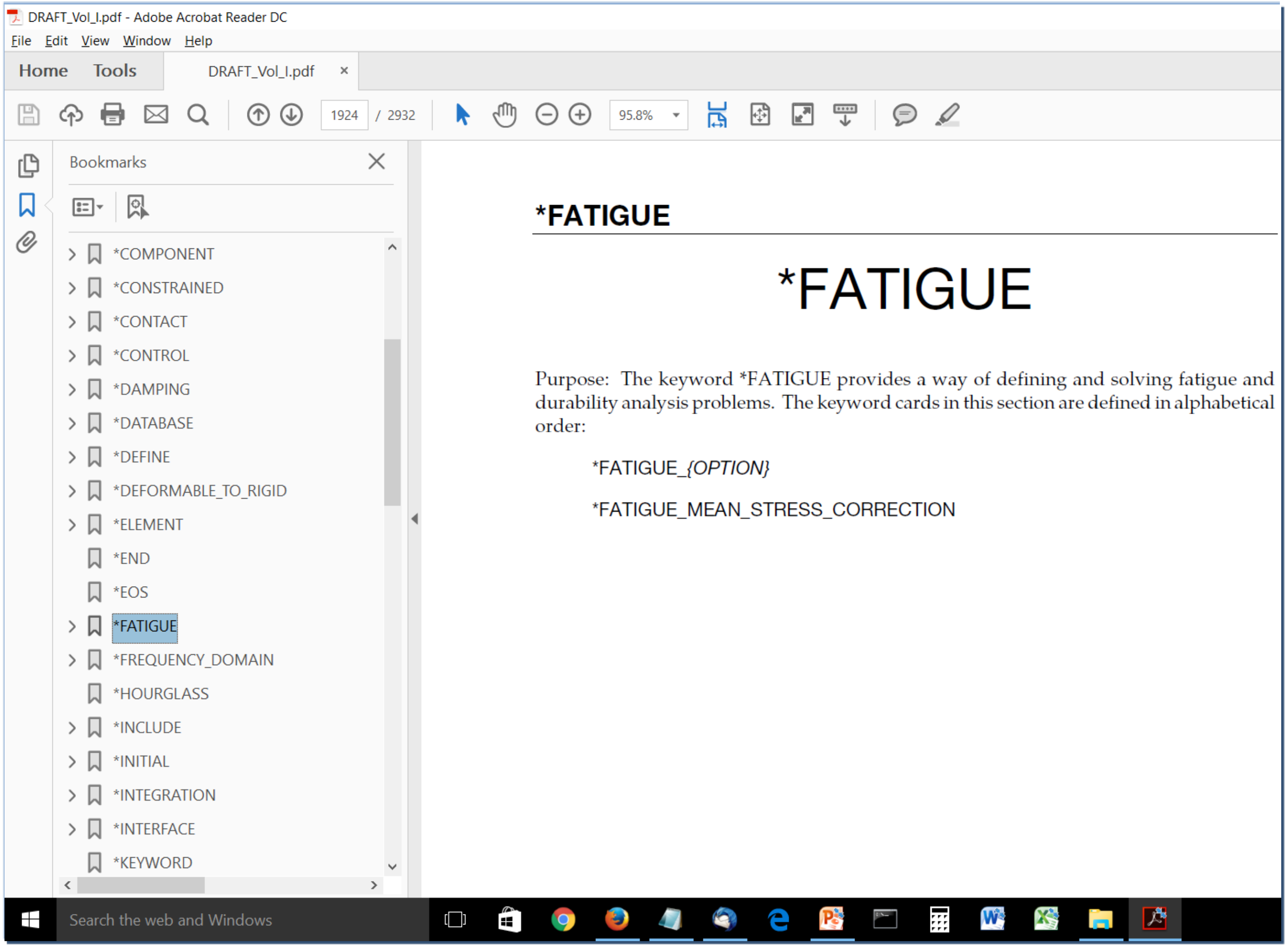
FREQUENCY_DOMAIN_MODE_*{OPTION}

*FREQUENCY_DOMAIN_PATH

FREQUENCY_DOMAIN_RANDOM_VIBRATION_*{OPTION}

*FREQUENCY_DOMAIN_RESPONSE_SPECTRUM

*FREQUENCY_DOMAIN_SSD



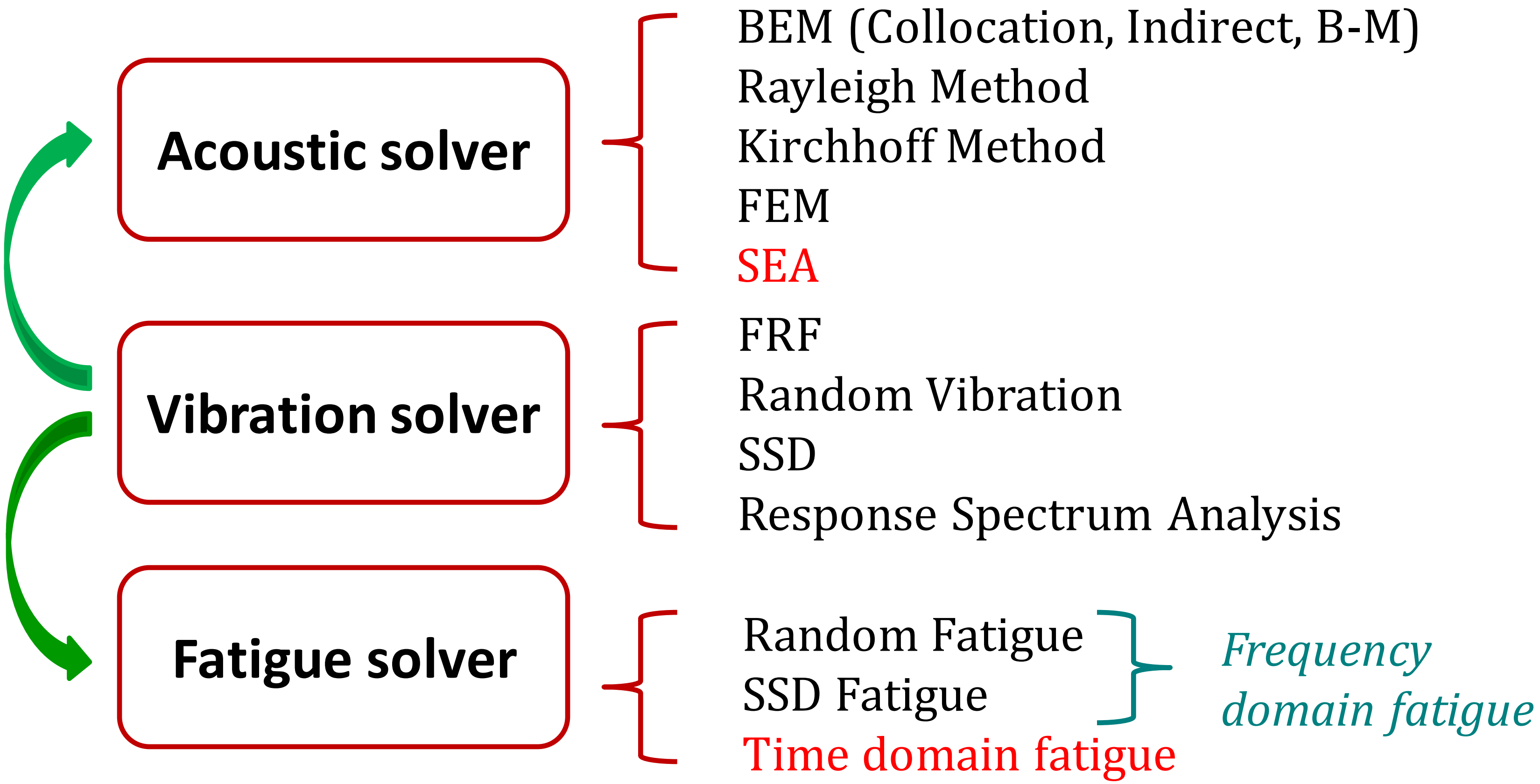
***FATIGUE**

*FATIGUE

Purpose: The keyword *FATIGUE provides a way of defining and solving fatigue and durability analysis problems. The keyword cards in this section are defined in alphabetical order:

FATIGUE_*{OPTION}

*FATIGUE_MEAN_STRESS_CORRECTION



Acoustic solver

- BEM (Collocation, Indirect, B-M)
- Rayleigh Method
- Kirchhoff Method
- FEM
- SEA

Vibration solver

- FRF
- Random Vibration
- SSD
- Response Spectrum Analysis

Fatigue solver

- Random Fatigue
- SSD Fatigue
- Time domain fatigue

Frequency domain fatigue

2) VIBRATION SOLVERS

2.1) FRF (frequency response function)

2.2) SSD (steady state dynamics)

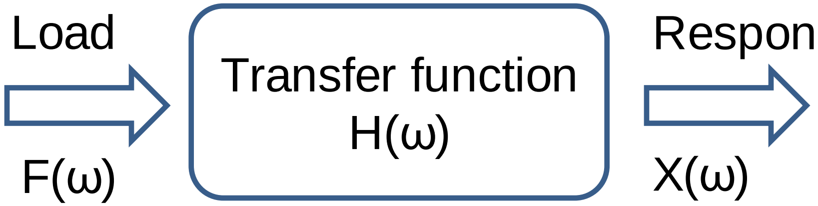
2.3) Random vibration analysis

2.4) Response spectrum analysis

2.5) Modal transient analysis

2.1) Frequency Response Function

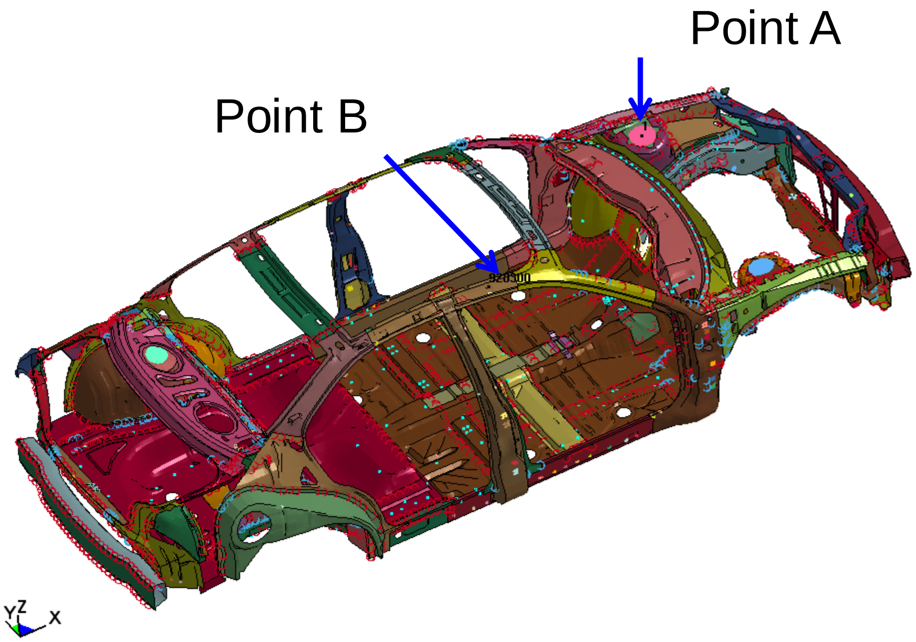
FRF (Frequency Response Function) provides a transfer function between excitations and response, and it can be used to locate the energy transfer path, or some important dynamic properties of structures

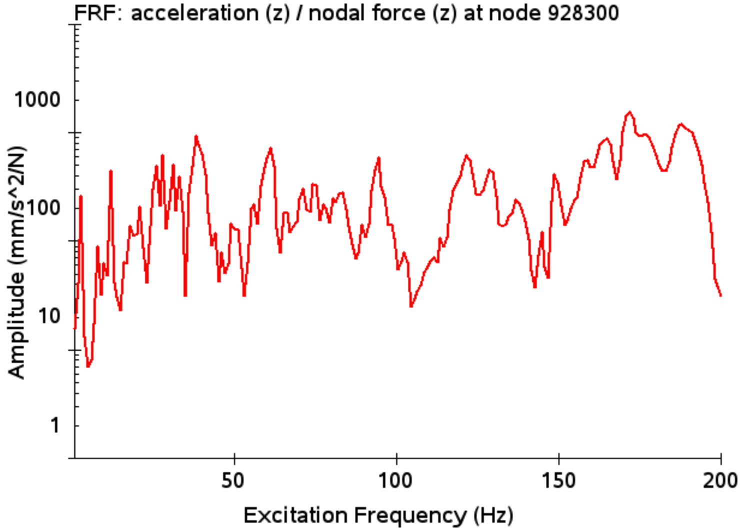


$$X(\omega) = H(\omega) \cdot F(\omega)$$

$$H(\omega) = \frac{X(\omega)}{F(\omega)}$$

FRF	Load	Response
Accelerance, Inertance	Force	Acceleration
Effective Mass	Acceleration	Force
Mobility	Force	Velocity
Impedance	Velocity	Force
Dynamic Compliance, Admittance, Receptance	Force	Displacement
Dynamic Stiffness	Displacement	Force





Nodal force is applied at point A, and FRF response in the form of accelerance is calculated at point B

Accelerance at point B

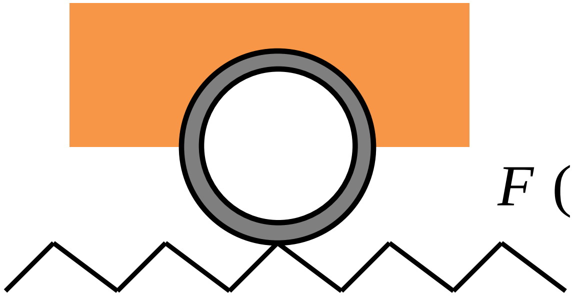
Engineering Research Nordic AB: Nilsson, Larsgunnar, "Model Frequency Response Analysis in LS-DYNA - Application on a BIW Railway Car", *2010 Nordic LSDYNA Users Forum*, Gothenburg, Sweden, October, 2010.

2.2) Steady State Dynamics

SSD (steady state dynamics) provides the steady state dynamic response of structures, subject to harmonic excitation.

Background

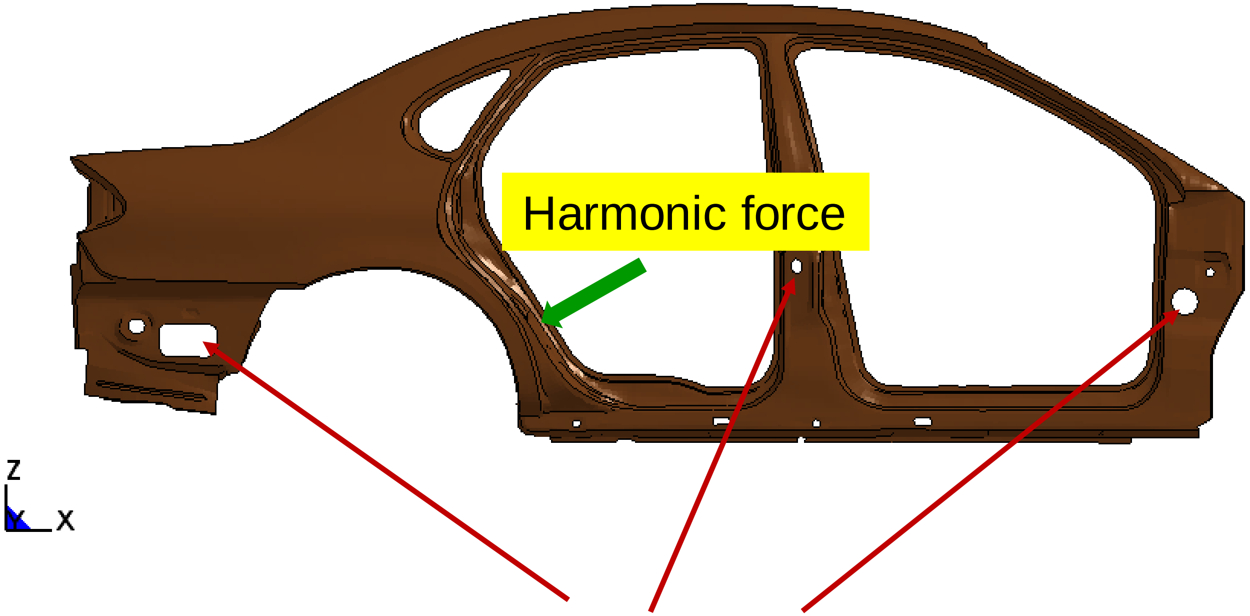
- Harmonic excitation is often encountered in engineering systems. It is commonly produced by the unbalance in rotating machinery.
- The load may also come from periodic load, e.g. in fatigue test.
- The excitation may also come from uneven base, e.g. the force on tires running on a zig-zag road (rough road shake test)
- May be called as
  - ✓ Harmonic vibration
  - ✓ Steady state vibration
  - ✓ Steady state dynamics



$$F(t) = F_0 \sin(\omega t + \phi)$$

IFB Automotive Pvt Ltd: Nirmal Gilbert & Aravind YS, “*Steady State Dynamics analysis on a automotive shroud using LS-DYNA*”, 2015 Kaizenat LS-DYNA users conference, Bangalore and Pune, India.

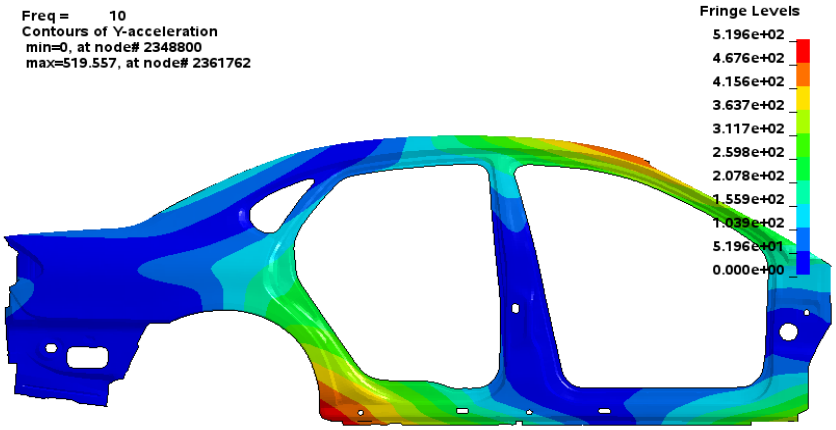
SSD analysis on a side frame model

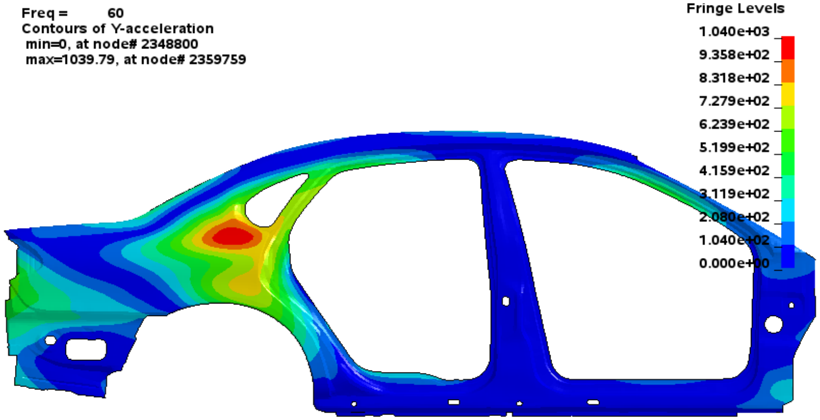


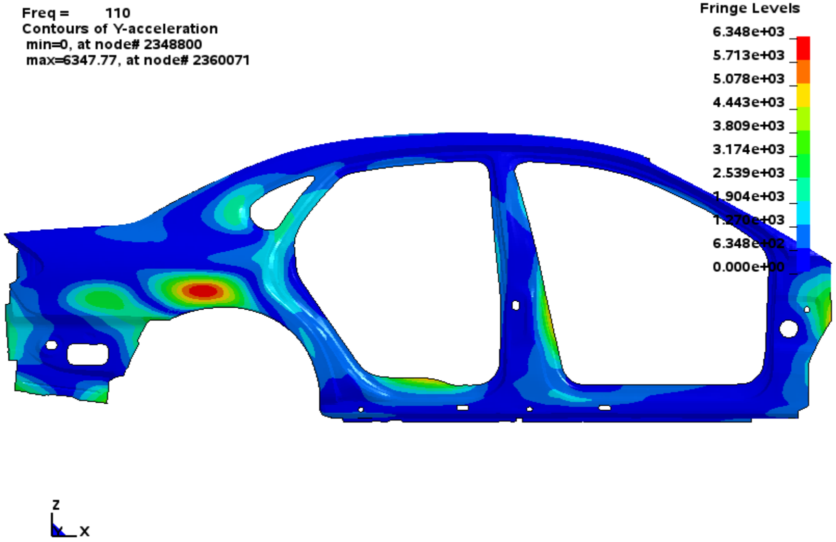
The frame is constrained via the 3 holes

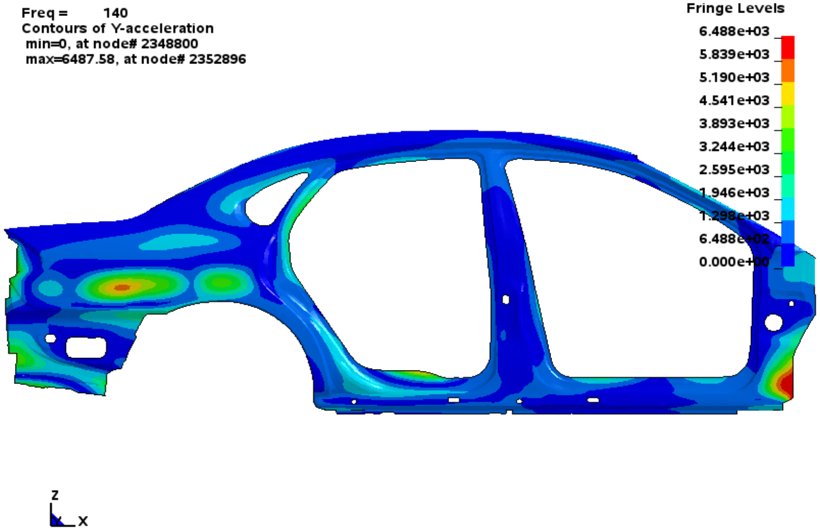
The excitation is given in the range of 10-140 Hz, in the form of harmonic unit nodal force in the direction vertical to the frame.

Acceleration SSD (by d3ssd)





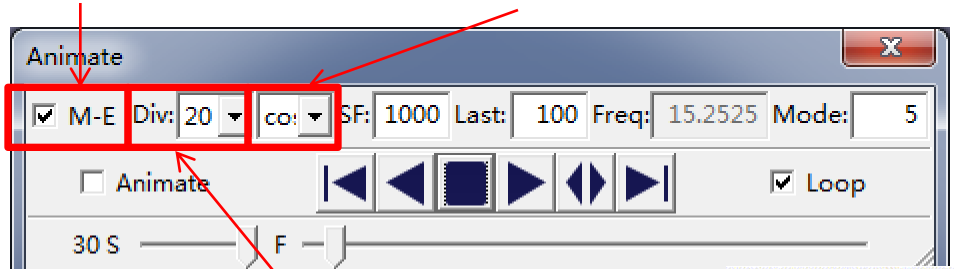




- Modal expansion (for D3SSD complex variable version)

Modal expansion

Function type (sin, cos)



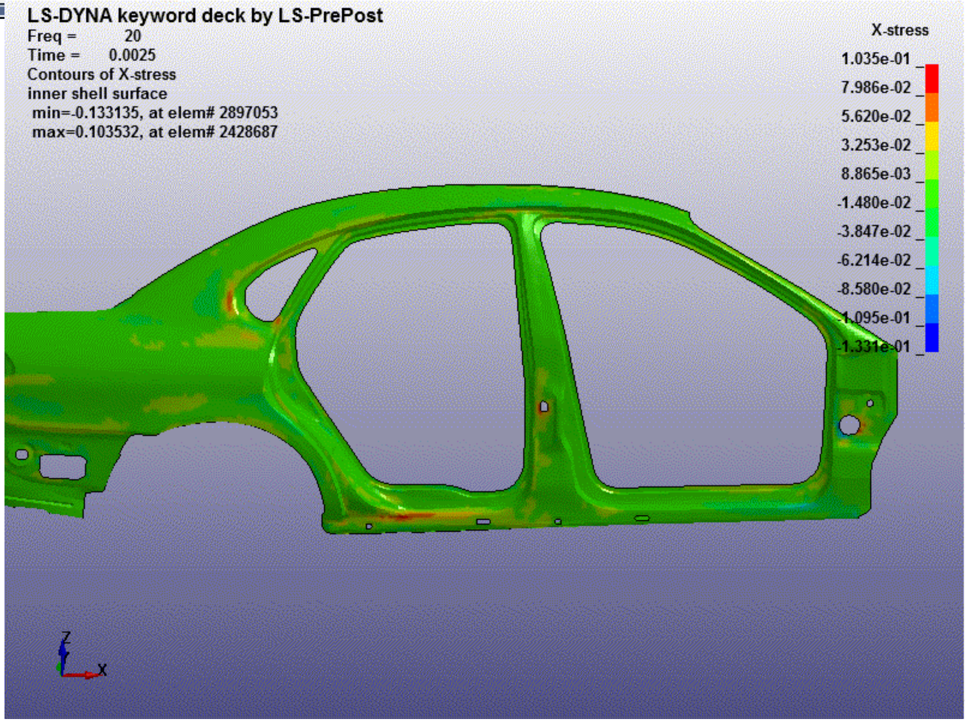
Animate

☑ M-E Div: 20 cos SF: 1000 Last: 100 Freq: 15.2525 Mode: 5

☐ Animate ☑ Loop

30 S F

Time step (N)



LS-DYNA keyword deck by LS-PrePost
Freq = 20
Time = 0.0025
Contours of X-stress
inner shell surface
min=-0.133135, at elem# 2897053
max=0.103532, at elem# 2428687

Equivalent Radiated Power (ERP)

FREQUENCY_DOMAIN_SSD_*{ERP}

Acoustic intensity

$$I\left(r_P\right)=\frac{1}{2}\mathrm{Re}\left[p\left(r_P\right)\cdot v_n\left(r_P\right)^*\right]$$

ERP density

$$ERP_\rho=\frac{1}{2}\rho_F c_F V_n \overline{V_n}$$

ERP absolute

$$ERP_{abs}=\int_S ERP_\rho dS$$

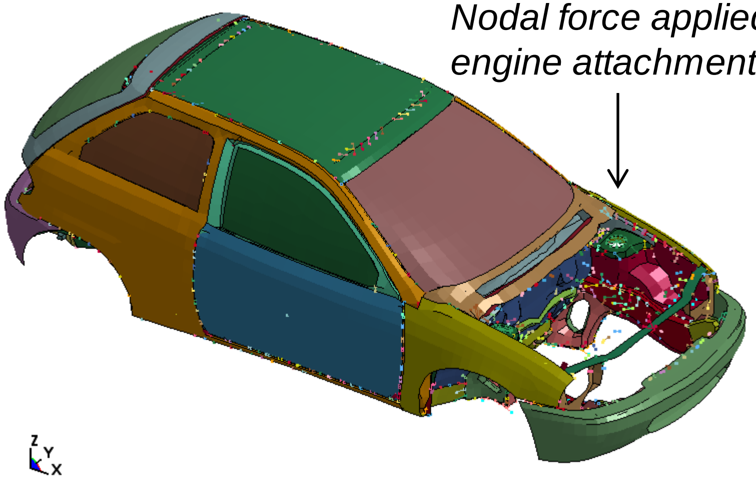
ERP in dB

$$ERP_{dB}=10\log_{10}\left(ERP_{abs}/ERP_{ref}\right)$$

Calculation of ERP is a simple and fast way to characterize the structure borne noise. It gives user a good look at how panels contribute to total noise radiation. It is a valuable tool in early phase of product development.

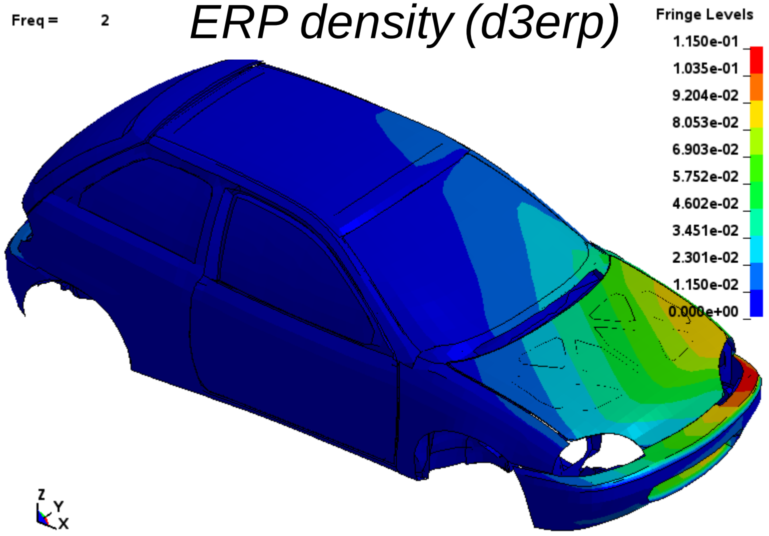
ERP calculation results are saved in

- *Binary database*
  - ✓ *d3erp*
- *ASCII xyplot files*
  - ✓ *ERP_abs*
  - ✓ *ERP_dB*

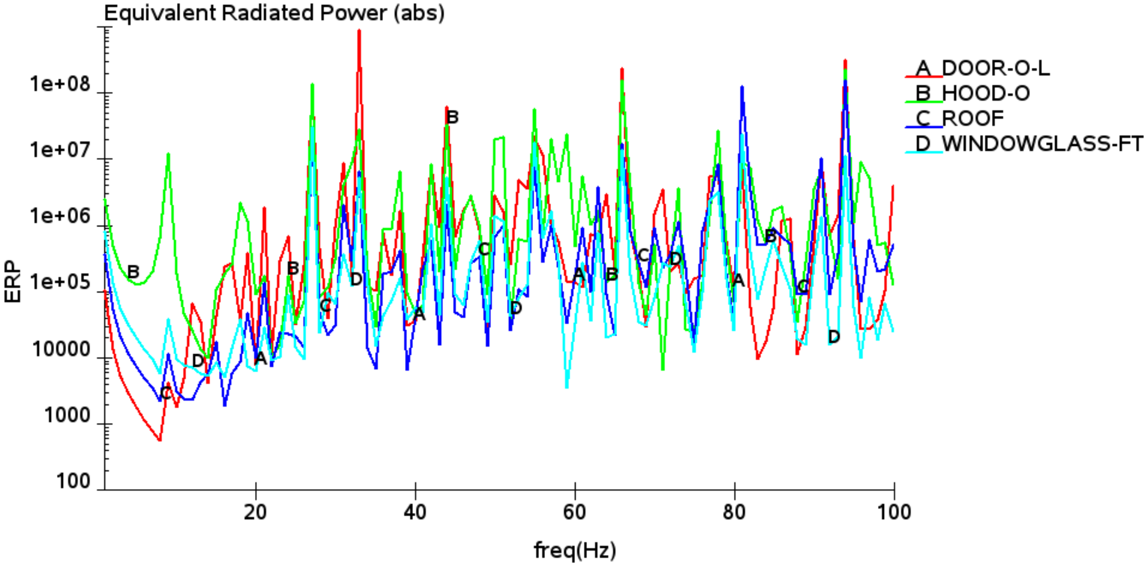


Nodal force applied at engine attachment point

ERP density (d3erp)



With ERP, one can study the contribution to the radiated noise from each panel.



Damping options

*FREQUENCY_DOMAIN_SSD

Card 2	1	2	3	4	5	6	7	8
Variable	DAMP	LCDAM	LCTYP	DMPMAS	DMPSTF	DMPFLG		
Type	F	I	I	F	F	I		
Default	0.0	0	0	0.0	0.0	0		

- Viscous damping $F_v = c \cdot v$
- Structural damping $F_s = i \cdot G \cdot k \cdot u$

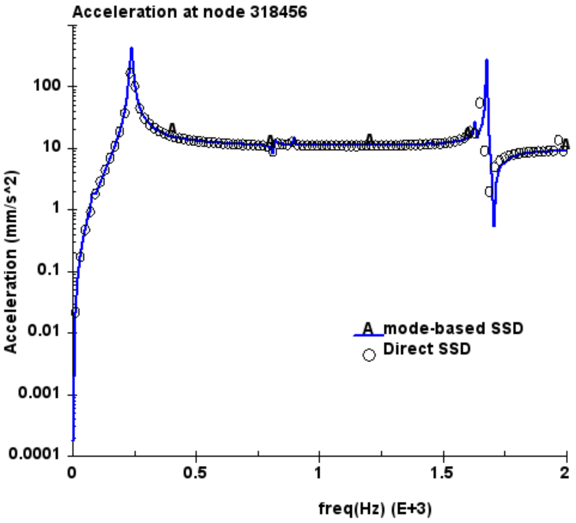
***DAMPING_PART_MASS**
***DAMPING_PART_STIFFNESS**
***DAMPING_STRUCTURAL**
***MAT_DAMPER_VISCOUS**

To be implemented / tested

Local viscous damping from more material models:
***MAT_LINEAR_ELASTIC_DISCRETE_BEAM**
***MAT_ELASTIC_SPRING_DISCRETE_BEAM**
Local structural damping

Option: direct solver

FREQUENCY_DOMAIN_SSD_*{DIRECT}

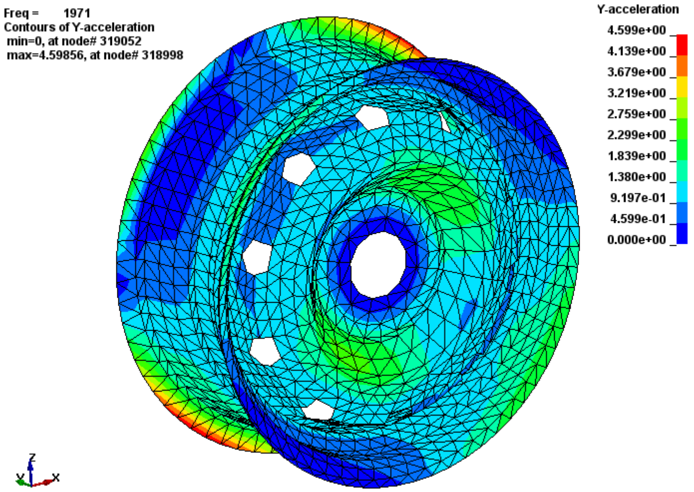


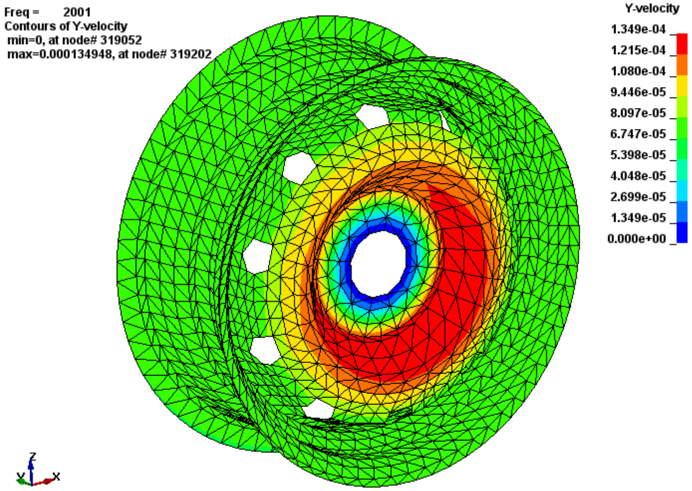
Indirect solver based on eigenmodes

- *Fast*
- *Constant material properties*

Direct solver (physical coordinates)

- *Slow*
- *Frequency dependent material properties*

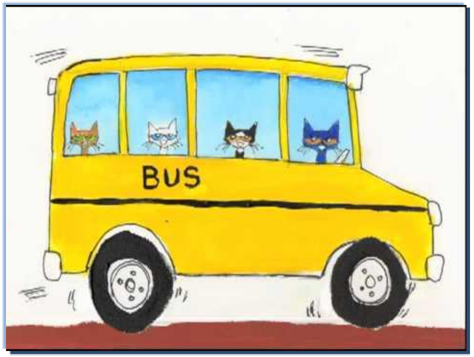




2.3) Random vibration analysis

Why we need random vibration analysis?

- The loading on a structure is not known in a definite sense
- Many vibration environments are not related to a specific driving frequency (may have input from multiple sources)
- Examples:
  - Wind-turbine
  - Air flow over a wing or past a car body
  - Acoustic input from jet engine exhaust
  - Earthquake ground motion
  - Wheels running over a rough road
  - Ocean wave loads on offshore platforms



"Multi Axis Shaker Table" by Davevandongen - Moog FCS company.
Licensed under Creative Commons Attribution 3.0 via Wikimedia Commons - http://commons.wikimedia.org/wiki/File:Multi_Axis_Shaker_Table.jpg#mediaviewer/File:Multi_Axis_Shaker_Table.jpg

Load

- Base acceleration
- Random pressure
- Plane wave
- Random progressive wave
- Reverberant wave
- Turbulent boundary layer
- Nodal force

Pre-stress condition

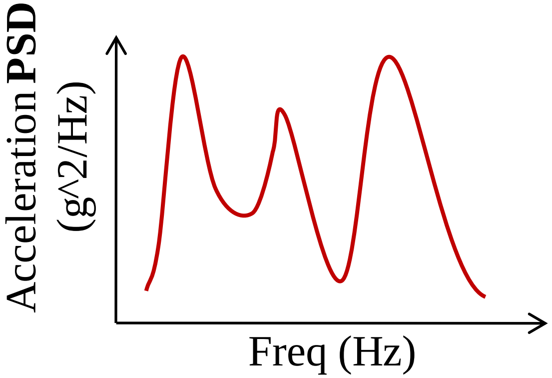
- Thermal pre-stress*
- Mechanical pre-stress

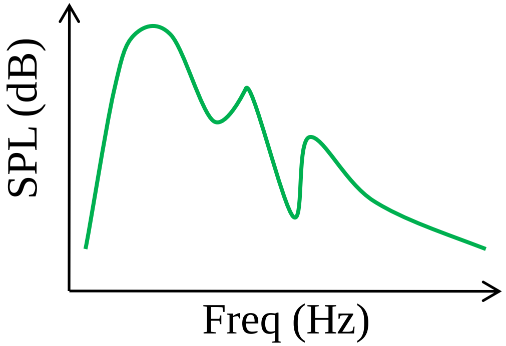
Results

- PSD of u, v, a and stresses
- RMS of u, v, a and stresses
- zero-crossing frequencies

Input curve:

PSD or SPL





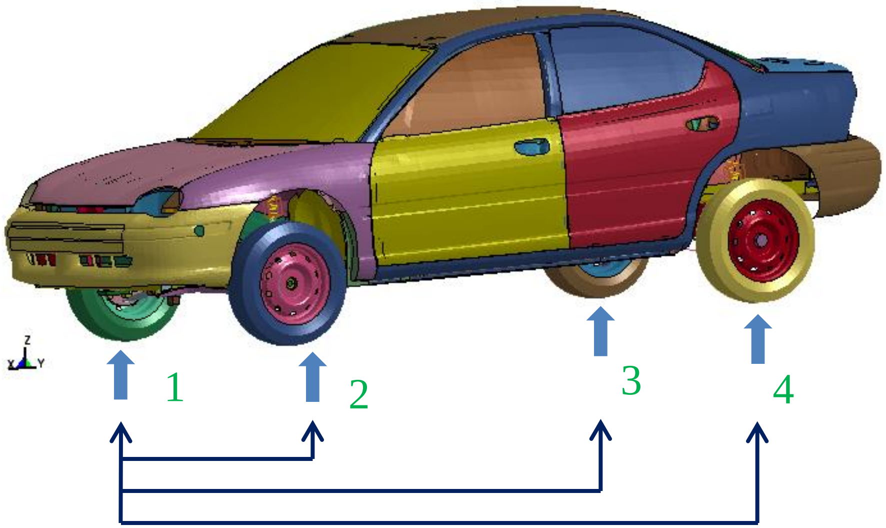
***Aerospace Portfolio, National Research Council, Canada:** Devon Downes, Manouchehr Nejad Ensan, *"Vibration Analysis of a Compressor Blade at High Temperature",* 14th International LS-DYNA Users Conference, Dearborn, Michigan, June 2016.

Auto PSD Cards. Include **NASPD** cards of this format, one per excitation.

Card 5a	1	2	3	4	5	6	7	8
Variable	SID	STYPE	DOF	LDPSD	LDVEL	LDFLW	LDSPN	CID
Type	I	I	I	I	I	I	I	I
Default					0	0	0	0

Cross PSD Card. Include **NCPSD** cards of this format, one per excitation.

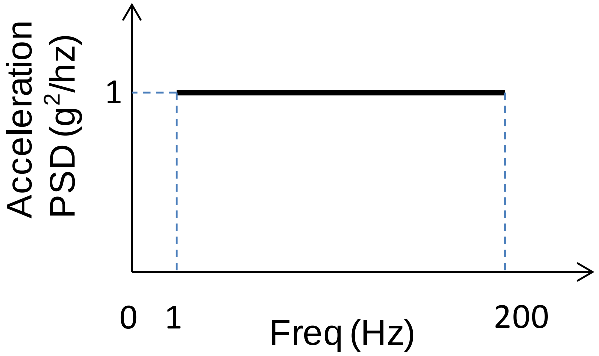
Card 5b	1	2	3	4	5	6	7	8
Variable	LOAD_I	LOAD_J	LCTYP2	LDPSD1	LDPSD2			
Type	I	I	I	I	I			
Default			0					



Mathematical analogy

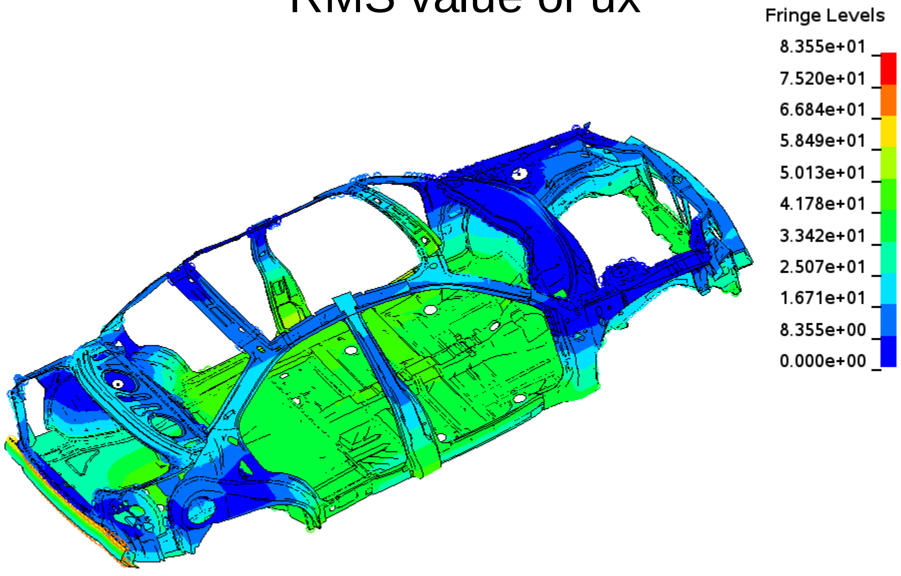
$$(x+y)^2 = x^2 + y^2 + 2xy$$

Random vibration analysis on a BIW model

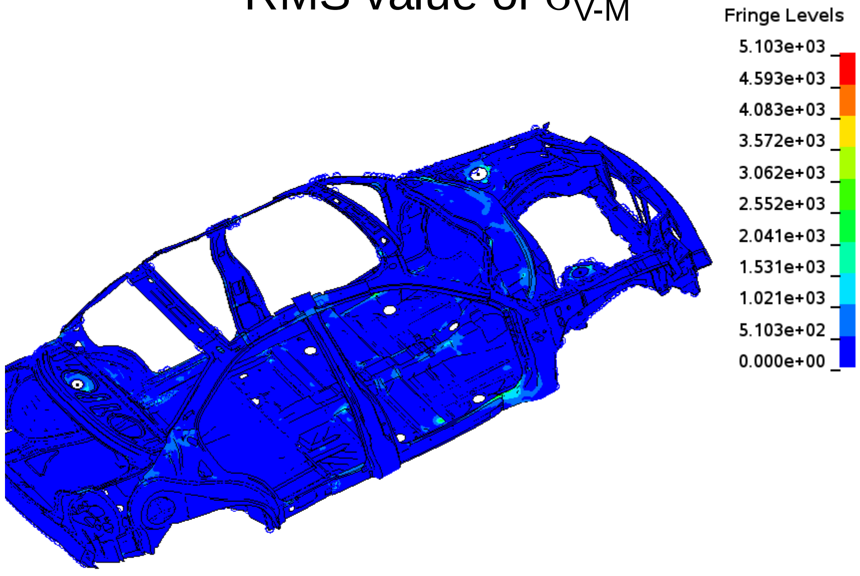


Random vibration analysis for the BIW model under base acceleration PSD excitation. The acceleration is specified in x-direction The PSD curve is given as white noise (1*g*^2/Hz) for the range of 1-200 Hz as follows.

RMS value of ux

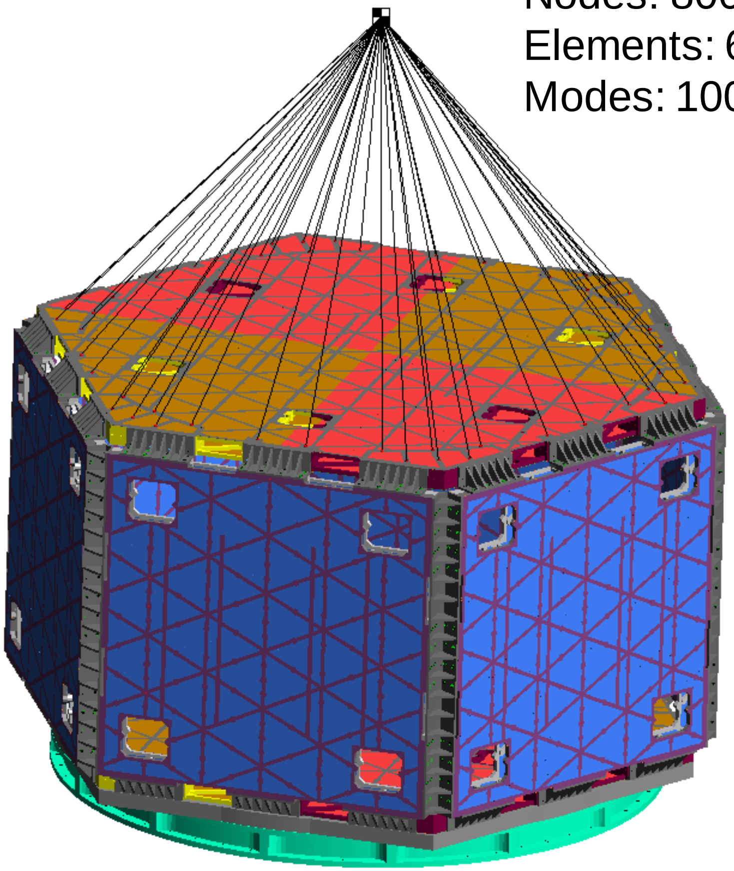


RMS value of $\sigma_{V\text{-}M}$

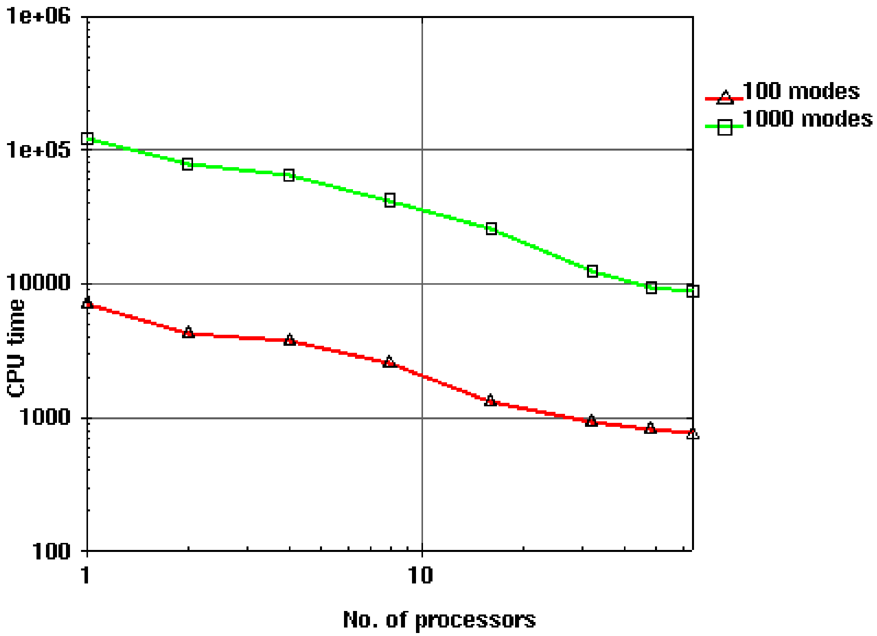


Random vibration analysis with MPP

Nodes: 800k
Elements: 660k
Modes: 1000



Model courtesy of Predictive Engineering



"Broad-Spectrum Stress and Vibration Analysis of Large Composite Container", Adrian Jensen, George Laird (**Predictive Engineering, Inc.**), Adrian Tayne (**ECS Case, Becklin Holdings, Inc.**), 14th International LS-DYNA Users Conference, Dearborn, Michigan, June 2016.

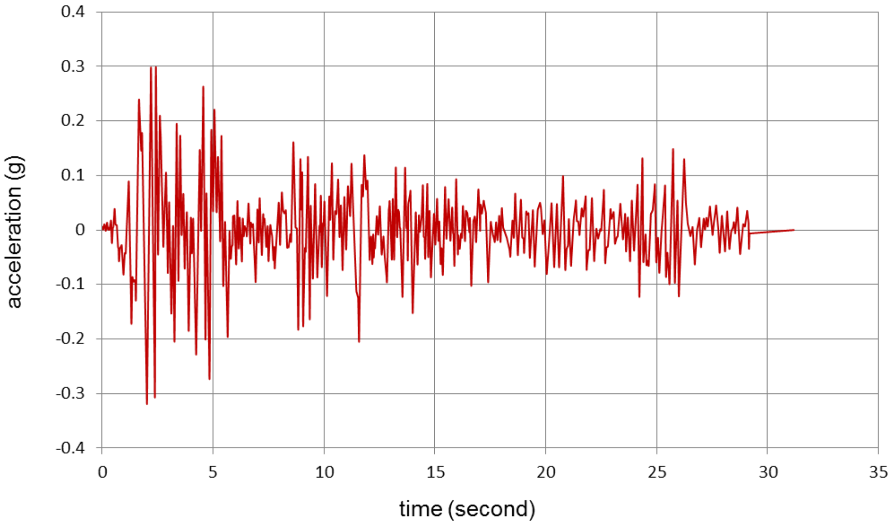
Rafael, Israel: Shor, O., Lev, Y., and Huang, Y., "*Simulation of a Thin Walled Aluminum Tube Subjected to Base Acceleration Using LS-DYNA's Vibro-Acoustic Solver*", 11th International LS-DYNA Users Conference, Dearborn, Michigan, June, 2010.

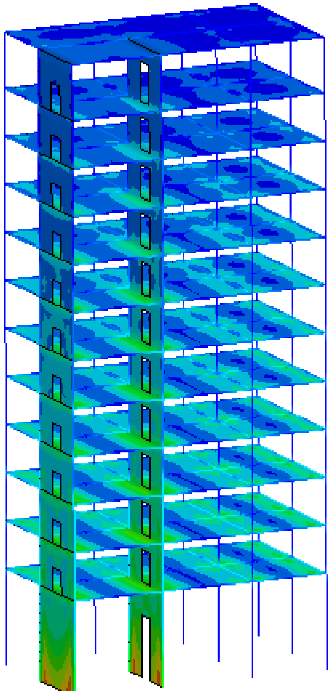
The Boeing Company: Rassaian M., Arakawa T., Huang Y., "*Structural analysis with vibro-acoustic loads*", 2012 Aircraft Airworthiness & Sustainment Conference (AA&S 2012), Baltimore, Maryland, April 2012.

GIGABYTE： Dongke Lu，"*Vibration fatigue analysis of computer servers*", the 2nd China LS-DYNA Users Conference, Shanghai, China, November, 2015.

2.4) Response spectrum analysis

- ☐ Use various mode combination methods to evaluate peak response of structure due to input spectrum.
- ☐ The input spectrum is the peak response (acceleration, velocity or displacement) of single dof system with different natural frequencies.
- ☐ Multiple curves to define the series of excitation spectrum corresponding to different damping ratio.
- ☐ It is an approximate method, but fast and effective.





Mode combination

- ○ SRSS method
- ○ NRC Grouping method
- ○ CQC method
- ○ Double Sum methods
  - ✓ Rosenblueth-Elorduy coefficient
  - ✓ Gupta-Cordero coefficient
  - ✓ Modified Gupta-Cordero coefficient
- ○ NRL SUM method
- ○ Rosenblueth method

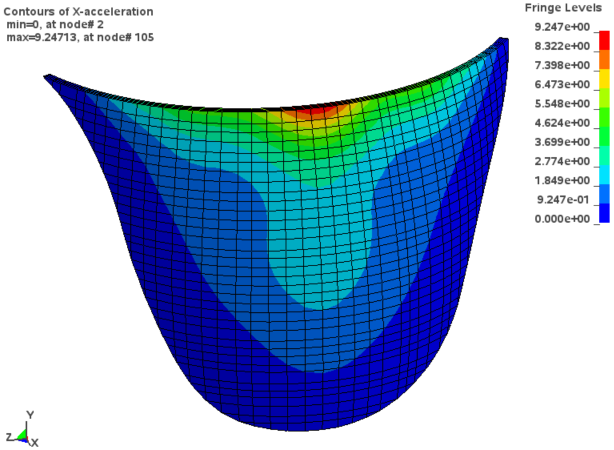
Input spectrum

- ○ Base velocity
- ○ Base acceleration
- ○ Base displacement
- ○ Nodal force
- ○ Pressure

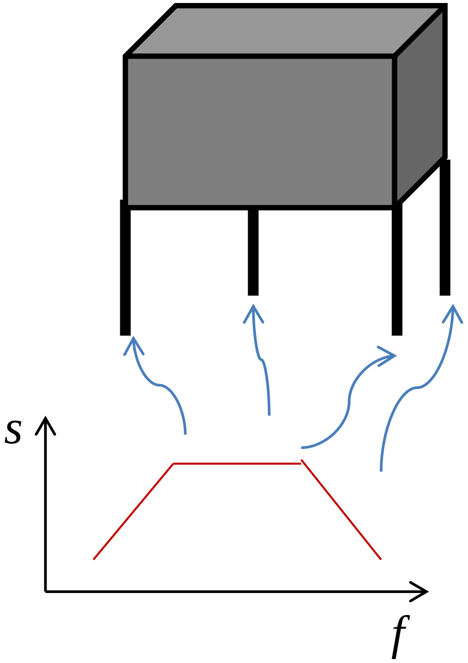
Applications

- ○ Civil / Hydraulic buildings
  - ✓ Dams
  - ✓ Bridges
  - ✓ High buildings
- ○ Nuclear power plants

El Atazar Dam

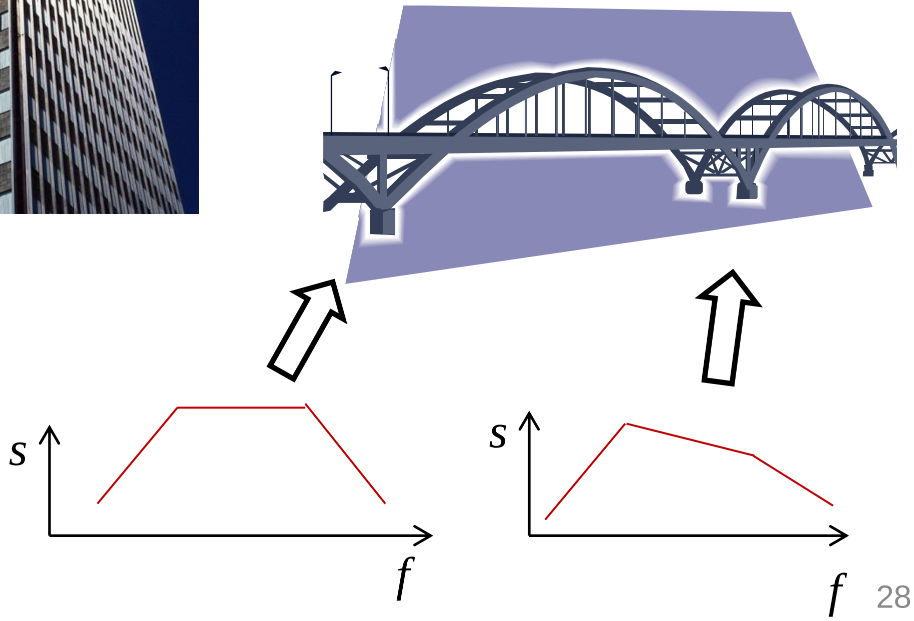


Multi-point response spectrum analysis



s = spectral value
f = frequency

A SRSS method is used to combine the responses for each of the input spectra.



Option: DDAM

DDAM (Dynamic Design Analysis Method)

- US Navy-developed analytical procedure
- It evaluates the design of equipment subject to dynamic loading caused by underwater explosions (UNDEX).
- The analysis uses a form of shock spectrum analysis that estimates the dynamic response of a component to shock loading caused by the sudden movement of a naval vessel.
- The analytical process simulates the interaction between the shock-loaded component and its fixed structure.
- It is a standard naval engineering procedure for shipboard structural dynamics.

All mission-essential equipment on board Naval ships and submarines must be qualified for shock loads caused by underwater explosions (UNDEX)

Shock Design Values

- NRL-1396
- User defined

Ship type

- Submarine
- Surface ship

Mounting type

- Hull mounted
- Deck mounted
- Shell Plating mounted

Material type

- Elastic
- Elasto-plastic

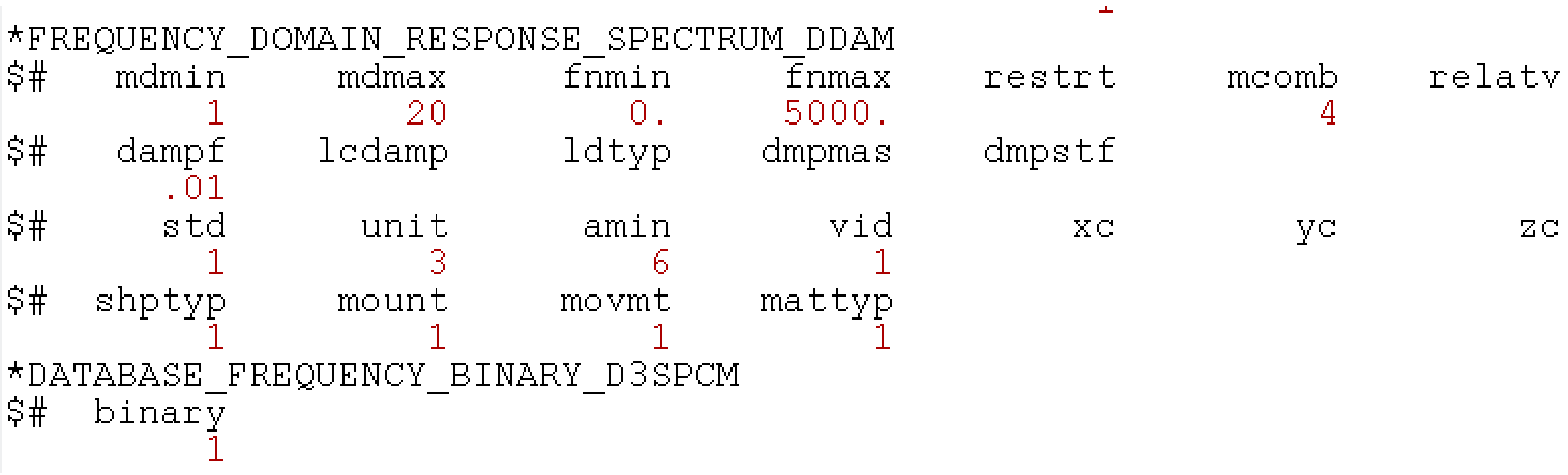
Load direction

- Vertical
- Athwartship
- Fore and Aft

Mode combination

- NRL Sum
- NRL Sum with CSM

"NAVSEA 0908-LP-000-3010 (Revision 1), Shock Design Criteria for Surface Ships". Naval Sea Systems Command. September 1995.



```
*FREQUENCY_DOMAIN_RESPONSE_SPECTRUM_DDAM
$#   mdmin     mdmax     fnmin     fnmax    restrt     mcomb    relatv
         1        20        0.     5000.                   4
$#   dampf    lcdamp     ldtyp    dmpmas    dmpstf
       .01
$#     std      unit      amin       vid        xc        yc        zc
         1         3         6         1
$#  shptyp     mount     movmt    mattyp
         1         1         1         1
*DATABASE_FREQUENCY_BINARY_D3SPCM
$#  binary
         1
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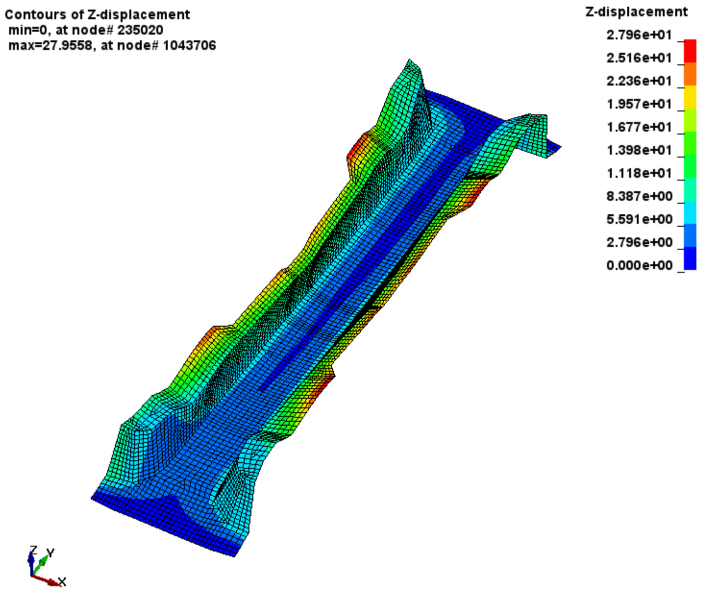
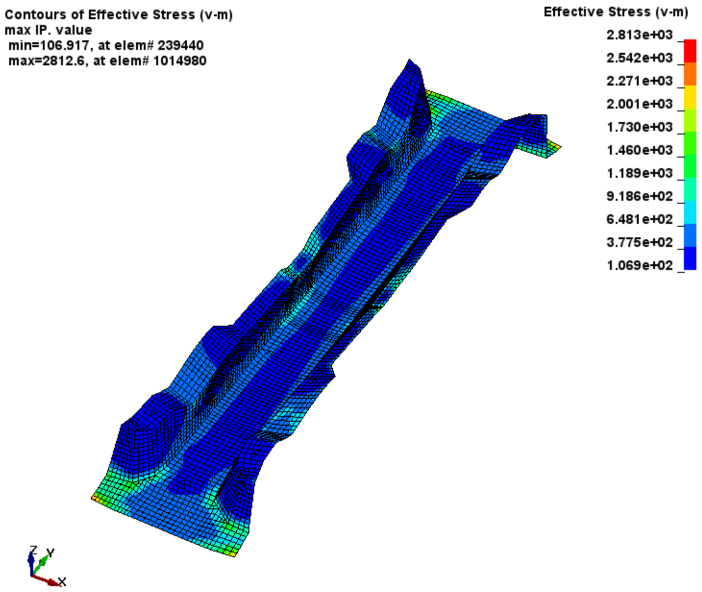




2.5) Modal transient analysis

***CONTROL_IMPLICIT_MODAL_DYNAMIC**

***CONTROL_IMPLICIT_MODAL_DYNAMIC_DAMPING**

***CONTROL_IMPLICIT_MODAL_DYNAMIC_MODE**

$$\mathbf{m}\ddot{\mathbf{u}} + \mathbf{c}\dot{\mathbf{u}} + \mathbf{k}\mathbf{u} = \mathbf{p}(t)$$

$$\boxed{\mathbf{u} = \sum_{n=1}^{N} \phi_n q_n(t) = \boldsymbol{\Phi}\mathbf{q}}$$

$$\boldsymbol{\Phi}^T \mathbf{m}\boldsymbol{\Phi}\ddot{\mathbf{q}} + \boldsymbol{\Phi}^T \mathbf{c}\boldsymbol{\Phi}\dot{\mathbf{q}} + \boldsymbol{\Phi}^T \mathbf{k}\boldsymbol{\Phi} q = \boldsymbol{\Phi}^T \mathbf{p}(t)$$

Significant improvement on the performance, by David Benson, in responding to request from Tesla Motors.

3) ACOUSTIC SOLVERS

3.1) Introduction

3.2) BEM acoustic solver

3.3) FEM acoustic solver

3.4) SEA for high frequency acoustics

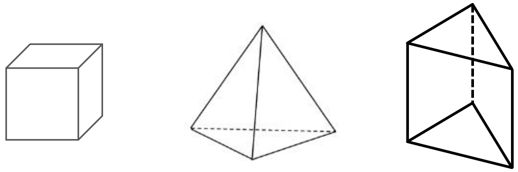
3.1) Introduction

Time domain acoustic solver in LS-DYNA

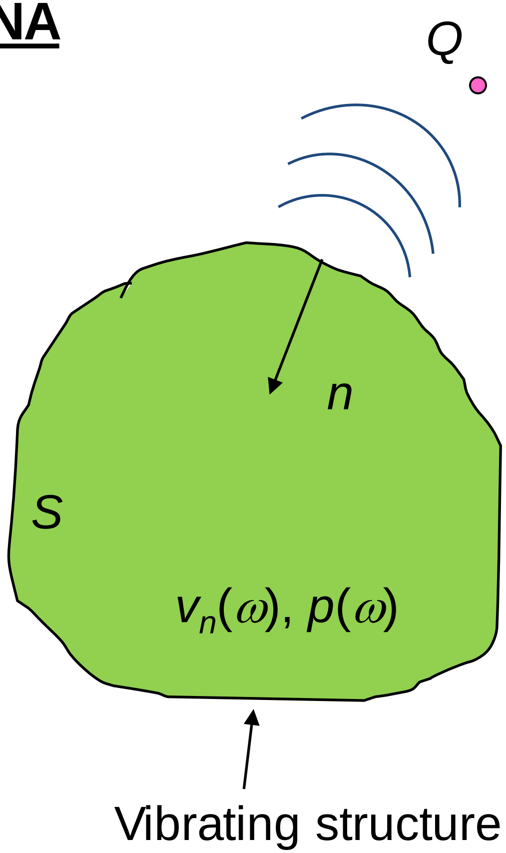
- ✓ MAT_ACOUSTIC with solid element 8 or 14

Frequency domain acoustic solver in LS-DYNA

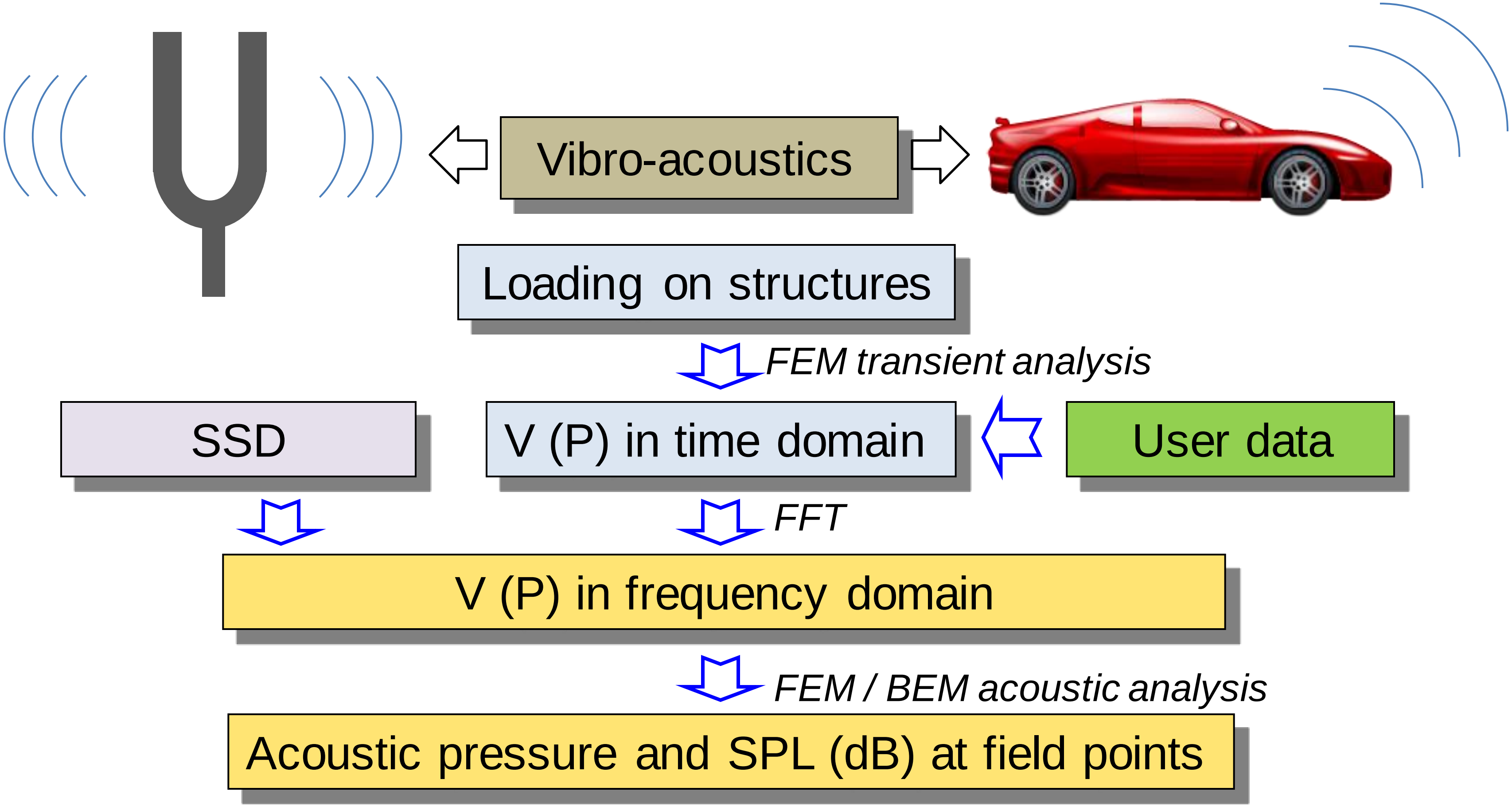
- ✓ FREQUENCY_DOMAIN_ACOUSTIC_BEM
  - ○ Rayleigh method
  - ○ Kirchhoff method
  - ○ Collocation BEM
  - ○ Variational indirect BEM
- ✓ FREQUENCY_DOMAIN_ACOUSTIC_FEM
  - ○ Hexahedron
  - ○ Tetrahedron
  - ○ Pentahedron



- ✓ SEA *(ongoing development)*



Vibro-acoustic analysis



Vibro-acoustics

Loading on structures

FEM transient analysis

SSD

V (P) in time domain

User data

FFT

V (P) in frequency domain

FEM / BEM acoustic analysis

Acoustic pressure and SPL (dB) at field points

National Taipei University of Technology, Taiwan: Guo-Ding Huang, Hsiu-Ying Hwang, Xijun Wang, “*Vibration Testing and Analysis for a Midsize Electric Bus*”, Proceedings of the 19th National Conference on Vehicle Engineering, Nov. 14, 2014, TIIT, Jhongli, Taiwan.

3.2) BEM acoustic solver

BEM (accurate)

- Indirect variational boundary element method
- Collocation boundary element method
  - Burton-Miller formulation
  - Sound power and radiation efficiency are computed

They used to be time consuming

A fast solver based on domain decomposition

MPP version

Approximate (simplified) methods

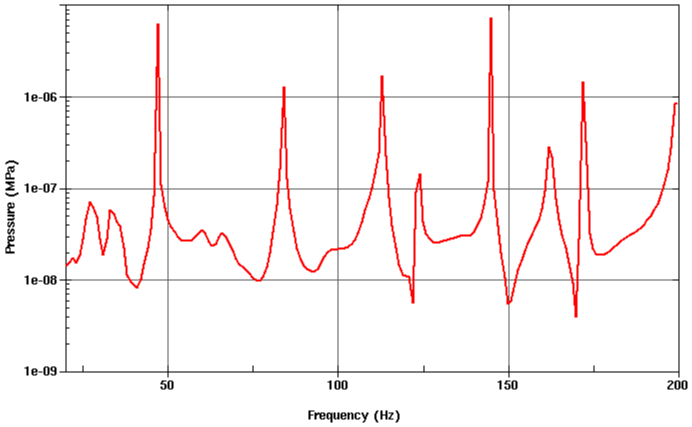
- Rayleigh method
- Kirchhoff method

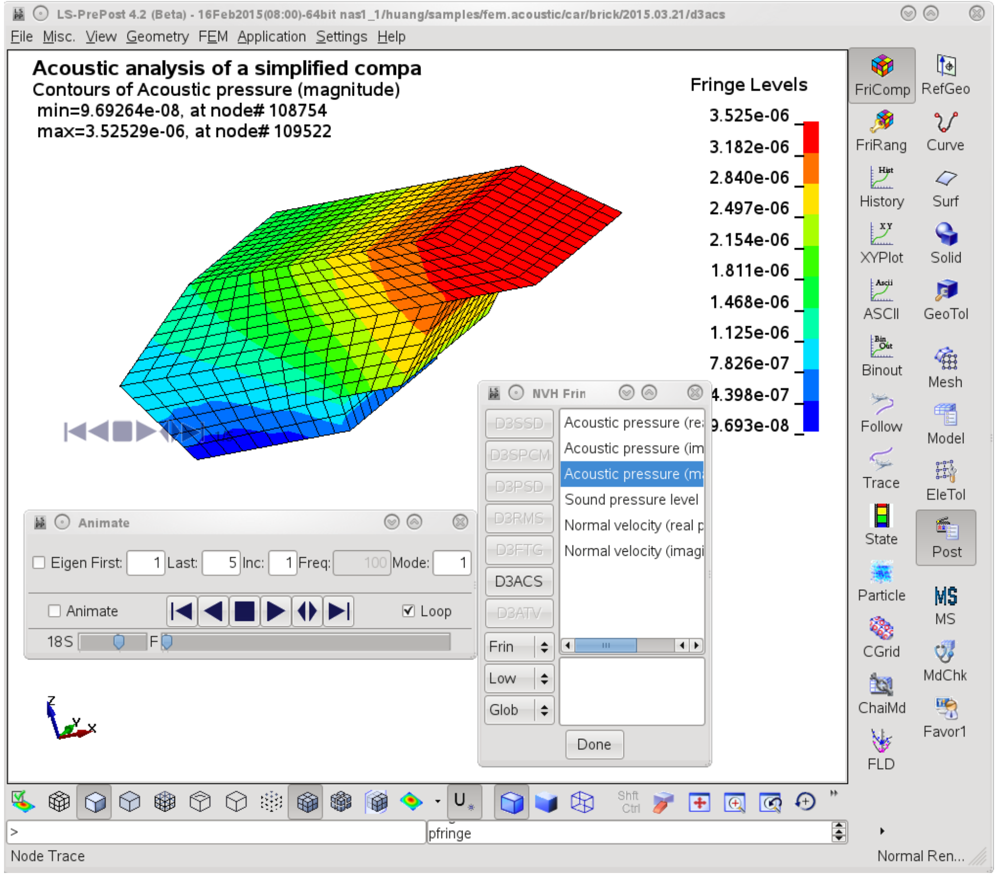
Assumptions and simplification in formulation

Very fast since no equation system to solve

Frequency domain acoustic results

- ✓ ASCII xplot files
  - *panel_contribution_NID*
  - *Press_Pa*
  - *Press_dB*
  - *Bepres*
- ✓ Binary plot databases
  - *D3ACS*





Acoustic analysis of a simplified compa
Contours of Acoustic pressure (magnitude)
min=9.69264e-08, at node# 108754
max=3.52529e-06, at node# 109522

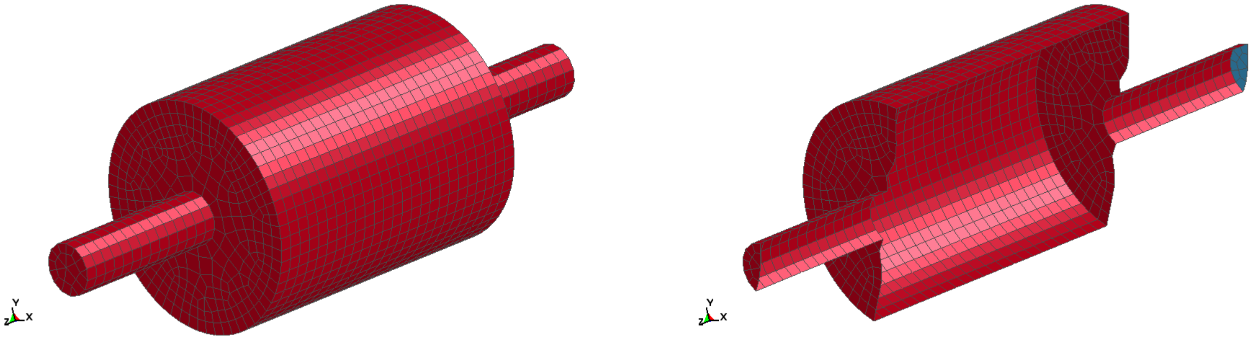
☐ Eigen First: 1 Last: 5 Inc: 1 Freq: 100 Mode: 1

☐ Animate ☑ Loop

18S F

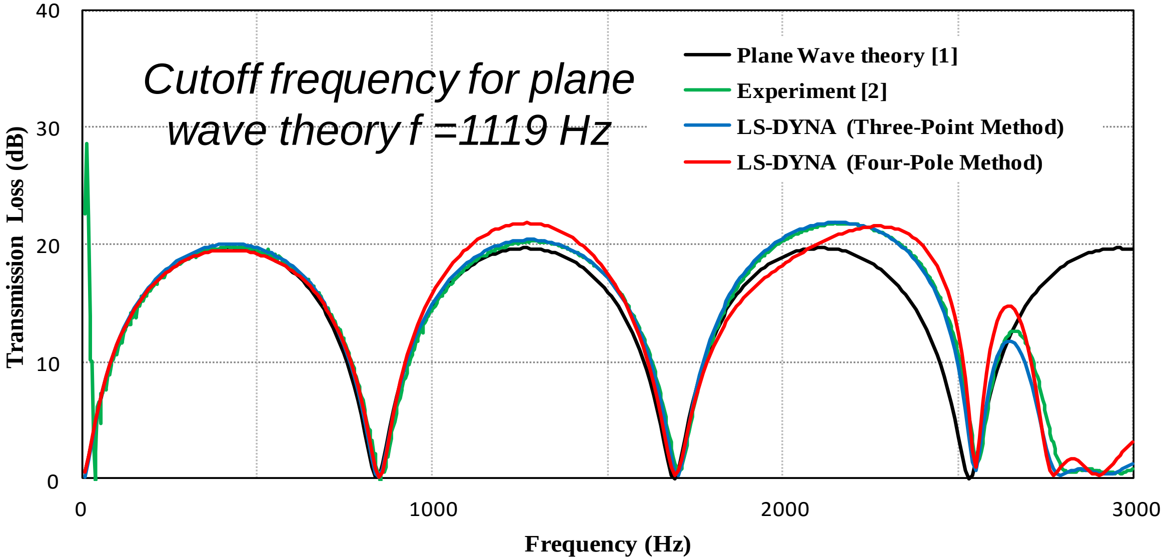
> pfringe

Muffler transmission loss analysis



TL is the difference in the SPL between the incident wave entering and the transmitted wave exiting the muffler when the muffler termination is anechoic (no reflection of sound).

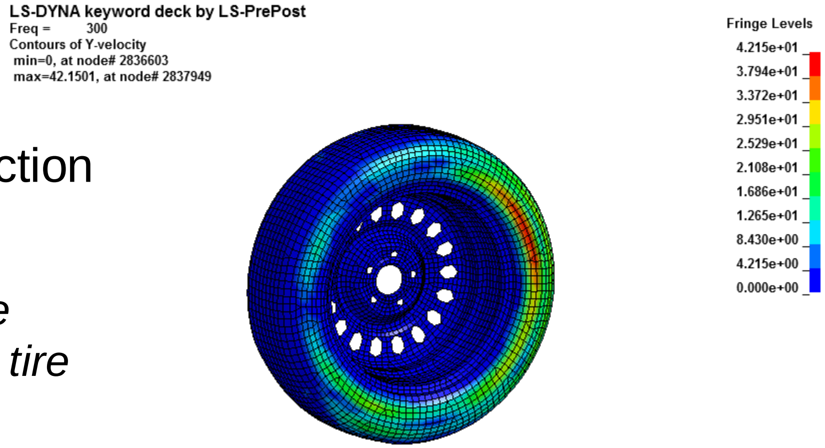
$$TL = 10\log_{10}\frac{W_i}{W_t}$$


Cutoff frequency for plane wave theory f =1119 Hz

Akrapovič d.d.: Marko Krebelj, “Transmission loss simulation of acoustic elements in LS-DYNA”, *9th European LS-DYNA Users’ Conference*, Manchester, UK, June, 2013.

Tire noise

Tire noise is one of main sources in automotive noise, especially pass-by noise.



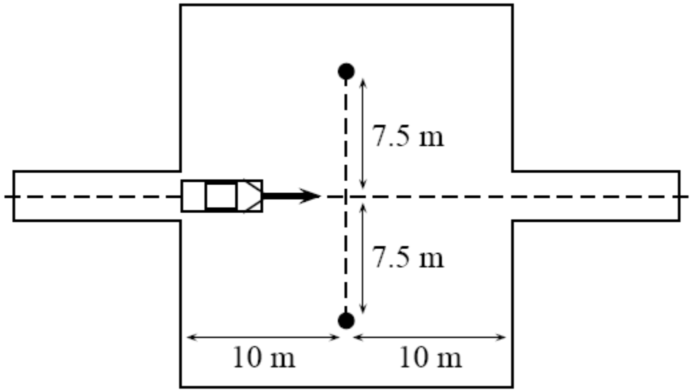
↑

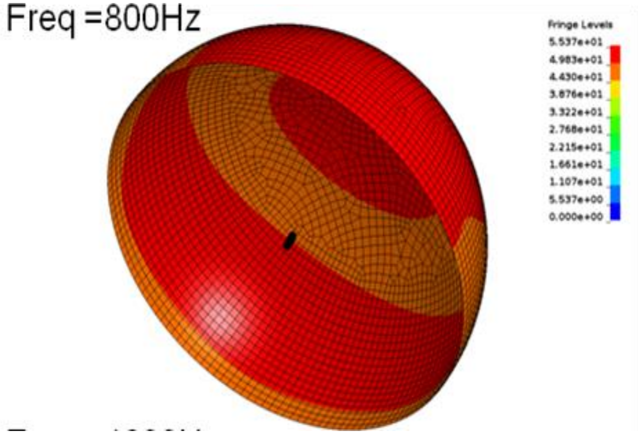
Unit force load

A numerical model of LS-DYNA for the prediction of tire noise radiation:

- *SSD describes the dynamic behavior of the tire*
- *BEM computes the amount of tire noise due to tire vibration*

The setup for a pass-by noise test from the ISO 362 Standard



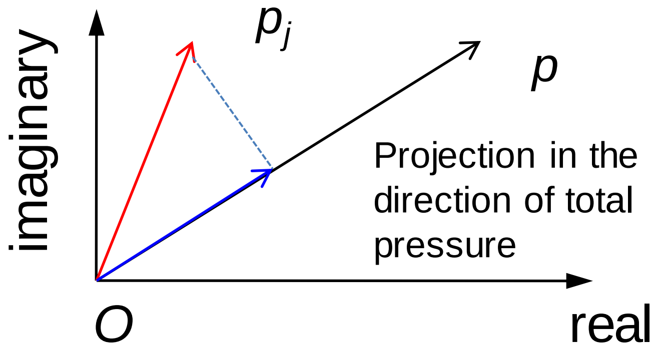


Zhe Cui, Yun Huang, “Sound Radiation Analysis of a Tire with LS-DYNA”, *13th International LS-DYNA Users Conference*, Detroit, June, 2014.

Acoustic panel contribution analysis

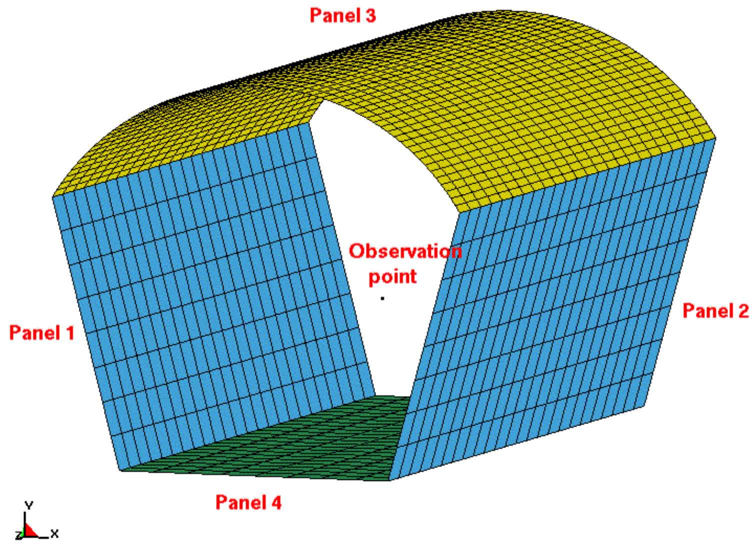
$$p(P)=\sum_{j=1}^{N}\int_{\Gamma_j}\left(G\frac{\partial p}{\partial n}-p\frac{\partial G}{\partial n}\right)d\,\Gamma_j$$

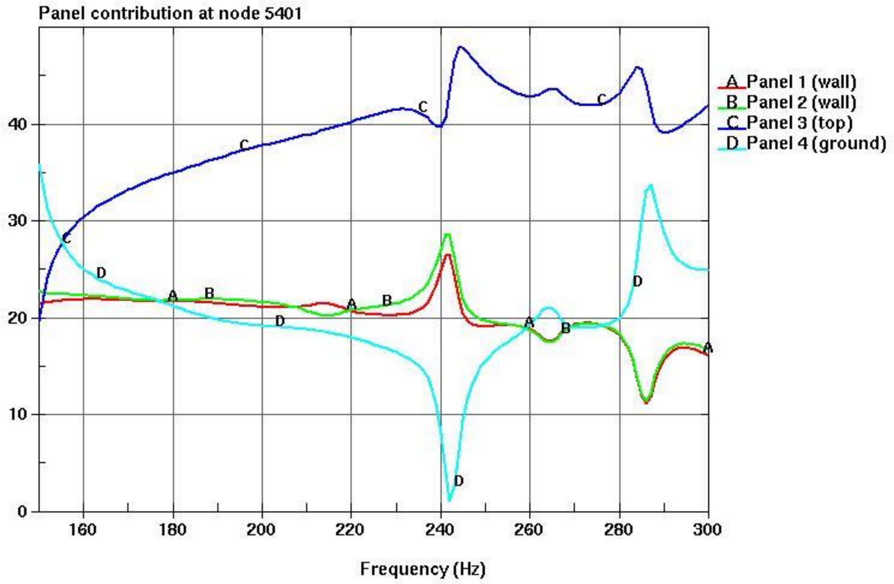
$$=\sum_{j=1}^{N}p_j(P)$$



Projection in the direction of total pressure

A simplified tunnel model



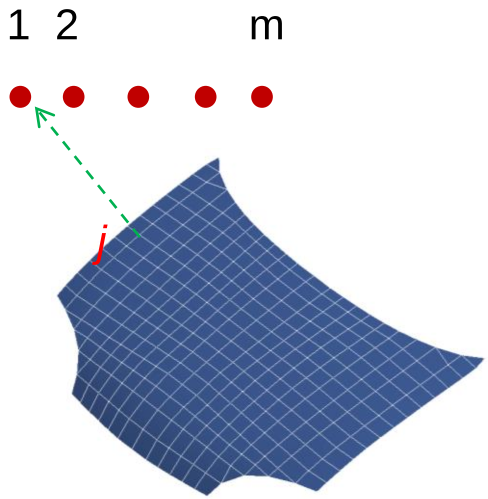


ATV and MATV to accelerate acoustic computation

- Acoustic Transfer Vector can be obtained by including the option **ATV** in the keyword.
- It calculates acoustic pressure (and sound pressure level) at field points due to **unit normal velocity** of each surface node.
- ATV is dependent on structure model, properties of acoustic fluid as well as location of field points.
- An important extension is **modal acoustic transfer vector**, which is based on excitation of the structure by modal shapes. Then the acoustic response for any frequency domain excitation can be obtained by superposition of a series of modal acoustic transfer vectors.

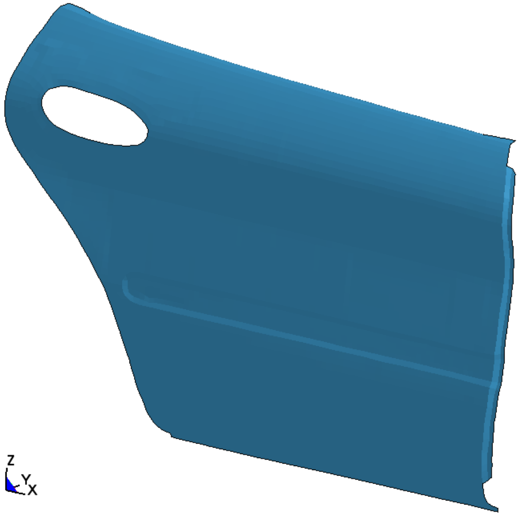
$$\{P\}_m = [ATV]_{m\times n}\{V\}_n$$

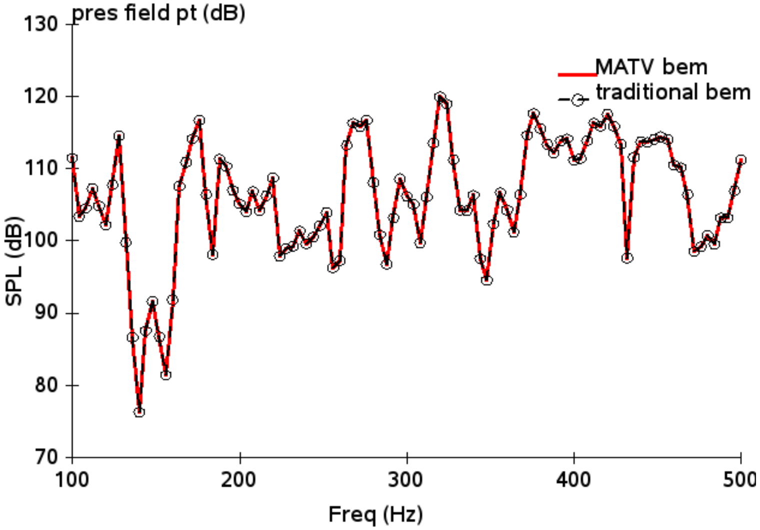
$$\{P\}_m = [MATV]_{m\times l}\{q\}_l$$



Example: using MATV for an auto door model

A simplified door model is used for MATV. It is fixed at the upper edge and the 4 holes near the lower edge. It is subjected to 10 loading cases. For each of the loading case, a nodal force spectrum is given at one node on the door.

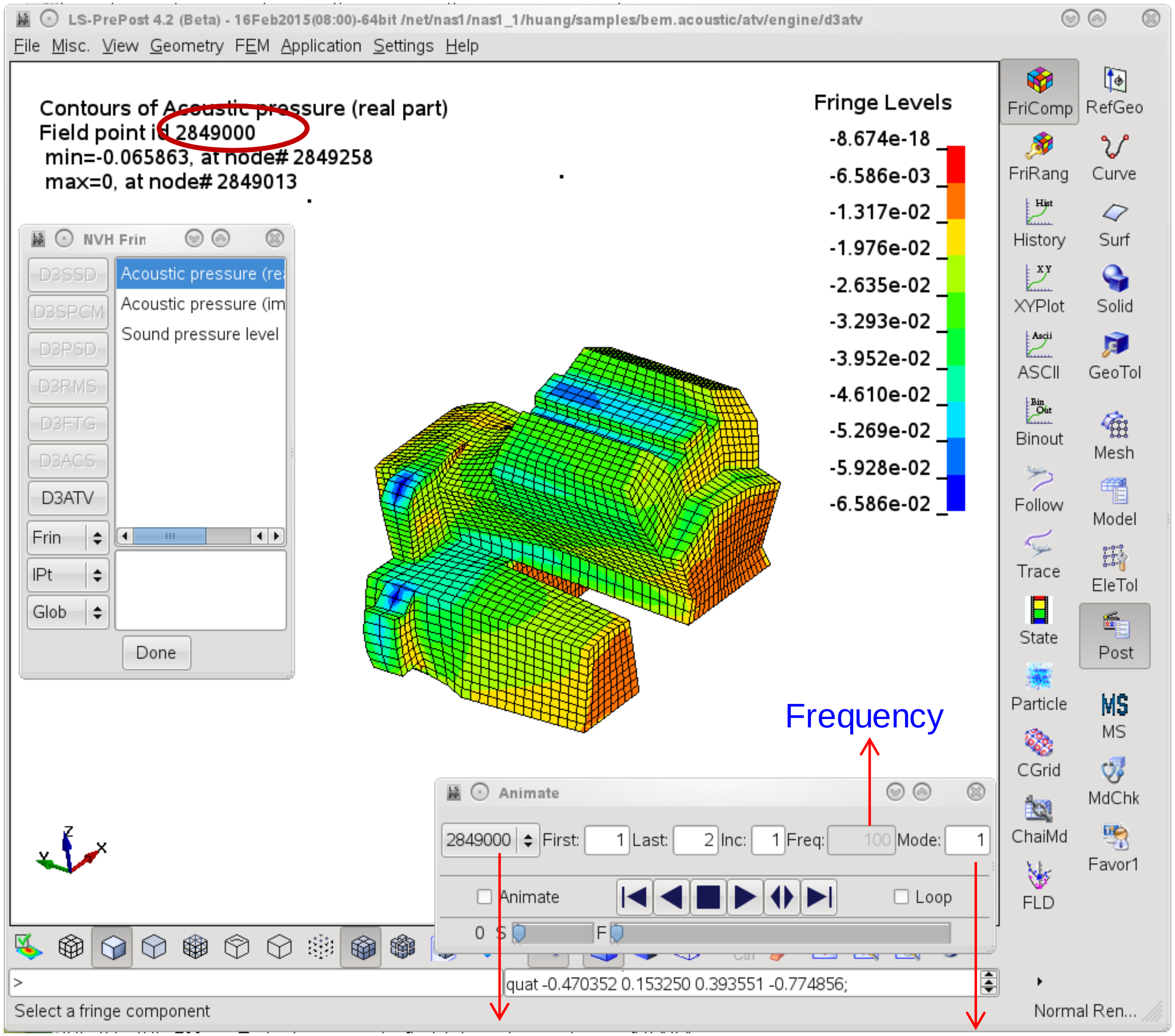




Cases	traditional BEM	MATV BEM
1 loading case	2 h 39 m 50 s	4 h 40 m 56 s
10 loading cases	26 h 38 m 18 s	4 h 41 m 10 s

Intel Xeron CPU E5504 @2.00 GHz (CPU MHz: 1596.00 cache size 4096 KB)

D3ATV


LS-PrePost 4.2 (Beta) - 16Feb2015(08:00)-64bit /net/nas1/nas1_1/huang/samples/bem.acoustic/atv/engine/d3atv

Contours of Acoustic pressure (real part)
Field point id 2849000
min=-0.065863, at node# 2849258
max=0, at node# 2849013

Fringe Levels

-8.674e-18
-6.586e-03
-1.317e-02
-1.976e-02
-2.635e-02
-3.293e-02
-3.952e-02
-4.610e-02
-5.269e-02
-5.928e-02
-6.586e-02

2849000 First: 1 Last: 2 Inc: 1 Freq: 100 Mode: 1

quat -0.470352 0.153250 0.393551 -0.774856;

Select a fringe component

Frequency

Field Point IDs

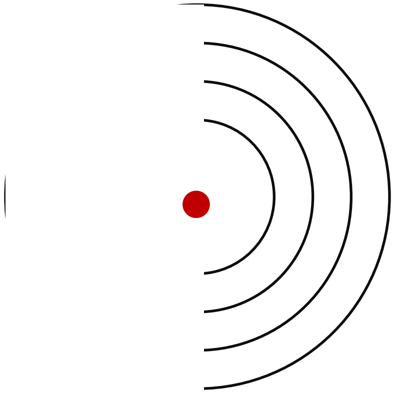
State Numbers

Incident acoustic wave

*FREQUENCY_DOMAIN_ACOUSTIC_INCIDENT_WAVE

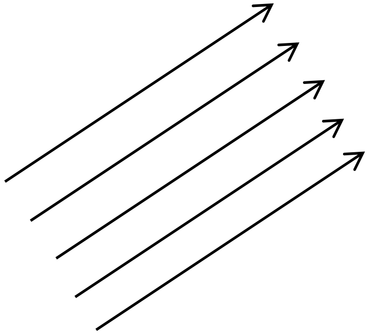
Card 1	1	2	3	4	5	6	7	8
Variable	TYP	MAG	XC	YC	ZC			
Type	I	F	F	F	F			
Default	1	None	None	None	None			

VARIABLE	DESCRIPTION
MAG	Magnitude of the incident sound wave GT.0: constant magnitude, LT.0: |MAG| is a curve ID, which defines the frequency dependent magnitude. See *DEFINE_CURVE.



Spherical wave

$$p^i = A\frac{e^{-ikR}}{R}$$



Plane wave

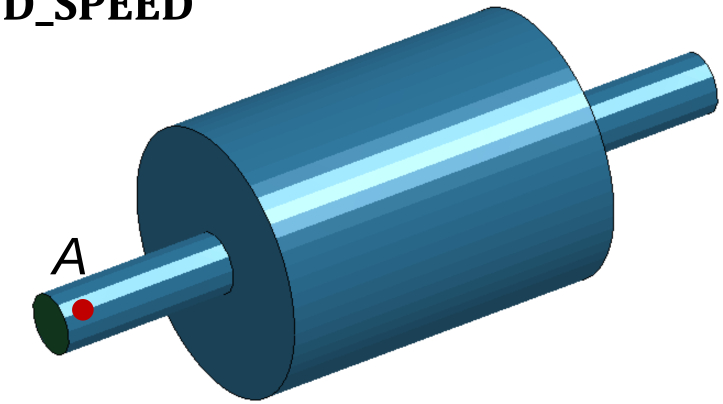
$$p^i = Ae^{-ik(\alpha x+\beta y+\gamma z)}$$

Complex sound speed

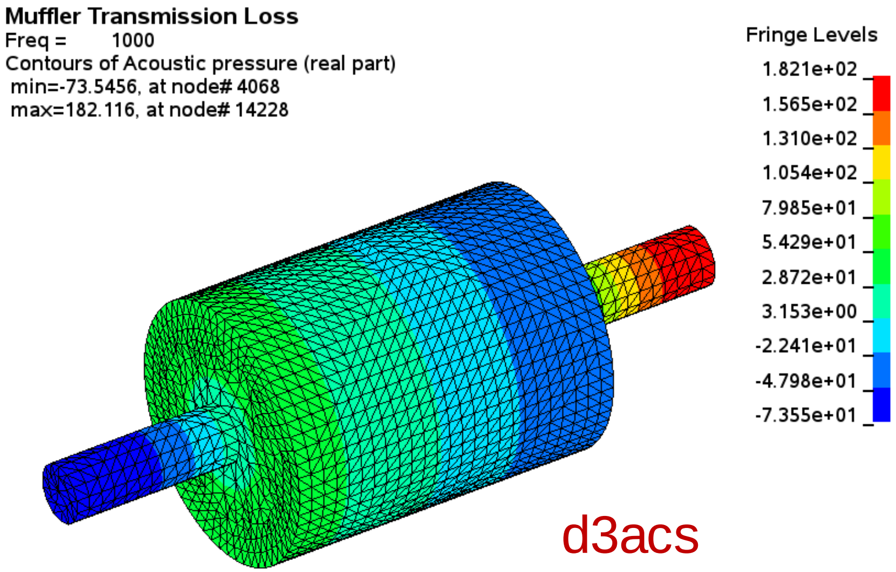
*FREQUENCY_DOMAIN_ACOUSTIC_SOUND_SPEED

Card 1	1
Variable	ID
Type	I
Default	None

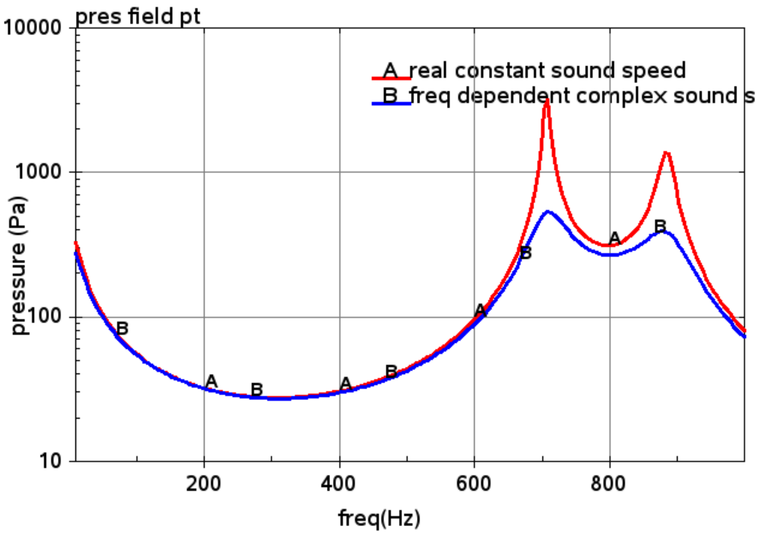
Card 2	1	2
Variable	LCID1	LCID2
Type	I	I
Default	None	None



Due to the damping in the acoustic system, the acoustic pressure at field point A is reduced, comparing to the case without damping.



d3acs



Acoustic fringe plot

*FREQUENCY_DOMAIN_ACOUSTIC_FRINGE_PLOT_*{OPTION}**

Purpose:

Define field points for acoustic pressure computation and use D3ACS binary database to visualize the pressure distribution.

Options:

PART

PART_SET

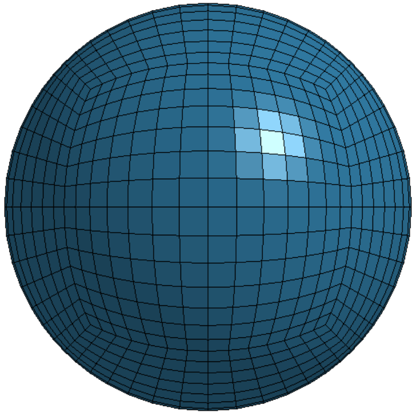
NODE_SET

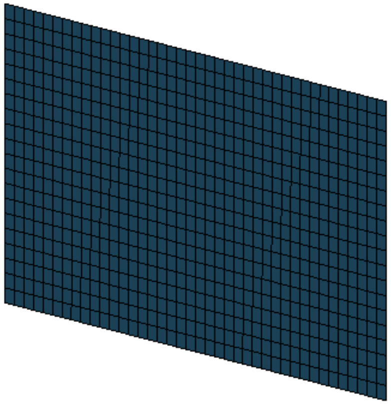
Existing structure components

SPHERE

PLATE

LS-DYNA generates mesh automatically



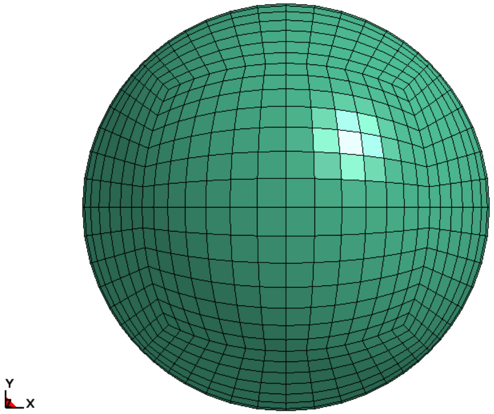


Results (D3ACS):

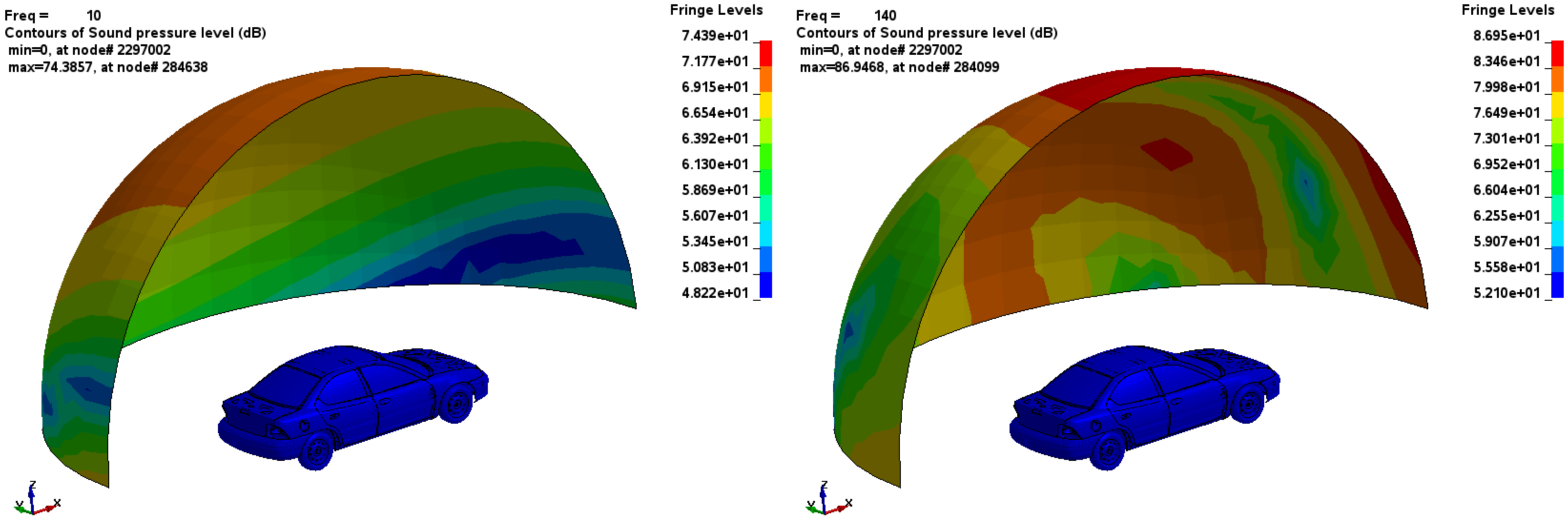
- Real part of acoustic pressure
- Imaginary part of acoustic pressure
- Absolute value of acoustic pressure
- Sound Pressure Level (dB)
- ✓ Supported by LS-PrePost 4.2 and above

FREQUENCY_DOMAIN_ACOUSTIC_FRINGE_PLOT_*{SPHERE}

R = 5000 mm
DENSITY = 15
No. of new nodes: 1178
No. of new elements: 1176



Radiated noise from vehicle



3.3) FEM acoustic solver

Background

1) FEM acoustics is an alternative method for simulating acoustics. It helps predict and improve sound and noise performance of various systems. The FEM simulates the entire propagation volume -- being air or water.
2) Compute acoustic pressure and SPL (sound pressure level)
3) Output binary database: d3acs (accessible by LS-PREPOST)
4) Output ASCII database: Press_Pa and Press_dB as xyplot files
5) Output frequency range dependent on mesh size
6) Very fast since
   - ✓ One unknown per node
   - ✓ The majority of the matrix is unchanged for all frequencies
   - ✓ Using a fast sparse matrix iterative solver

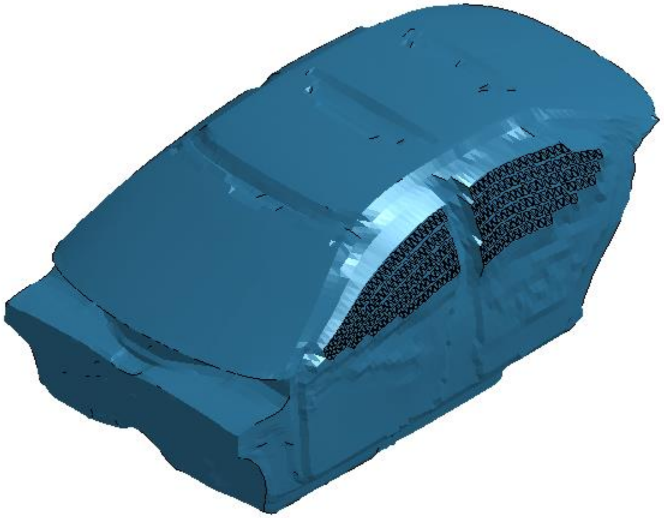
Acoustic analysis on passenger compartment

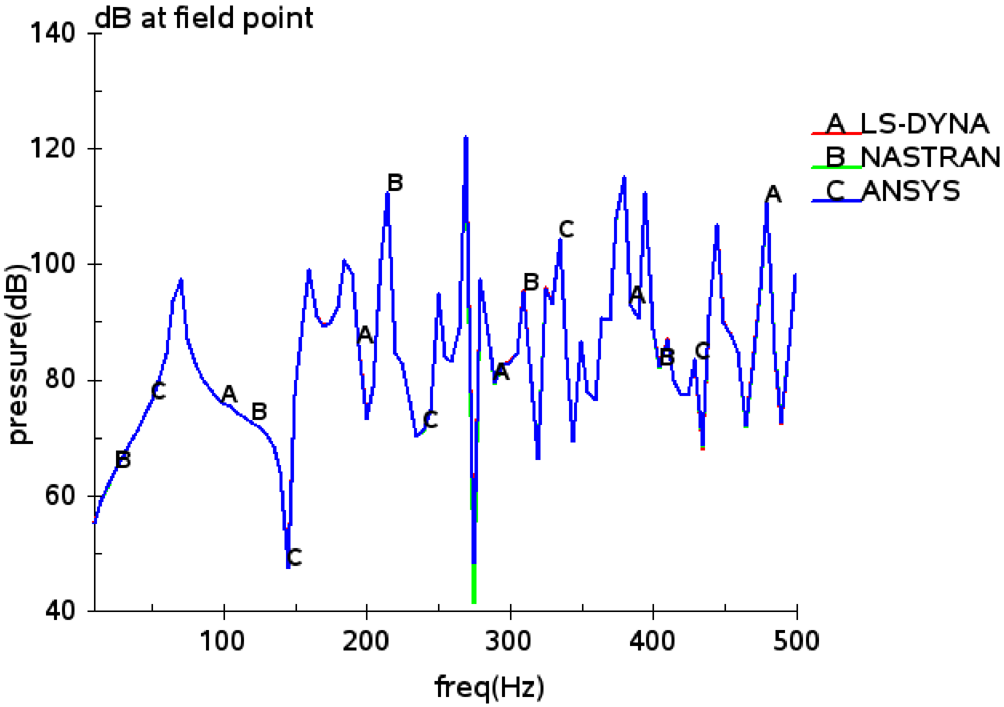
Number of Nodes: 114221
Number of Tetra elements: 643619
Two loading cases:

1. Base excitation + open window
2. Base excitation + impedance

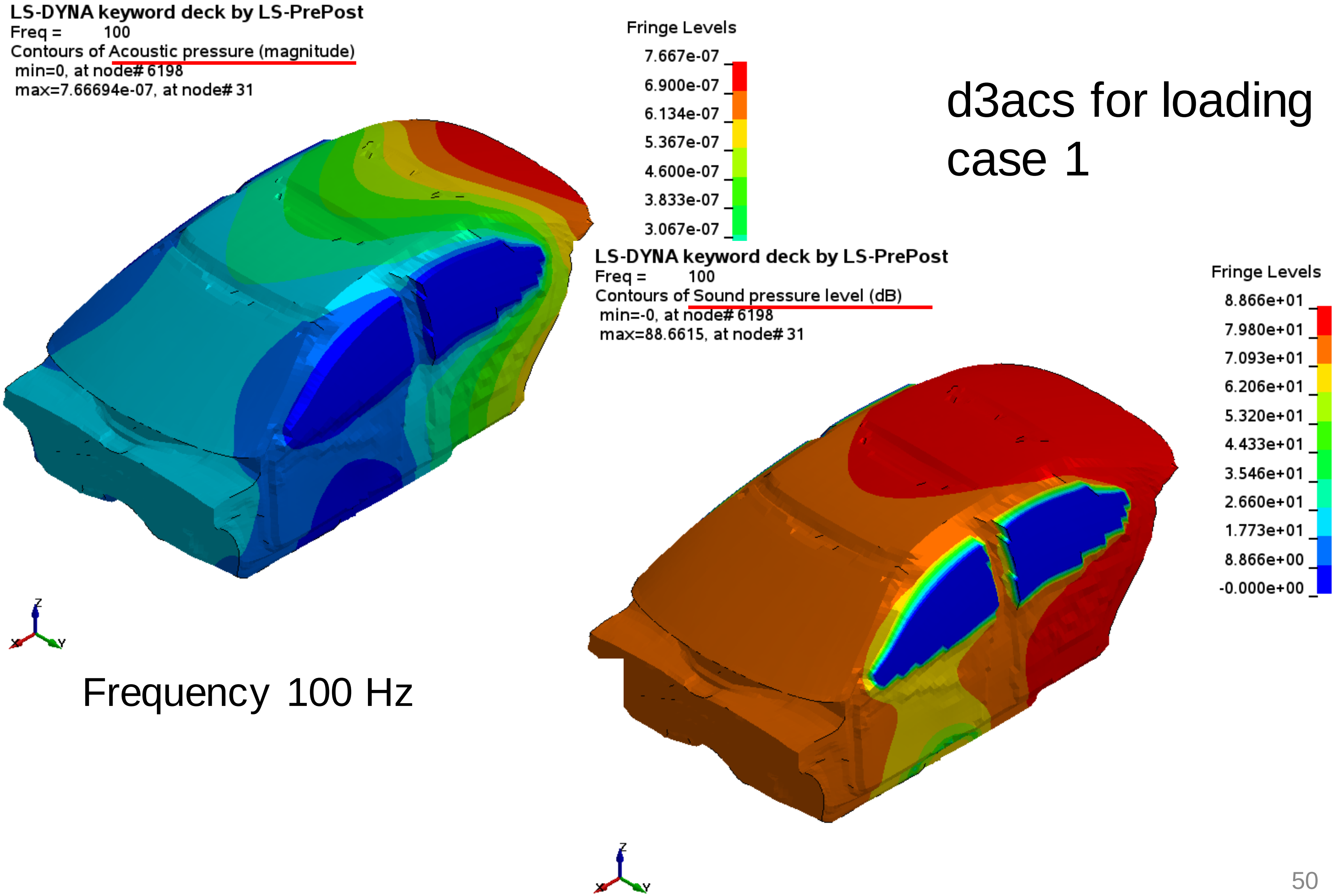
Loading case 1:

- Base excitation 4mm/s for 10-500 Hz.
- Open windows





d3acs for loading case 1



LS-DYNA keyword deck by LS-PrePost
Freq = 100
Contours of Acoustic pressure (magnitude)
min=0, at node# 6198
max=7.66694e-07, at node# 31

LS-DYNA keyword deck by LS-PrePost
Freq = 100
Contours of Sound pressure level (dB)
min=-0, at node# 6198
max=88.6615, at node# 31

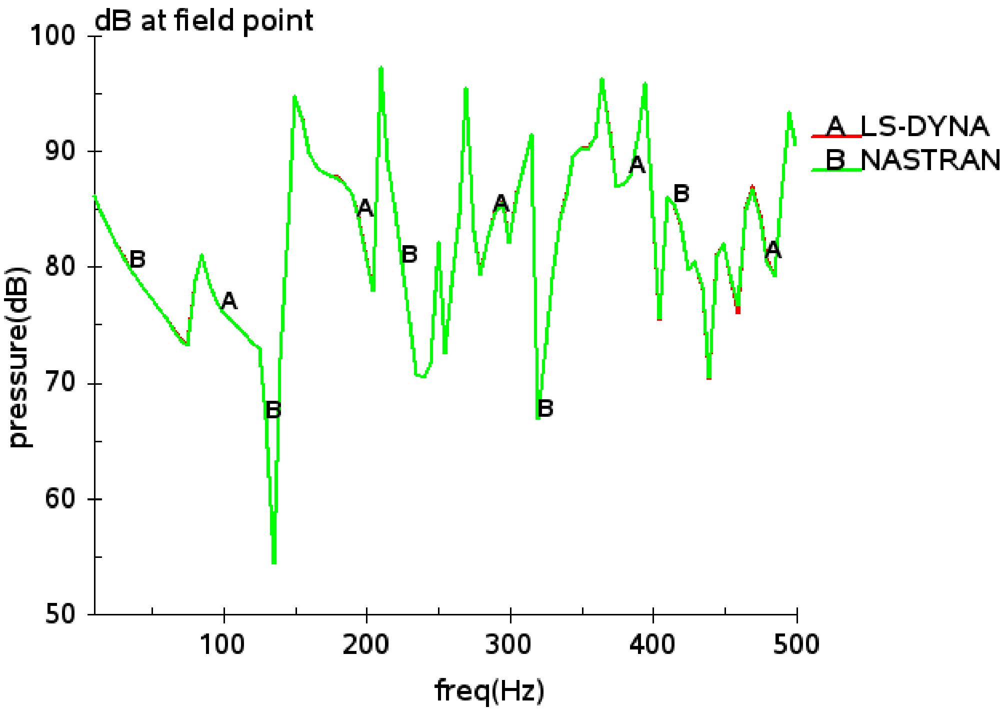
Frequency 100 Hz

Loading case 2:

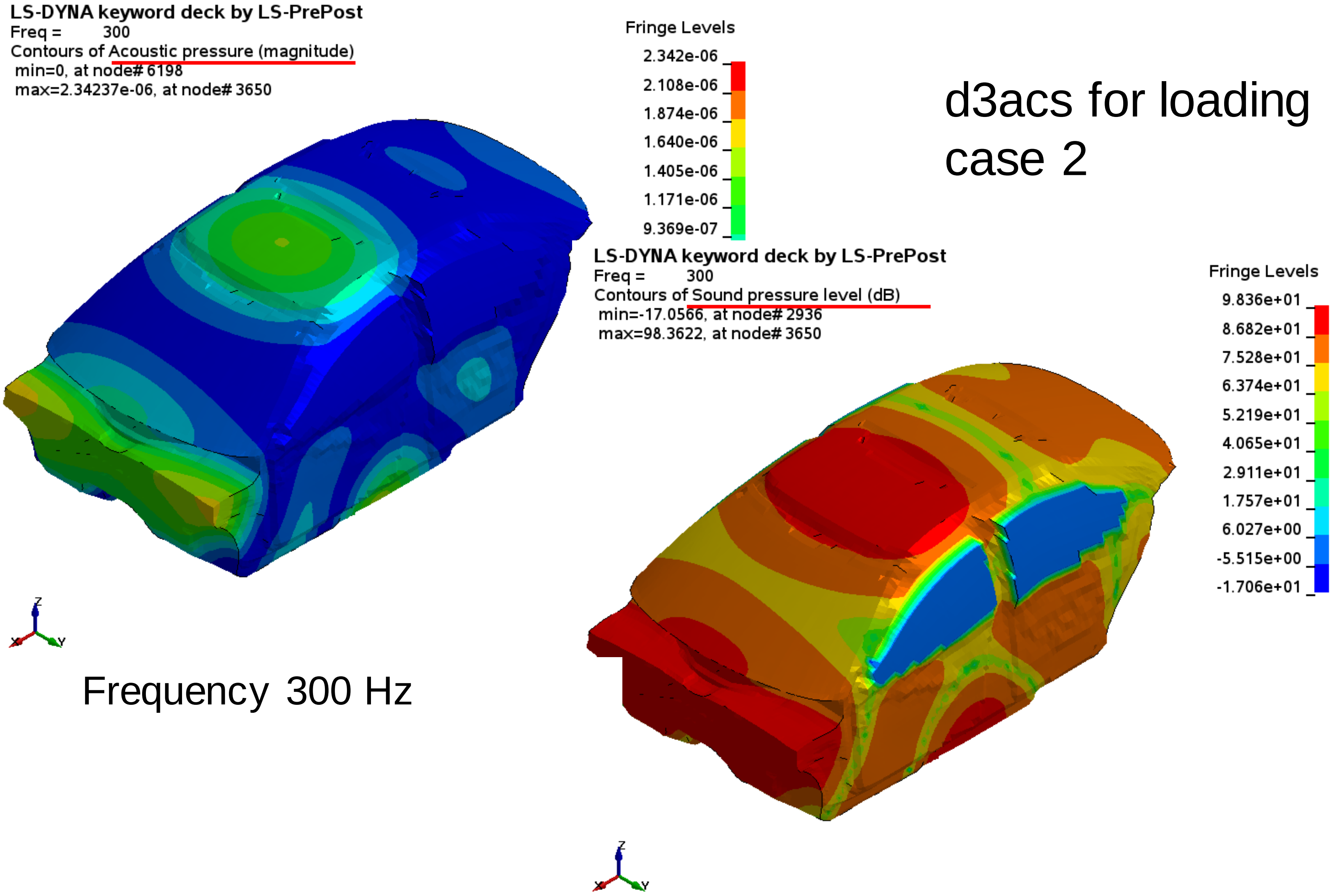
characteristic impedance

- Base excitation 4mm/s for 10-500 Hz.
- Impedance b.c. on the top (to model the sound absorbing material)

$$p / v_n = \rho c$$



d3acs for loading case 2



```
LS-DYNA keyword deck by LS-PrePost
Freq =        300
Contours of Acoustic pressure (magnitude)
 min=0, at node# 6198
 max=2.34237e-06, at node# 3650
```

```
LS-DYNA keyword deck by LS-PrePost
Freq =        300
Contours of Sound pressure level (dB)
 min=-17.0566, at node# 2936
 max=98.3622, at node# 3650
```

Frequency 300 Hz

Acoustic Eigenvalue Analysis

***FREQUENCY_DOMAIN_ACOUSTIC_FEM_EIGENVALUE**

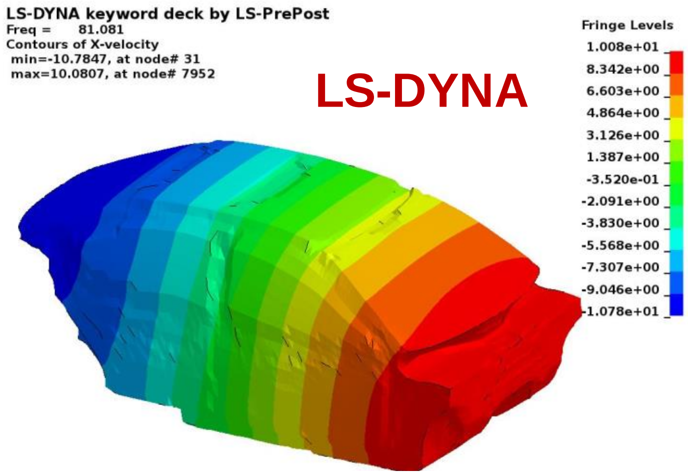
$$\left([K_a] + j\omega[C_a] - \omega^2[M_a]\right)\{p_i\} = \{F_a\}$$

$$[K_a]\{\phi_i\} = \omega_i^2[M_a]\{\phi_i\} \quad i = 1, \ldots N_a$$

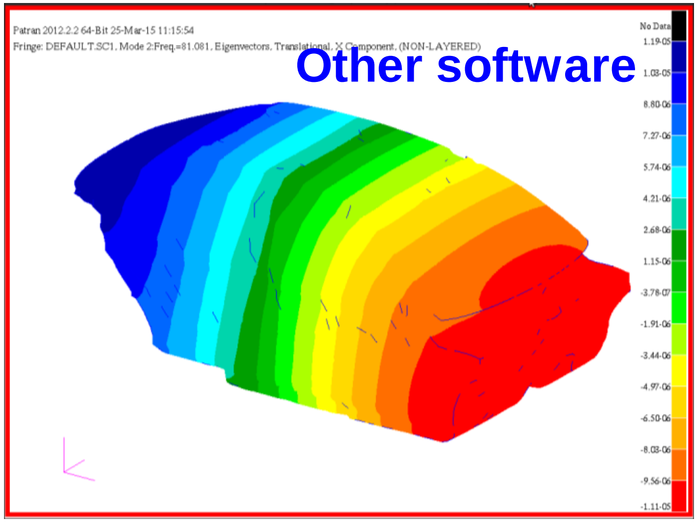
New databases:
- **EIGOUT_AC**
- **D3EIGV_AC**

A closed compartment model

Mode	LS-DYNA	Other software
1	3.93144E-06	6.698696E-06
2	8.10808E+01	8.108078E+01
3	1.25457E+02	1.254568E+02
4	1.47780E+02	1.477799E+02
5	1.52190E+02	1.521901E+02
6	1.72872E+02	1.728723E+02
7	1.98448E+02	1.984481E+02
8	2.08590E+02	2.085895E+02
9	2.14581E+02	2.145808E+02
10	2.23230E+02	2.232297E+02

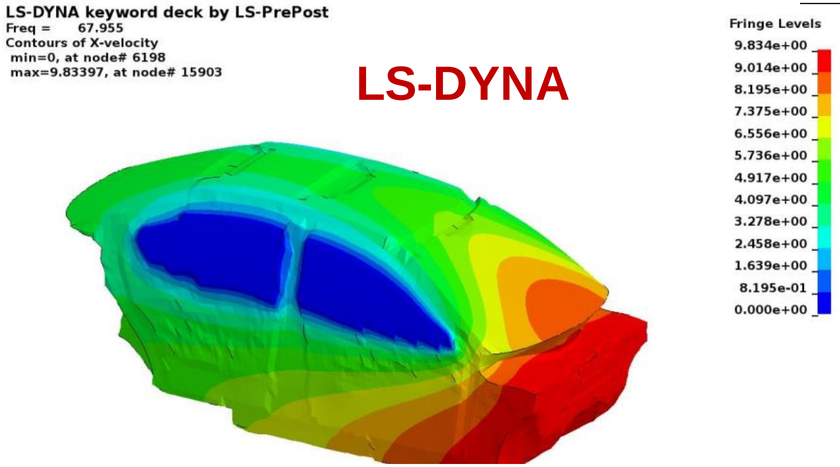

LS-DYNA keyword deck by LS-PrePost
Freq = 81.081
Contours of X-velocity
min=-10.7847, at node# 31
max=10.0807, at node# 7952

Eigenvector for the 2^{nd} mode


Patran 2012.2.2 64-Bit 25-Mar-15 11:15:54
Fringe: DEFAULT.SC1, Mode 2:Freq.=81.081, Eigenvectors, Translational, X Component, (NON-LAYERED)

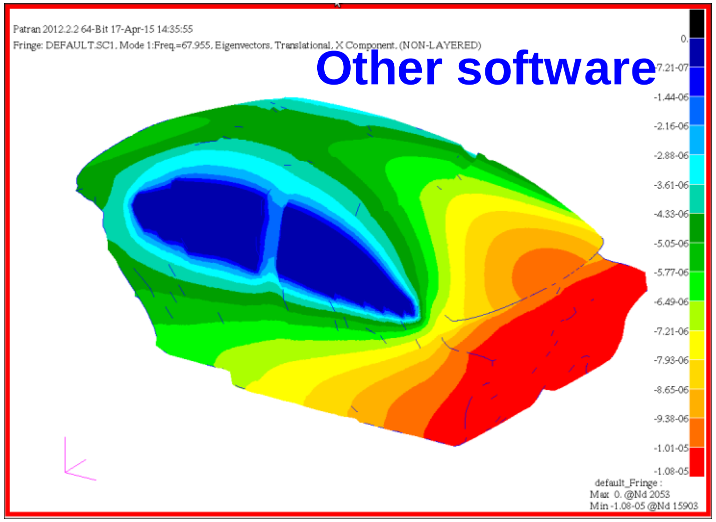
A compartment model with windows open

Mode	**LS-DYNA**	**Other software**
1	6.79551E+01	6.795506E+01
2	1.03346E+02	1.033460E+02
3	1.47853E+02	1.478530E+02
4	1.58211E+02	1.582106E+02
5	1.74781E+02	1.747807E+02
6	1.87460E+02	1.874595E+02
7	2.10906E+02	2.109057E+02
8	2.14993E+02	2.149934E+02
9	2.28861E+02	2.288609E+02
10	2.50667E+02	2.506669E+02



LS-DYNA keyword deck by LS-PrePost
Freq = 67.955
Contours of X-velocity
min=0, at node# 6198
max=9.83397, at node# 15903

LS-DYNA



Patran 2012.2.2 64-Bit 17-Apr-15 14:35:55
Fringe: DEFAULT.SC1, Mode 1:Freq.=67.955, Eigenvectors, Translational, X Component, (NON-LAYERED)

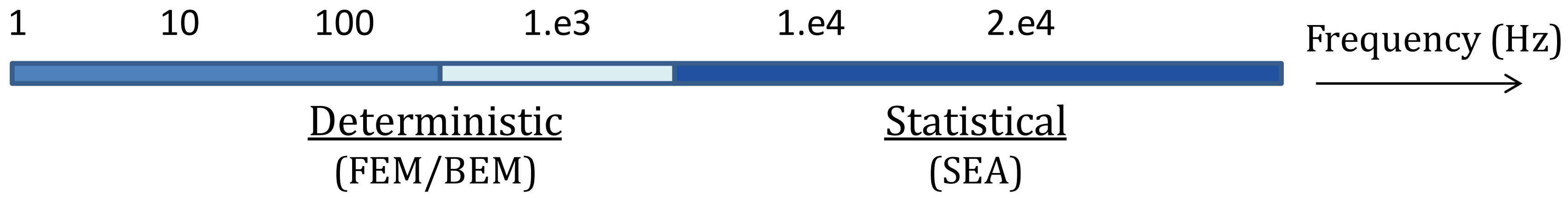
Other software

default_Fringe :
Max 0. @Nd 2053
Min -1.08-05 @Nd 15903

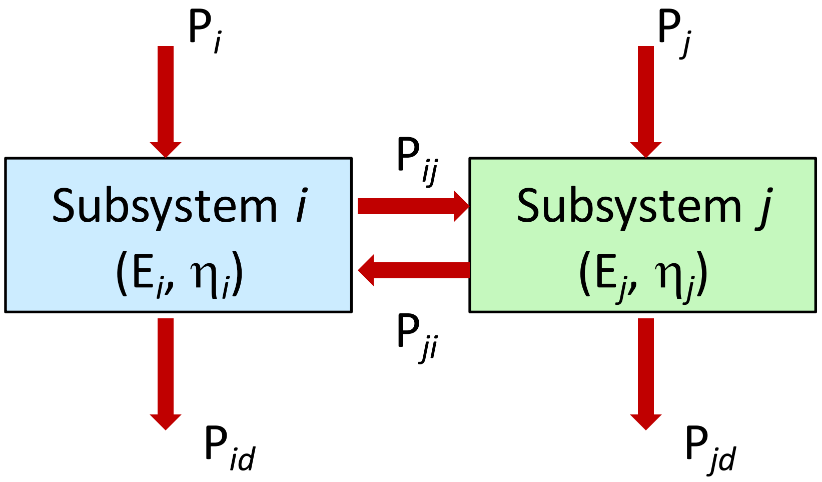
Eigenvector for the 1st mode

3.4) SEA (Statistical Energy Analysis)

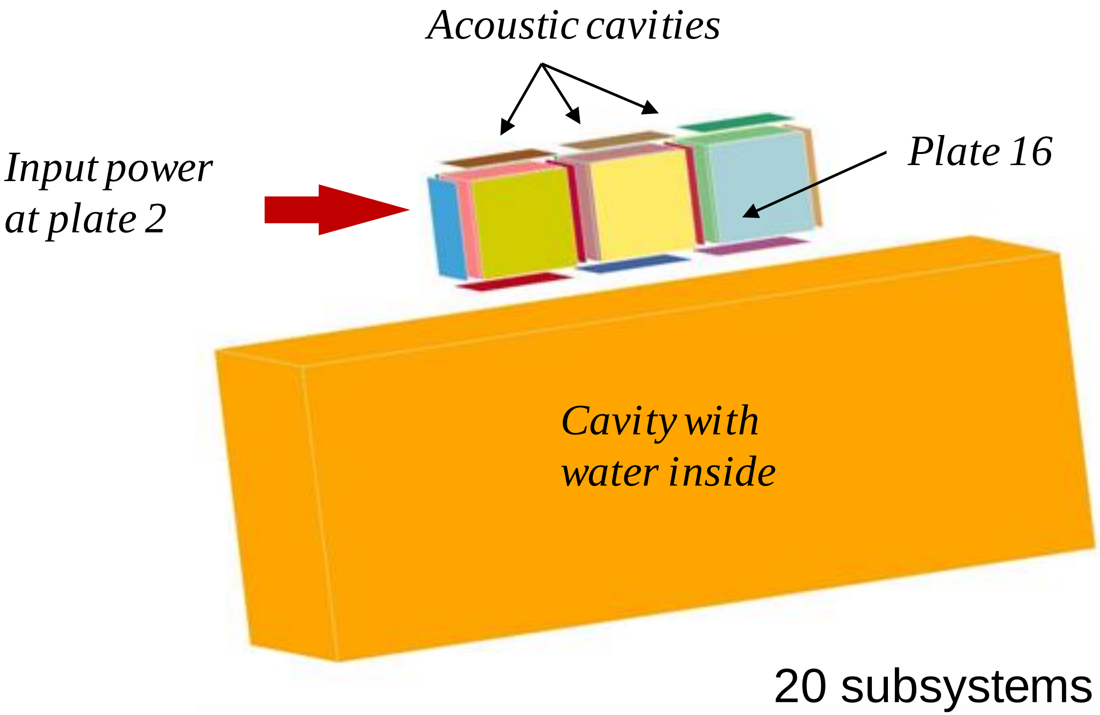
***FREQUENCY_DOMAIN_SEA_SUBSYSTEM**
***FREQUENCY_DOMAIN_SEA_CONNECTION**
***FREQUENCY_DOMAIN_SEA_INPUT_POWER**



- SEA is a statistical method for studying vibration and acoustics in high frequency range, without using elements or mesh.
- In SEA a system is represented in terms of a number of coupled subsystems and a set of linear equations are derived that describe the input, storage, transmission and dissipation of energy within each subsystem.

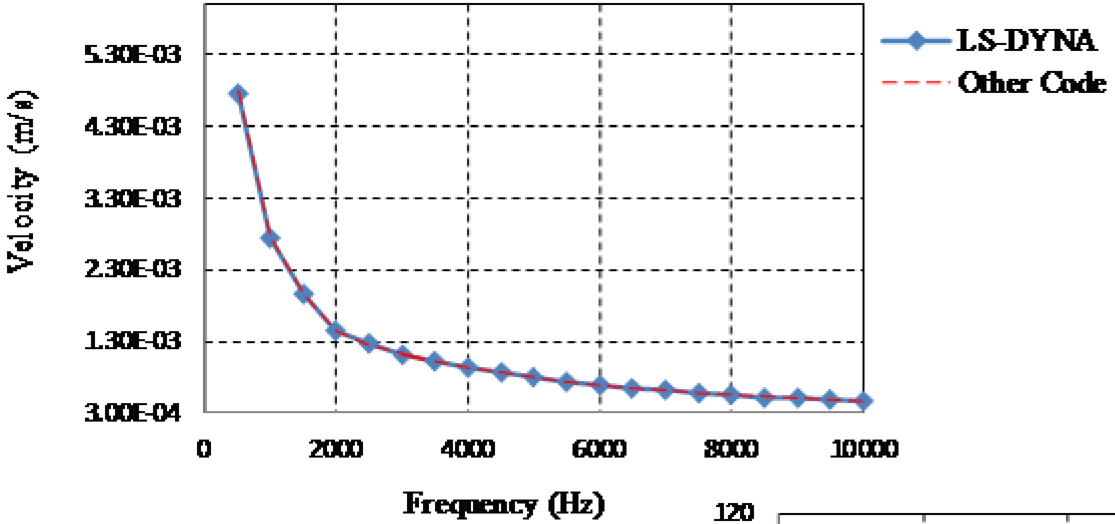


SEA model of 2 subsystems

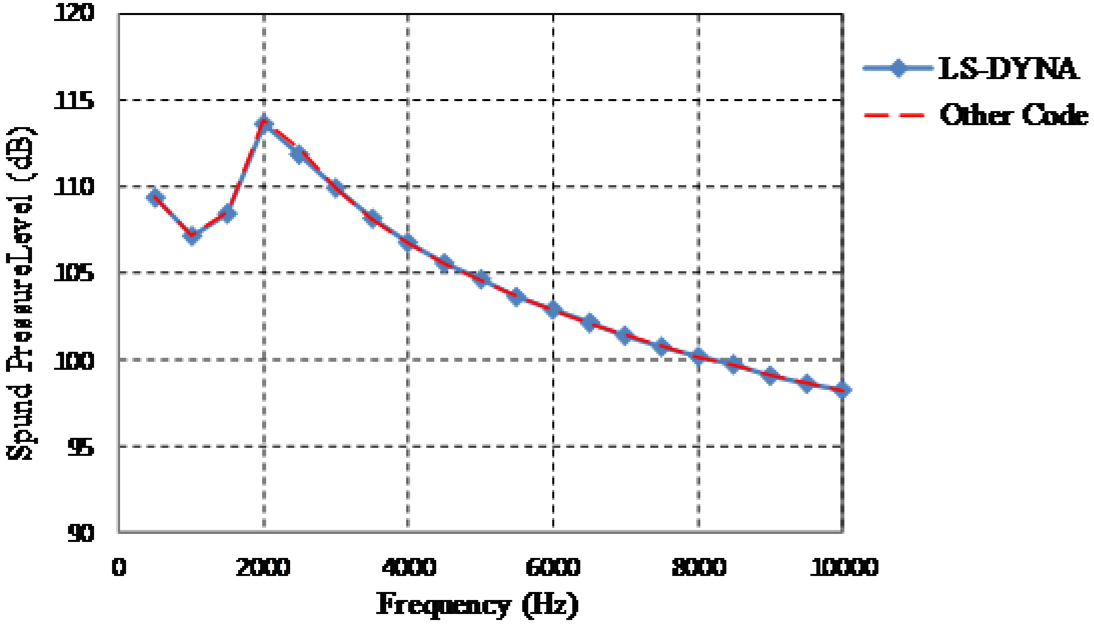


20 subsystems
- 16 steel plates
- 3 acoustic cavities (air)
- 1 acoustic cavity (water)

Courtesy of Numerical Engineering Solutions, Australia



Velocity of plate 16



Sound pressure level of cavity with air (next to plate 2)

- **California Polytechnic State University:** Roger Sharpe, "*A prediction of the acoustic output of a golf driver head using finite elements*", M.S. Thesis, 2010.
- **California Polytechnic State University:** Scott Moreira, "*Predicting the acoustic response of the golf club & ball impact using finite elements and the boundary element method*", M.S. Thesis, 2011.
- **California Polytechnic State University:** Mase, T., Sharpe, R., Volkoff-Shoemaker, N., and Moreira, S., "*Modeling the Sound of a Golf Club Impact*", Journal of Sports Engineering and Technology, Vol. 226, Issue 2, pp. 107-113, June, 2012.
- **Sheffield Hallam University:** Tom Allen, Jim Gough, David Koncan, David James, Eric Morales, Paul Wood, "*Modeling the acoustics of a golf ball impacting a titanium plate*", Procedia Engineering, 72 (2014) 587-592.
- **Flotrend Corporation, Taiwan:** Leo Chen, "*Noise Analysis of AC devices by LS-DYNA*", Flotrend technique note, 2014.
- **National Taipei University of Technology, Taiwan:** Hsiu-Ying Hwang, Guo-Ding Huang, Zong-Syun Jhang, "*Improvement of Noise and Vibration for a Midsize Electric Bus*", Proceedings of the 31st National Congress of Chinese Mechanical Engineering Society, Taizhong, Taiwan.

4) FATIGUE SOLVERS

- *4.1) Introduction*
- *4.2) SN curve and EN curve*
- *4.3) Frequency domain fatigue analysis*
    - *4.3.1) Fatigue analysis in random vibration*
    - *4.3.2) Fatigue analysis in SSD*
- *4.4) Time domain fatigue analysis*
    - *4.4.1) Stress based approach*
    - *4.4.2) Strain based approach*
- *4.5) Fatigue analysis database: d3ftg*

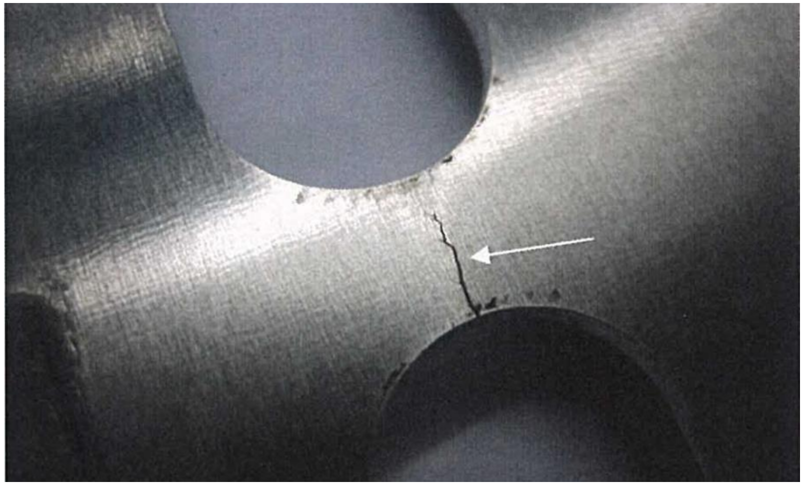
4.1) Introduction

What is fatigue?

- ❑ Fatigue is a process in which damage accumulates due to the repetitive application of loads that may be well below the yield point.
- ❑ Fatigue is a complex process involving many steps but it can be broken down into initiation and propagation of fatigue cracks.
- ❑ It is estimated that fatigue failures are responsible for 90% of all metallic failures.
- ❑ For many years, fatigue has been a significant and challenging problem for engineers, especially for those who design structures such as aircrafts, railroad vehicles, automotives, bridges, pressure vessels, and cranes.

How to run fatigue analysis?

- ❑ Fatigue analysis can be performed in time domain and frequency domain.
- ❑ Two frequency domain approaches based on **random vibration theory and harmonic vibration (SSD) theory** have been implemented in LS-DYNA for fatigue and durability analysis.
- ❑ Recently we implemented time domain fatigue, including **one based on stress** and **the other based on strain** (further testing and validation needed)



4.2) S-N curve and E-N curve

S-N curve *(high cycle, low stress)*

***MAT_ADD_FATIGUE**

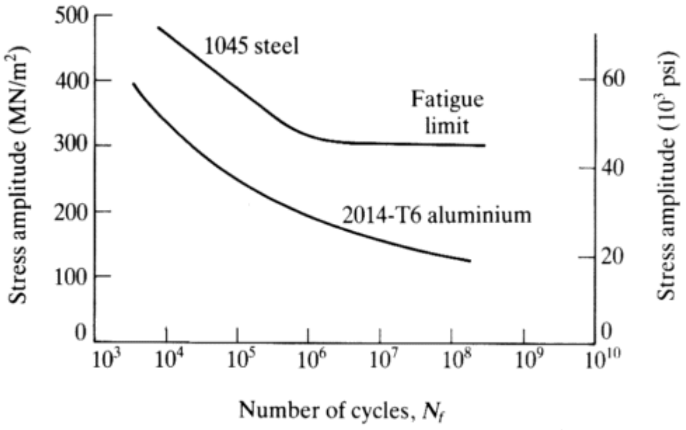
Card 1	1	2	3	4	5	6	7	8
Variable	MID	LCID	LTYPE	A	B	STHRES	SNLIMT	SNTYPE
Type	I	I	I	F	F	F	I	I
Default	none	-1	0	0.0	0.0	none	0	0

- By ***DEFINE_CURVE**
- By equation

$$\boxed{N \cdot S^m = a}$$

$$\boxed{\log(S) = a - b \cdot \log(N)}$$

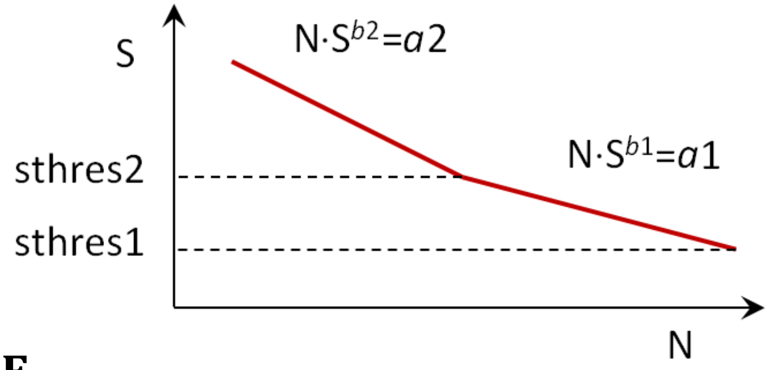
N: number of cycles for fatigue failure
S: stress



Source of information: http://www.efunda.com

- Fatigue life of stress below fatigue threshold

SNLIMT	*Fatigue life for stress lower than the lowest stress on S-N curve.* *EQ.0: use the life at the last point on S-N curve* *EQ.1: extrapolation from the last two points on S-N curve* *EQ.2: infinity.*



*MAT_ADD_FATIGUE

Card 1	1	2	3	4	5	6	7	8
Variable	MID	LCID	LTYPE	A	B	STHRES	SNLIMT	SNTYPE
Type	I	I	I	F	F	F	I	I
Default	none		0	0.0	0.0	none	0	0

Card 2	1	2	3	4	5	6	7	8
Variable				Ai	Bi	STHRESi		
Type				F	F	F		
Default				0.0	0.0	none		

To define S-N curve with multiple slopes, the S-N curve can be split into multiple segments and each segment is defined by a set of parameters Ai, Bi and STHRESi. Up to 8 sets of the parameters (Ai, Bi and STHRESi) can be defined. The lower limit of the *i*-th segment is represented by the threshold stress STHRESi.

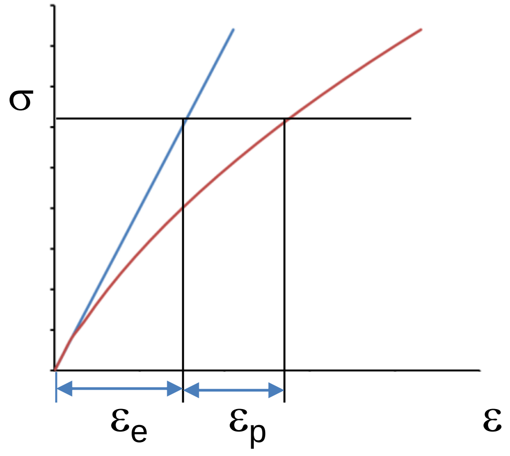
E-N curve *(low cycle, high stress)*

***MAT_ADD_FATIGUE_EN**

Card 1	1	2	3	4	5	6	7	8
Variable	MID	KP	NP	SIGMAP	EPSP	B	C	
Type	I	F	F	F	F	F	F	
Default	none	none	none	none	none	none	none	

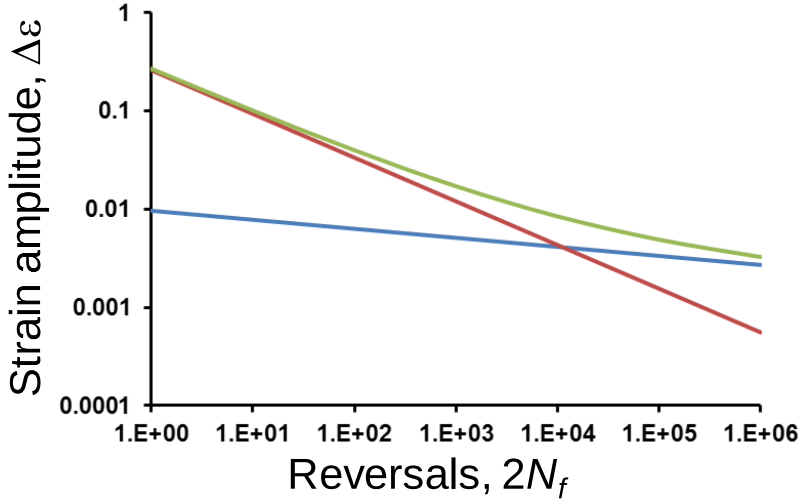
Cyclic stress strain curve

$$\boxed{\varepsilon = \frac{\sigma}{E} + \left(\frac{\sigma}{K'}\right)^{1/n'}}$$



Local strain−life relationship

$$\boxed{\frac{\Delta\varepsilon}{2} = \frac{\sigma'_f}{E}\left(2N_f\right)^b + \varepsilon'_f\left(2N_f\right)^c}$$



4.3) Frequency domain fatigue method

4.3.1) Fatigue analysis in random vibration

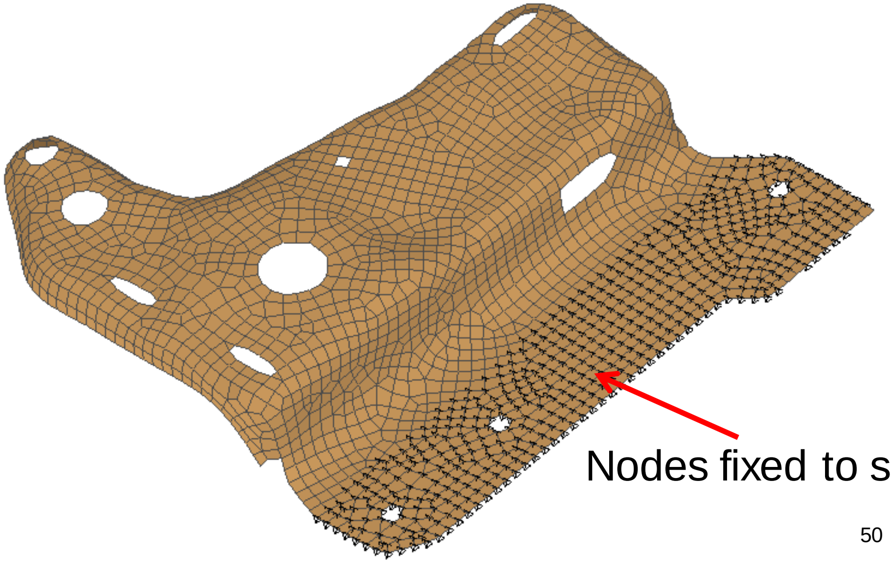
- Keyword ***FREQUENCY_DOMAIN_RANDOM_VIBRATION_*FATIGUE***
- Calculate fatigue life of structures under random vibration
- Based on S-N fatigue curve
- Based on probability distribution & Miner's Rule of Cumulative Damage Ratio

$$R = \sum_i \frac{n_i}{N_i}$$

- Schemes:
  - ✓ *Steinberg's Three-band technique considering the number of stress cycles at the 1σ, 2σ, and 3σ levels.*
  - ✓ *Dirlik method based on the 4 Moments of PSD.*
  - ✓ *Narrow band method*
  - ✓ *Wirsching method*
  - ✓ …

PDF
(*probability density function*)

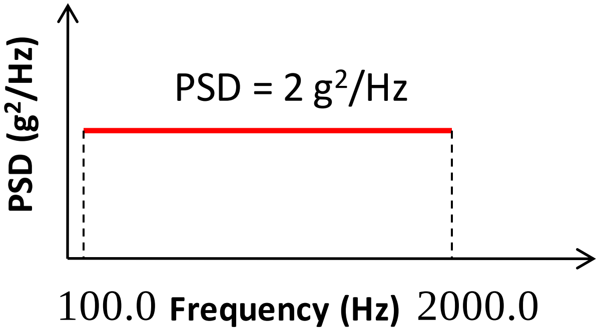
Examples of random vibration fatigue

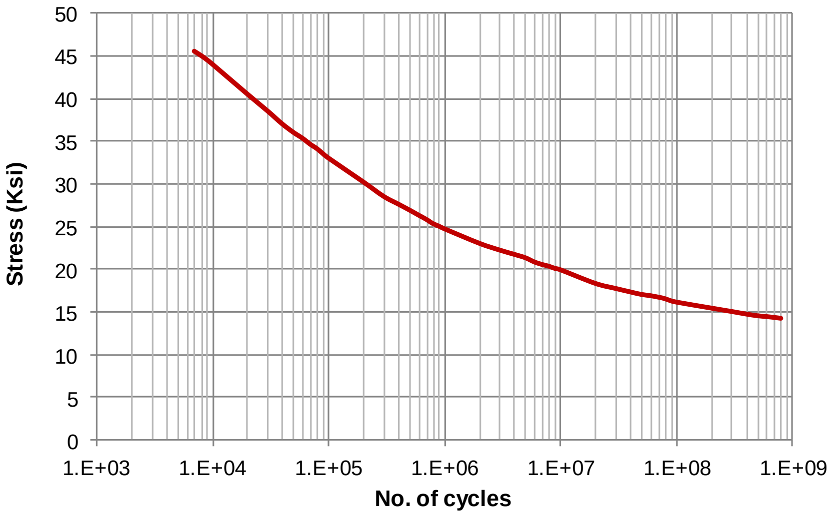

Nodes fixed to shaker table

Aluminum 2014 T6
$\rho = 2800\ \mathrm{kg/m^3}$
$E = 72{,}400\ \mathrm{MPa}$
$\nu = 0.33$

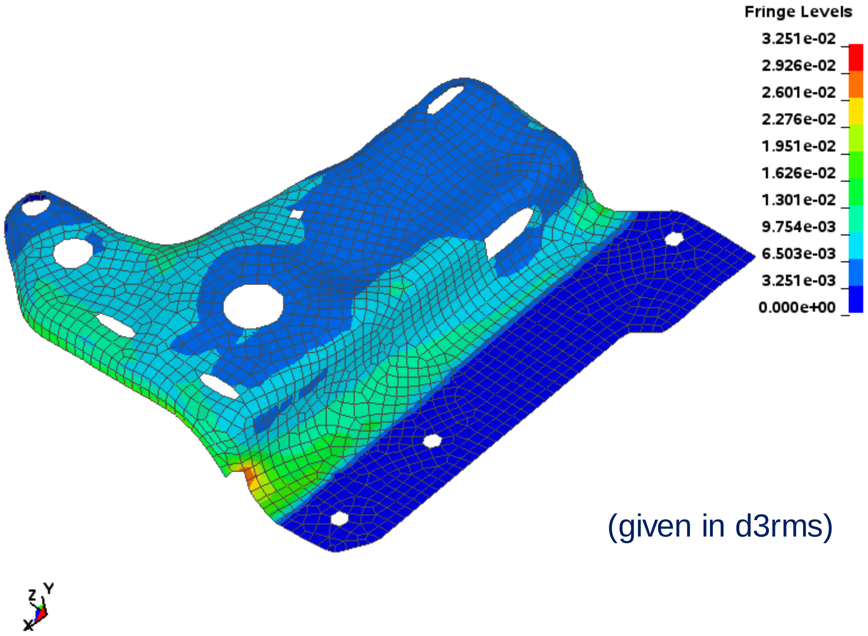
Time of exposure: 4 hours

Z-acceleration PSD



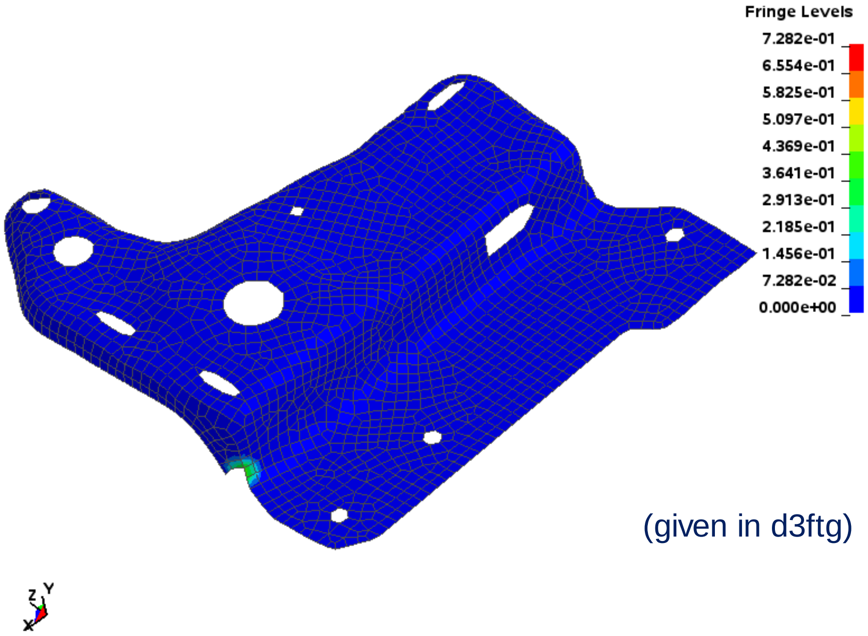


S-N fatigue curve



(given in d3rms)

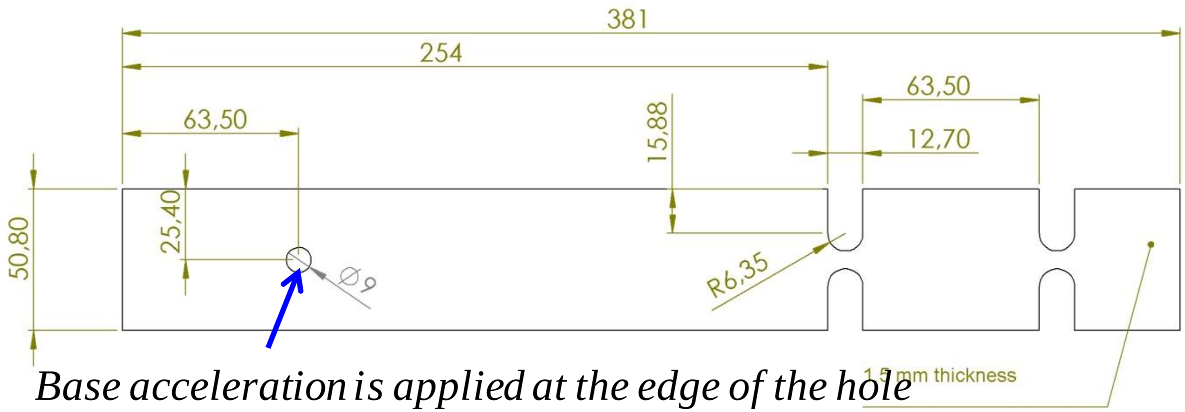
RMS of Von-Mises stress (unit: GPa)



(given in d3ftg)

Accumulative damage ratio

(by Steinberg’s method)



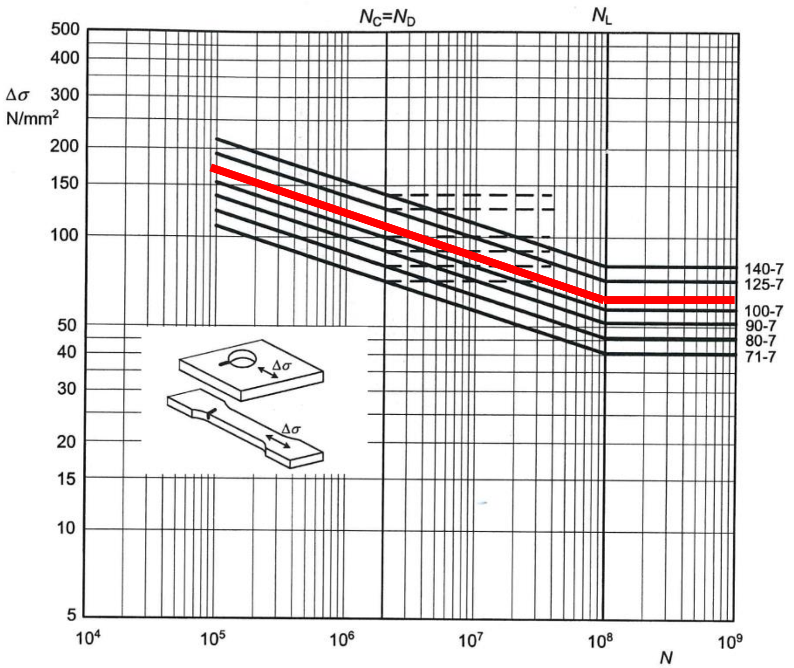
Base acceleration is applied at the edge of the hole

Aluminum alloy 5754

ρ = 2700 kg/m^3

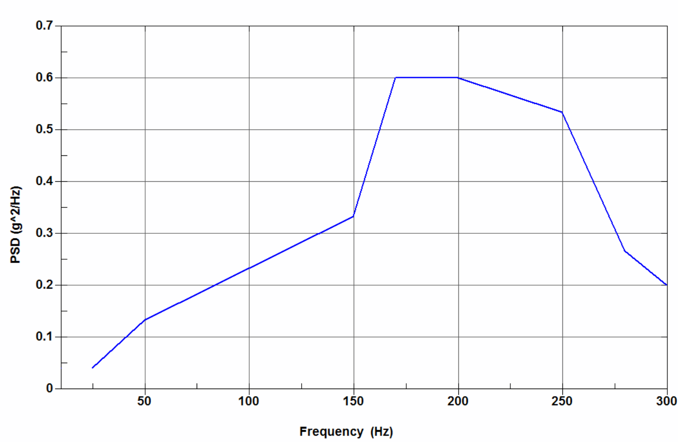
E = 70,000 MPa

ν = 0.33



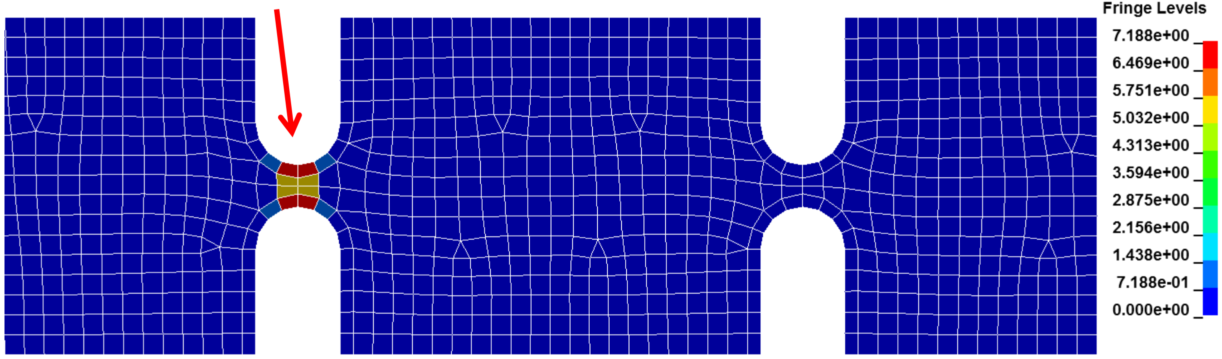
Acceleration PSD

(exposure time: 1800 seconds)



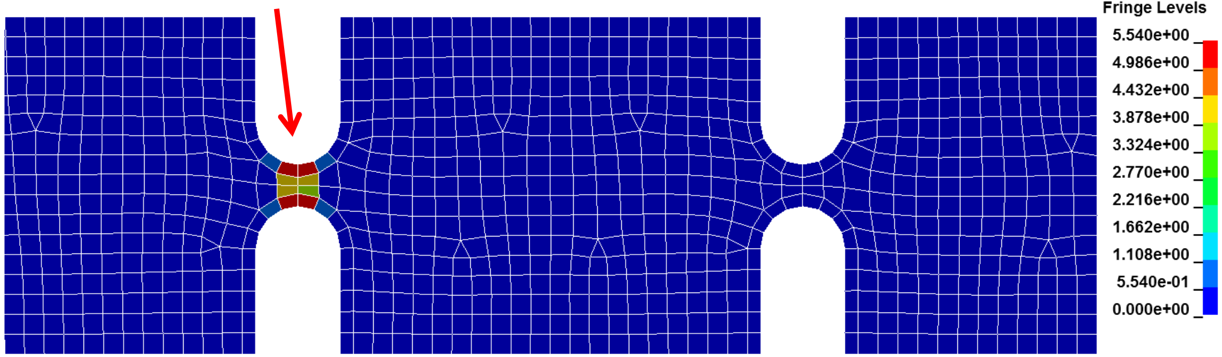
Model courtesy of CIMES France

Damage ratio = 7.188



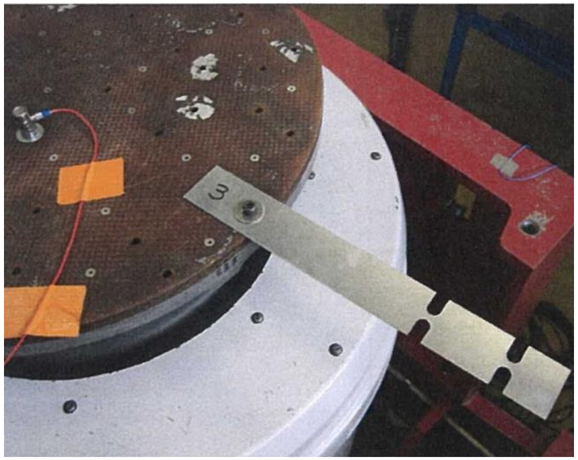
Cumulative damage ratio by Steinberg’s method

Damage ratio = 5.540

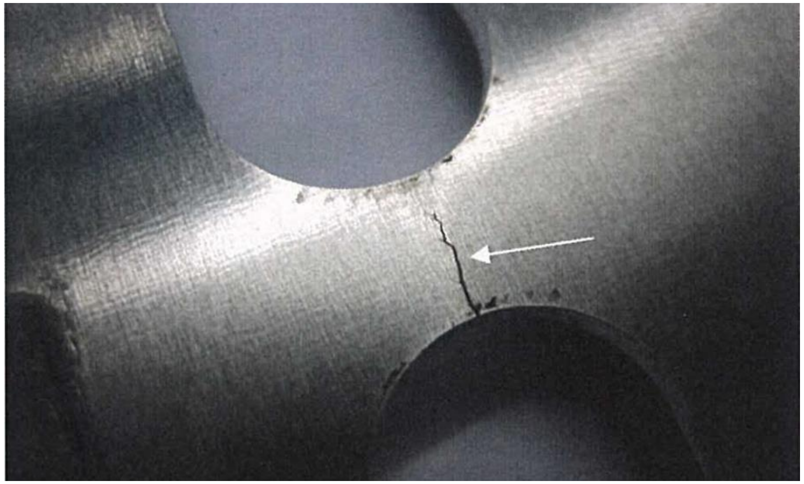


Cumulative damage ratio by Dirlik method

Experiment setup



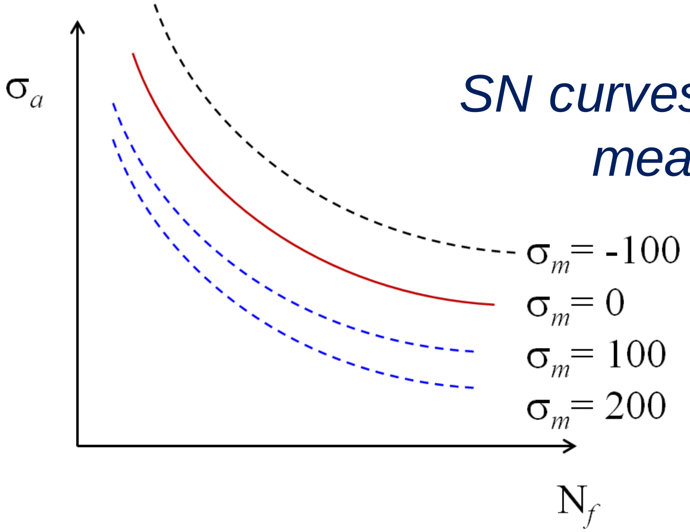
Failure at the notched point in experiment



CIMES France: Ringeval, A., and Huang, Y., "Random Vibration Fatigue Analysis with LS-DYNA", *12th International LS-DYNA Users Conference*, Dearborn, Michigan, June, 2012.

King Saud University: Al-Bahkali Essam, Elkenani Hisham, Souli Mhamed,,“NVH and Random Vibration Fatigue Analysis of a Landing Gear’s Leg for an Un-Manned Aerial Vehicle Using LS-DYNA”, *9th European LS-DYNA Users’ Conference*, Manchester, UK, June, 2013.

Mean stress correction



SN curves with different mean stress

- ☐ Completely reversed tests

$$N \cdot S^m = a$$

- ☐ Mean stress correction equations

Goodman

$$S = \frac{\sigma_a}{1 - \sigma_m / \sigma_u}$$

Soderberg

$$S = \frac{\sigma_a}{1 - \sigma_m / \sigma_y}$$

Gerber

$$S = \frac{\sigma_a}{1 - (\sigma_m / \sigma_u)^2}$$

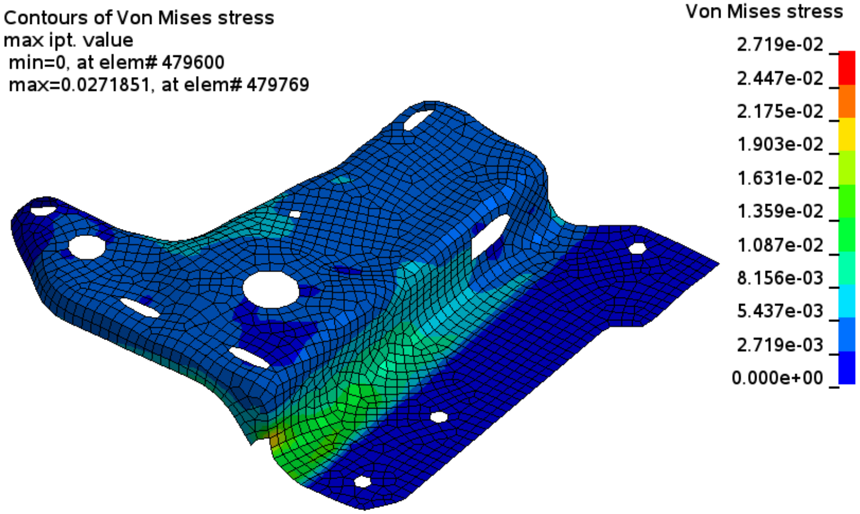
S = Fatigue strength for N cycles under zero mean stress
σ_a = Fatigue strength for N cycles under mean stress σ_m
σ_u = Ultimate tensile strength
σ_y = Yield strength

*FATIGUE_MEAN_STRESS_CORRECTION

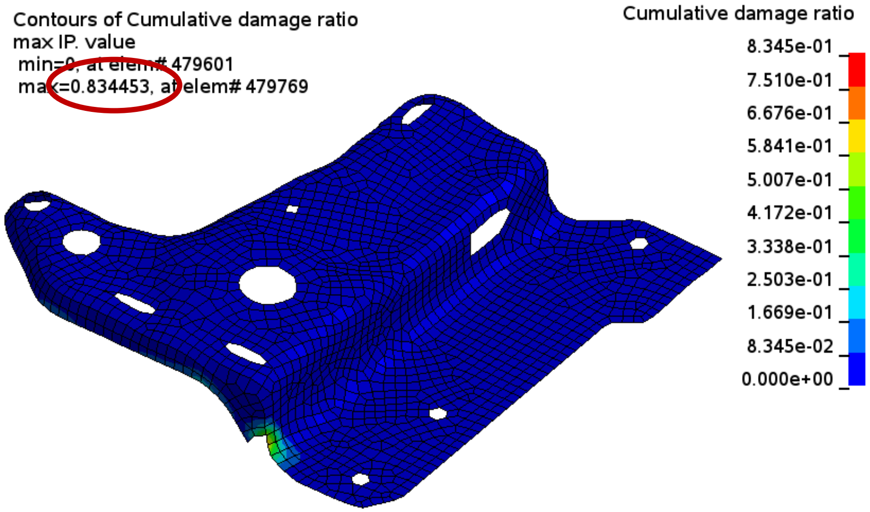
Card 1	1	2	3	4	5	6	7	8
Variable	METHOD							
Type	I							

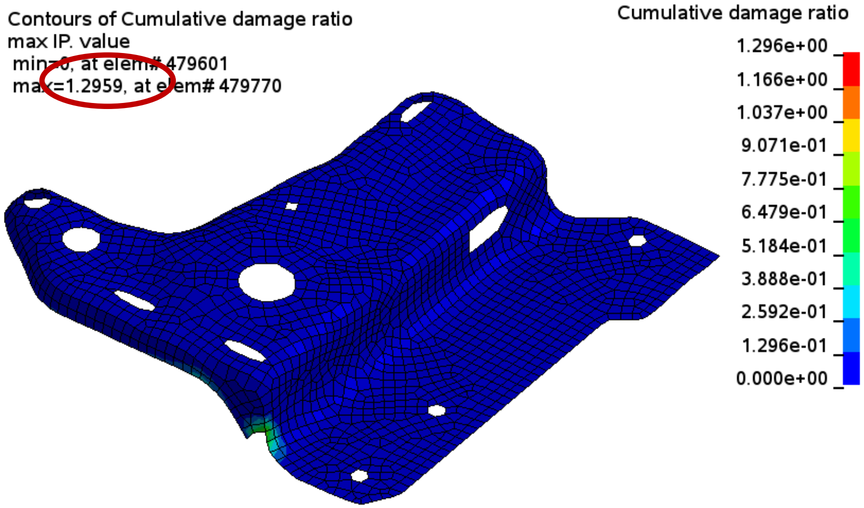
Card 2	1	2	3	4	5	6	7	8
Variable	MID	SIGMA						
Type	I	F						

VARIABLE	DESCRIPTION
METHOD	Mean stress correction method: EQ.0: Goodman equation EQ.1: Soderberg equation EQ.2: Gerber equation EQ.3: Goodman tension only EQ.4: Gerber tension only EQ.11: Morrow equation EQ.12: Smith-Watson-Topper equation

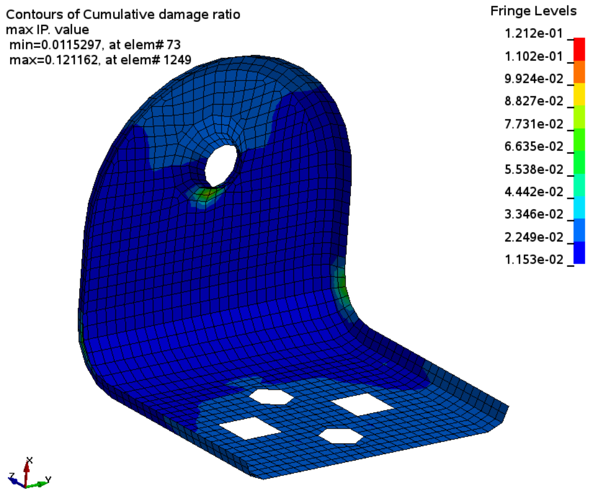


- ✓Mean stress is introduced by ***INITIAL_STRESS_SHELL** ***INITIAL_STRESS_SOLID …**
- ✓Mean stress correction method is set by ***FATIGUE_MEAN_STRESS_ CORRECTION**

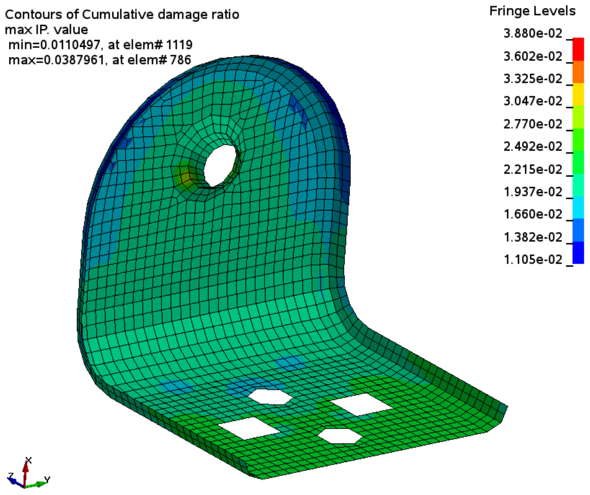




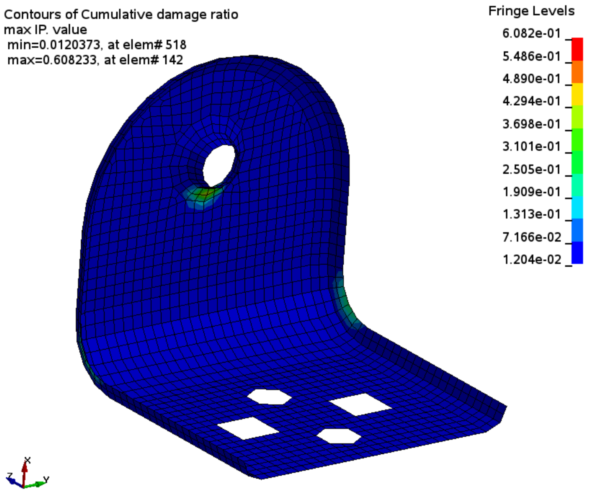
Initial damage ratio in fatigue



1: Acceleration PSD in x



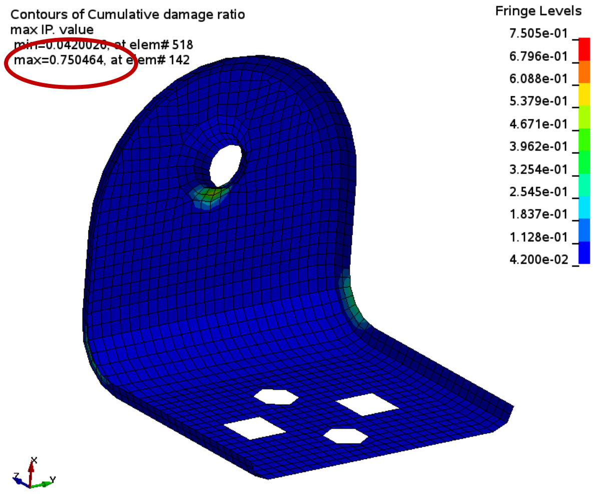
2: Acceleration PSD in y



3: Acceleration PSD in z

***INITIAL_FATIGUE_DAMAGE_RATIO**

Loading cases	Damage ratio at element 142 (upper)
1	1.21162×10^{-1}
2	2.10685×10^{-2}
3	6.08233×10^{-1}
Total	7.50464×10^{-1}



Cumulative damage ratio

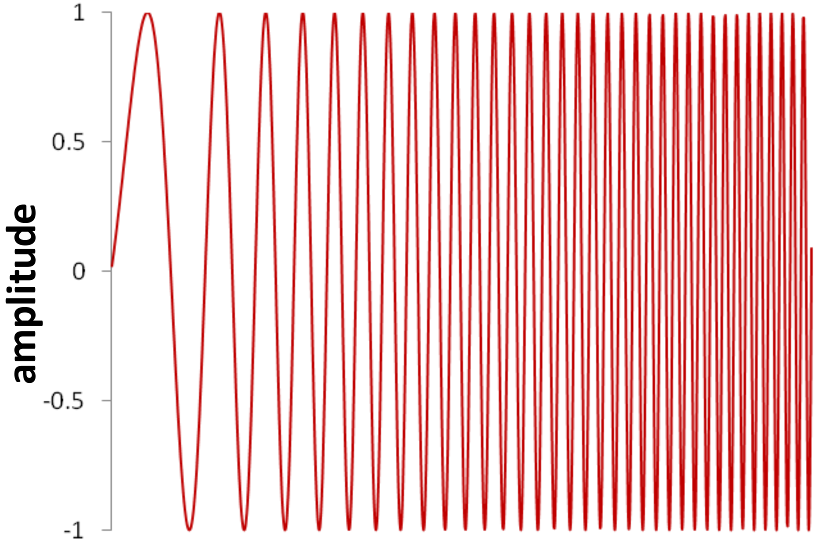
4.3.2) Fatigue analysis in SSD

Introduction

FREQUENCY_DOMAIN_SSD_*FATIGUE

- Calculate fatigue life of structures under steady state vibration (e.g. sine sweep)
- Based on S-N fatigue curve
- Based on Miner's Rule of Cumulative Damage Ratio
- Rainflow counting algorithm for each frequency for one period

$$R = \sum_i \frac{n_i}{N_i}$$



Sine sweep

Keyword

*FREQUENCY_DOMAIN_SSD_*FATIGUE**

Card 3	1	2	3	4	5	6	7	8
Variable					STRTYP	NOUT	NOTYP	NOVA
Type					I	I	I	I
Default					0	0	0	0

Card 4	1	2	3	4	5	6	7	8
Variable	NID	NTYP	DOF	VAD	LC1	LC2	LC3	VID
Type	I	I	F	F	I	I	I	I
Default	none	0	none	none	none	none	0	0

VARIABLE	DESCRIPTION
STRTYP	Stress type used in fatigue analysis = 0 Von Mises stress = 1 Maximum principal stress = 2 Maximum shear stress
LC3	Load Curve ID defining load duration for each frequency. This parameter is optional and is only needed for simulating sine sweep vibration

Example of SSD fatigue

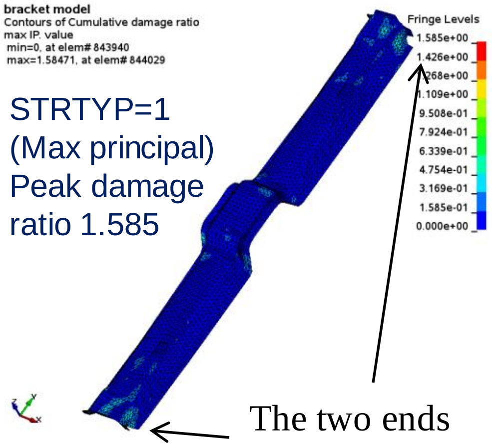
Loading condition

Freq (Hz)	Acl (g)	Duration (min)
16	0.5	12
20	0.5	12
25	0.5	12
31.5	0.5	12
…	…	…
2000	0.5	12

SN fatigue curve

σ (MPa)	N
100	8×10^4
10	8×10^5
1.	8×10^6
0.1	8×10^7
0.01	8×10^8

STRTYP=0
(Von Mises)
Peak damage
ratio 1.705



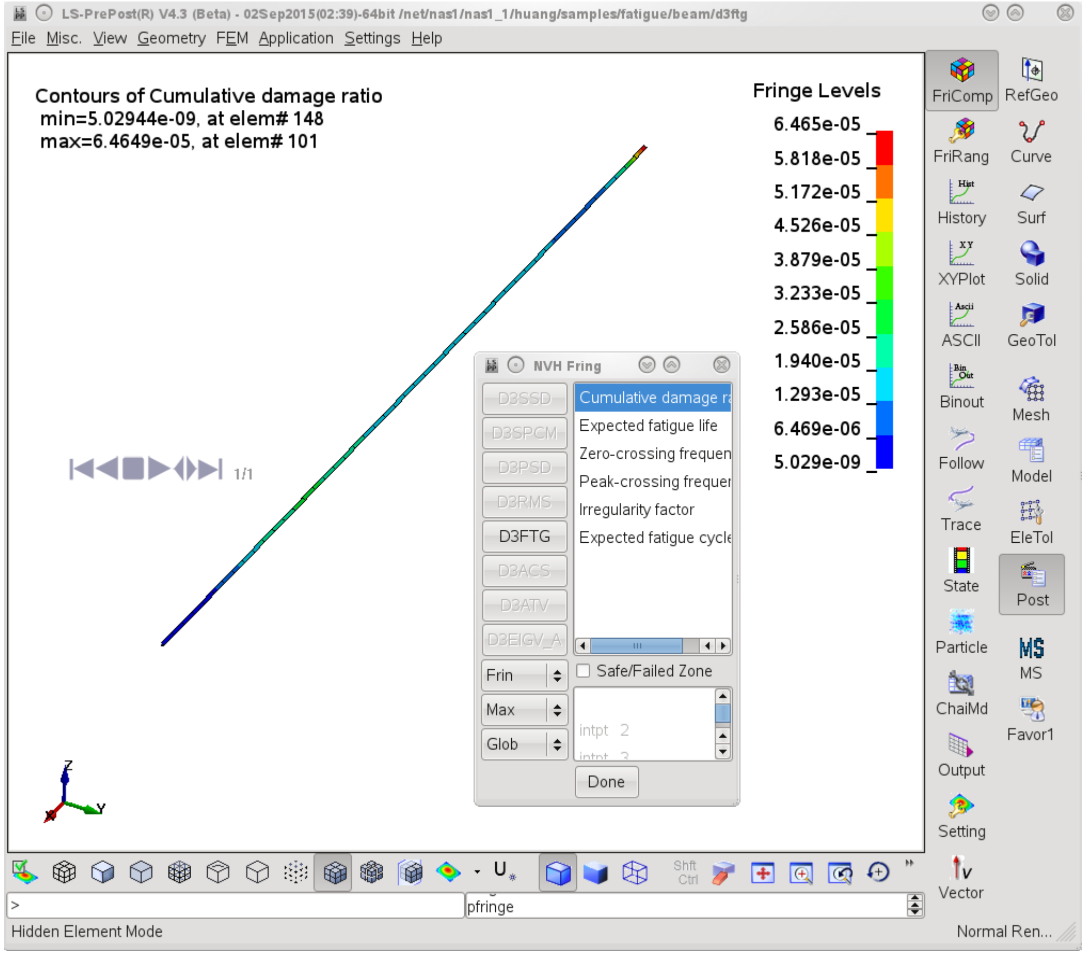
STRTYP=1
(Max principal)
Peak damage
ratio 1.585

The two ends
are constrained

STRTYP=2
(Max shear)
Peak damage
ratio 1.518

Fatigue analysis with beams

- Applicable to beam elements which are not based on resultant formulation
- User need to turn on **BEAMIP** in ***DATABASE_EXTENT_BINARY**
- Results are saved in d3ftg, supported by **LS-PrePost 4.3**



LS-PrePost(R) V4.3 (Beta) - 02Sep2015(02:39)-64bit /net/nas1/nas1_1/huang/samples/fatigue/beam/d3ftg

Contours of Cumulative damage ratio
min=5.02944e-09, at elem# 148
max=6.4649e-05, at elem# 101

Fringe Levels
6.465e-05
5.818e-05
5.172e-05
4.526e-05
3.879e-05
3.233e-05
2.586e-05
1.940e-05
1.293e-05
6.469e-06
5.029e-09

NVH Fring
Cumulative damage ra
Expected fatigue life
Zero-crossing frequen
Peak-crossing frequen
Irregularity factor
Expected fatigue cycle

☐ Safe/Failed Zone

Hidden Element Mode

4.4) Time domain fatigue method

FATIGUE_*{OPTION}

Card 1	1	2	3	4	5	6	7	8
Variable	SSID	SSTYPE						
Type	I	I						

Card 2	1	2	3	4	5	6	7	8
Variable	DT							
Type	I							

Card 3	1	2	3	4	5	6	7	8
Variable	STRES	INDEX	RESTRT					
Type	I	I	I					

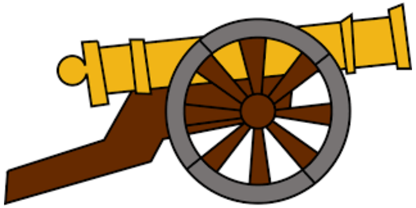
VARIABLE	DESCRIPTION	VARIABLE	DESCRIPTION
STRES	Type of fatigue analysis variable: EQ.0: Stress (default) EQ.1: Strain	INDEX	Stress / strain index: EQ.0: Von-Mises stress/ strain EQ.1: Maximum principal stress/strain EQ.2: Maximum shear stress/strain EQ.-1: xx-stress/strain EQ.-2: yy-stress/strain EQ.-3: zz-stress/strain EQ.-4: xy-stress/strain EQ.-5: yz-stress/strain EQ.-6: zx-stress/strain

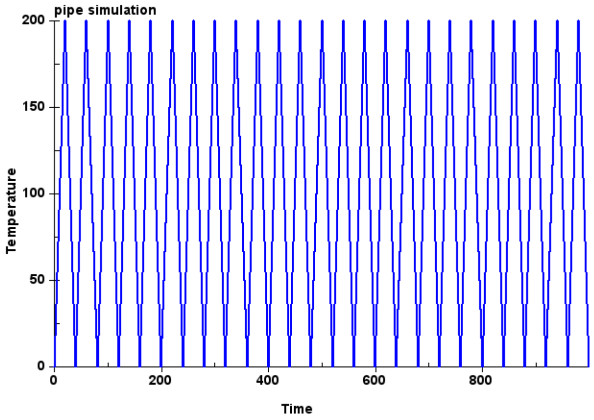
OPTION:

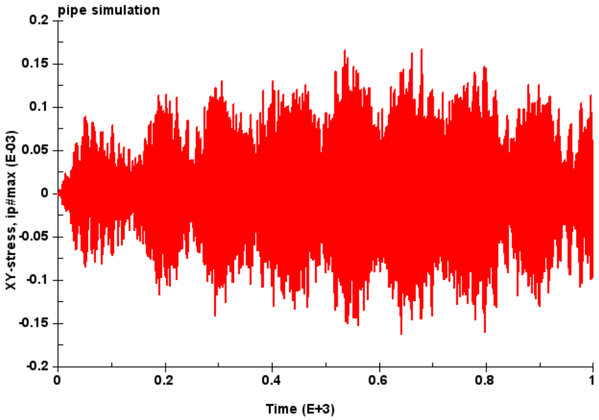
ELOUT

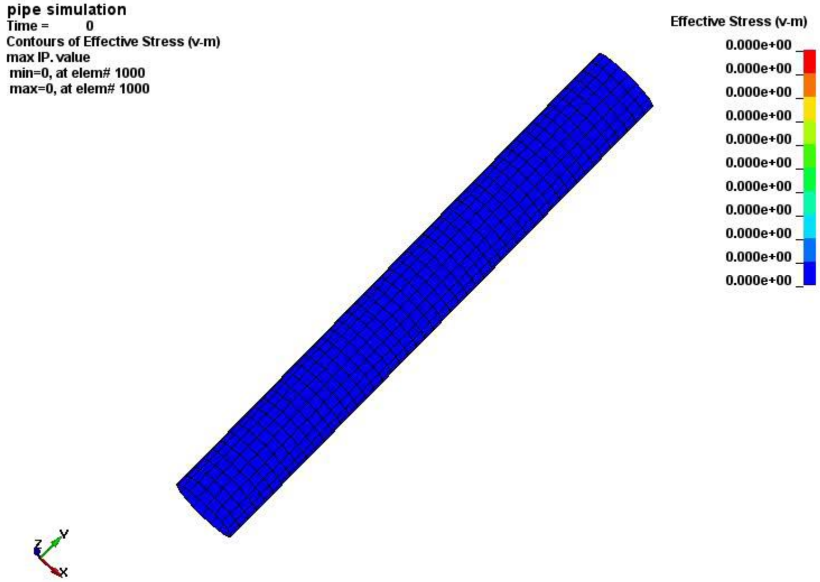
4.4.1) Stress-based approach

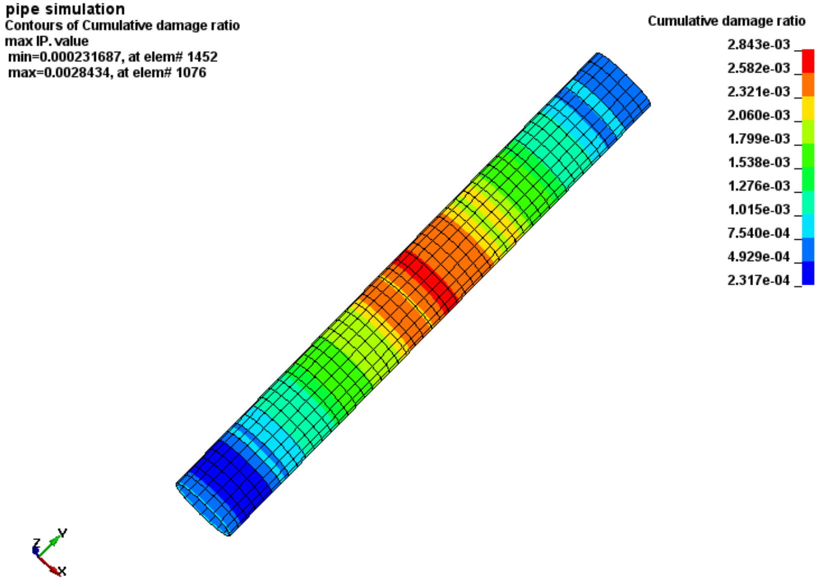
***LOAD_THERMAL_LOAD_CURVE**
***MAT_ELASTIC_PLASTIC_THERMAL**











4.4.2) Strain-based approach

Local strain life equation

$$\frac{\Delta\varepsilon}{2}=\frac{\sigma'_f}{E}\left(2N_f\right)^b+\varepsilon'_f\left(2N_f\right)^c$$

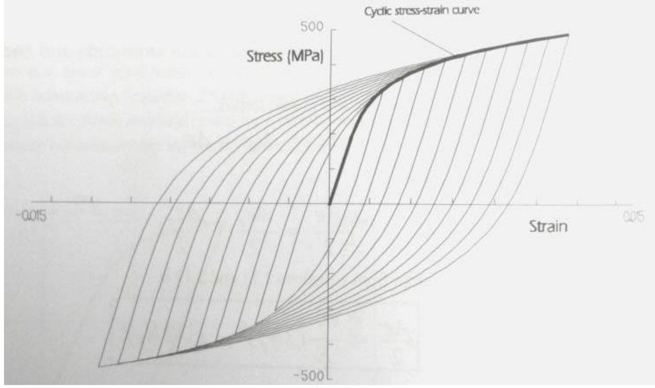
Smith-Watson-Topper mean stress correction

$$\frac{\Delta\varepsilon}{2}\sigma_{\max}=\frac{\left(\sigma'_f\right)^2}{E}\left(2N_f\right)^{2b}+\sigma'_f\varepsilon'_f\left(2N_f\right)^{b+c}$$

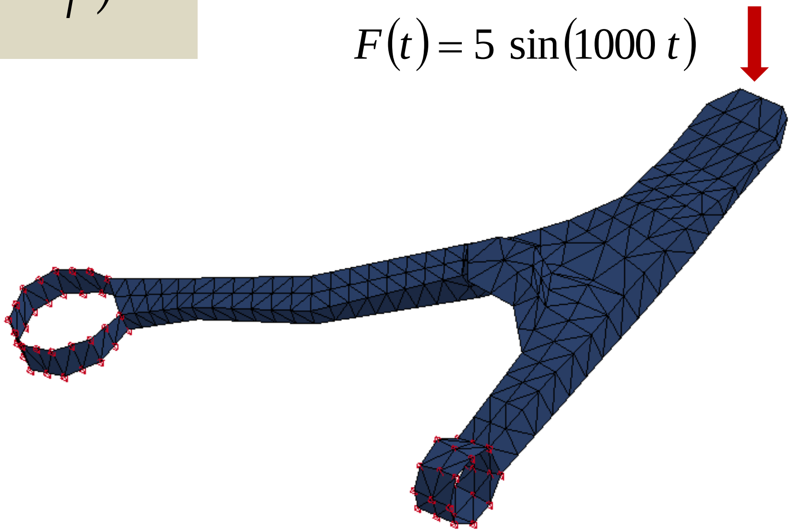
Morrow's mean stress correction

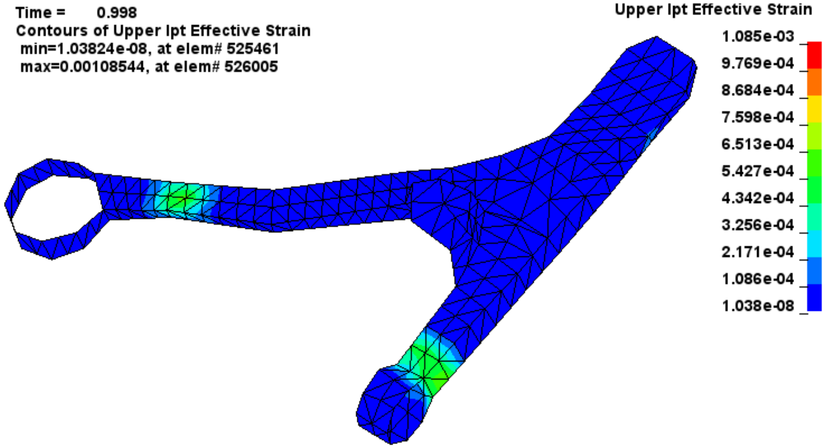
$$\frac{\Delta\varepsilon}{2}=\frac{\sigma'_f-\sigma_m}{E}\left(2N_f\right)^b+\varepsilon'_f\left(2N_f\right)^c$$

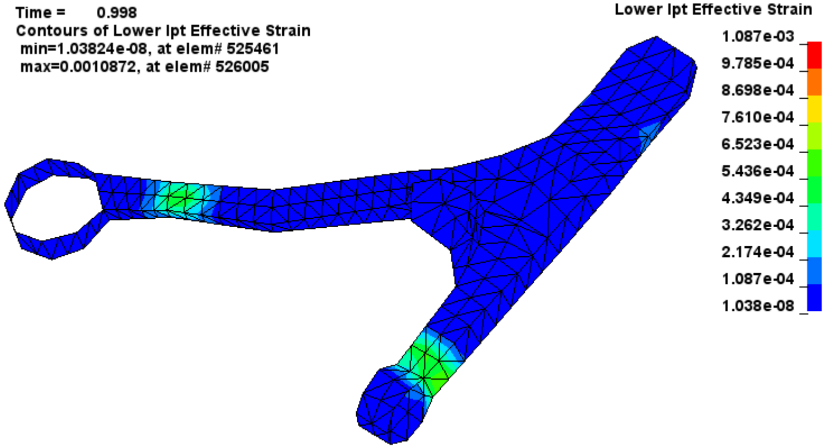
Cyclic stress-strain hysteresis loop

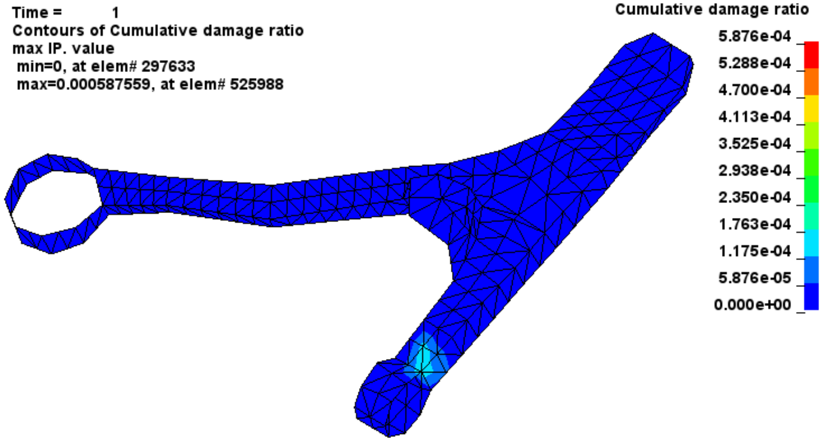


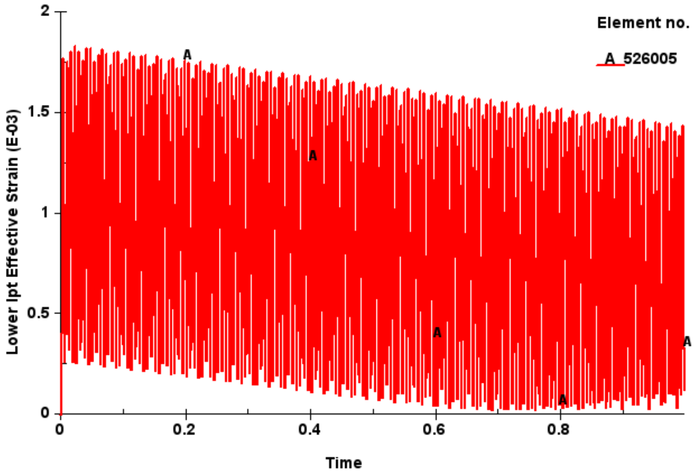
$$F(t)=5\,\sin(1000\,t)$$




Time = 0.998
Contours of Upper Ipt Effective Strain
min=1.03824e-08, at elem# 525461
max=0.00108544, at elem# 526005
Upper Ipt Effective Strain
1.085e-03
9.769e-04
8.684e-04
7.598e-04
6.513e-04
5.427e-04
4.342e-04
3.256e-04
2.171e-04
1.086e-04
1.038e-08


Time = 0.998
Contours of Lower Ipt Effective Strain
min=1.03824e-08, at elem# 525461
max=0.0010872, at elem# 526005
Lower Ipt Effective Strain
1.087e-03
9.785e-04
8.698e-04
7.610e-04
6.523e-04
5.436e-04
4.349e-04
3.262e-04
2.174e-04
1.087e-04
1.038e-08

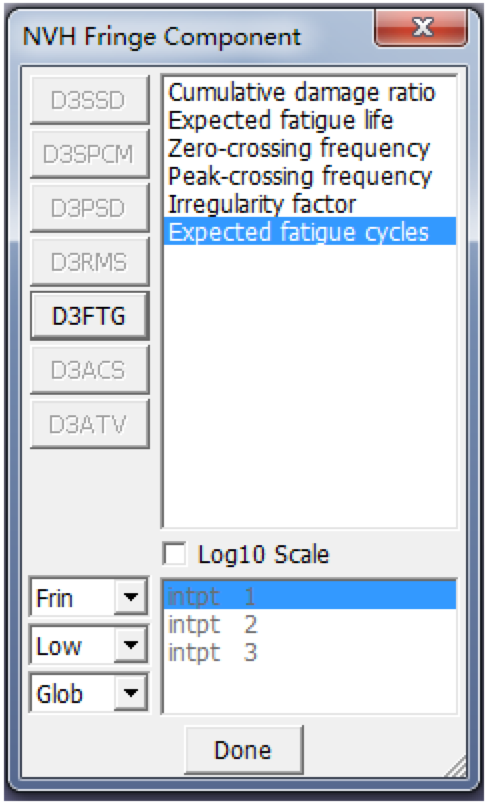

Time = 1
Contours of Cumulative damage ratio
max IP. value
min=0, at elem# 297633
max=0.000587559, at elem# 525988
Cumulative damage ratio
5.876e-04
5.288e-04
4.700e-04
4.113e-04
3.525e-04
2.938e-04
2.350e-04
1.763e-04
1.175e-04
5.876e-05
0.000e+00


2
1.5
1
0.5
0
Lower Ipt Effective Strain (E-03)
0
0.2
0.4
0.6
0.8
Time
Element no.
A 526005
A

4.5) Fatigue analysis database: d3ftg

- Make sure to show only the parts subjected to fatigue computation.
- Results are given at integration points.
- Six results are included in d3ftg file:
  - ✓ Result 1: cumulative damage ratio (check the Max for integration pts)
  - ✓ Result 2: expected fatigue life (check the Min for integration pts) (for random vibration only)
  - ✓ Result 3: zero-crossing frequencies (for random vibration only)
  - ✓ Result 4: peak-crossing frequencies (for random vibration only)
  - ✓ Result 5: irregularity factor (for random vibration only)
  - ✓ Result 6: expected fatigue cycles (for random vibration only)

d3ftg fringe component

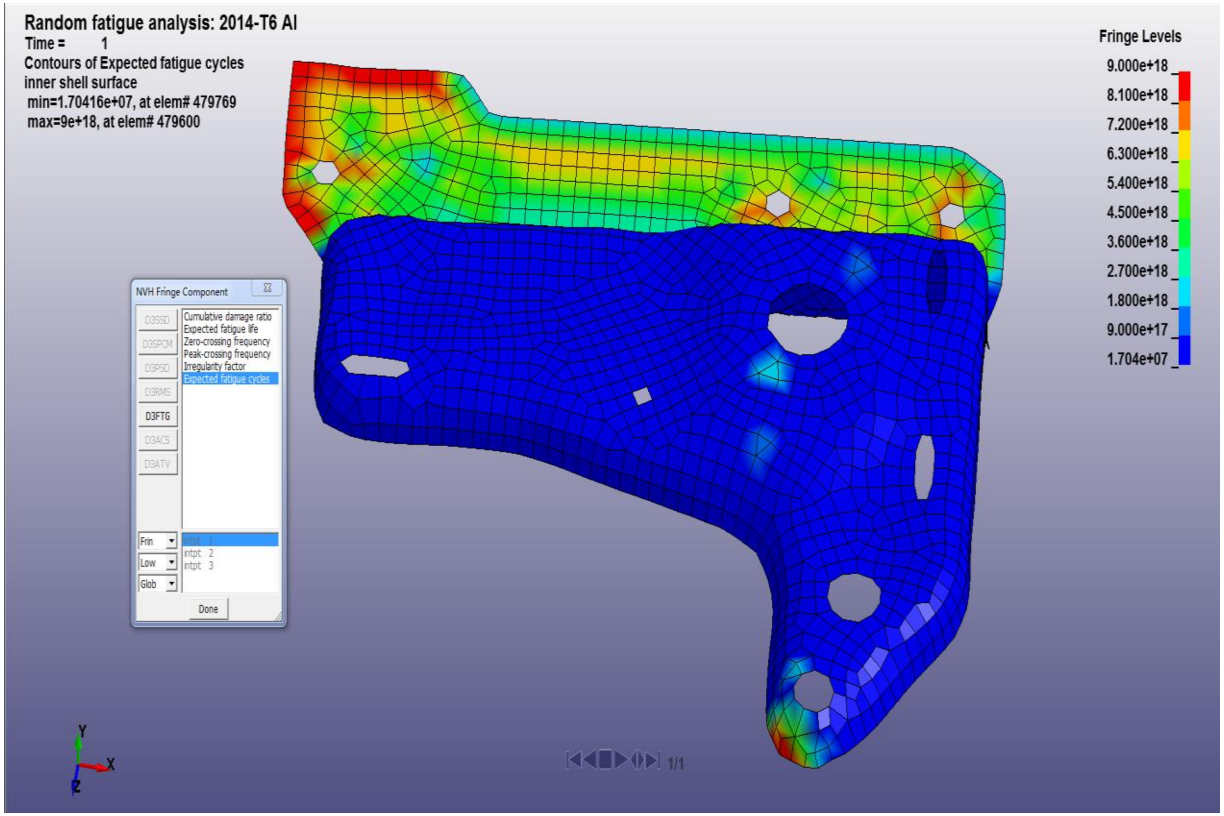


NVH Fringe Component

- Cumulative damage ratio
- Expected fatigue life
- Zero-crossing frequency
- Peak-crossing frequency
- Irregularity factor
- Expected fatigue cycles

☐ Log10 Scale

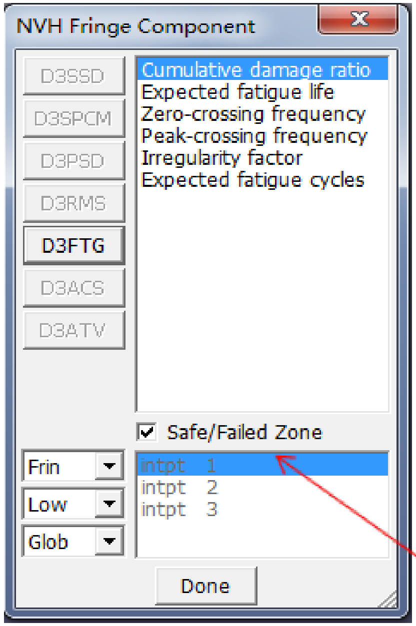
- intpt 1
- intpt 2
- intpt 3



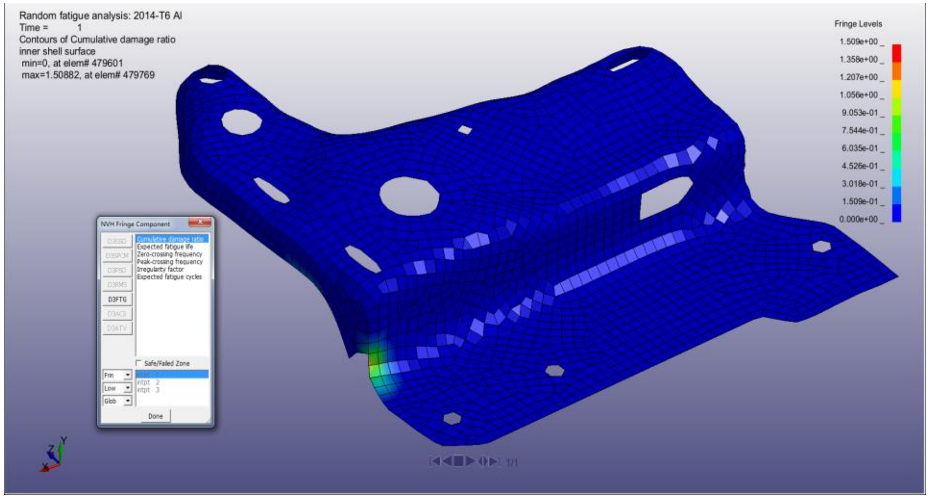
Random fatigue analysis: 2014-T6 Al
Time = 1
Contours of Expected fatigue cycles
inner shell surface
min=1.70416e+07, at elem# 479769
max=9e+18, at elem# 479600

d3ftg

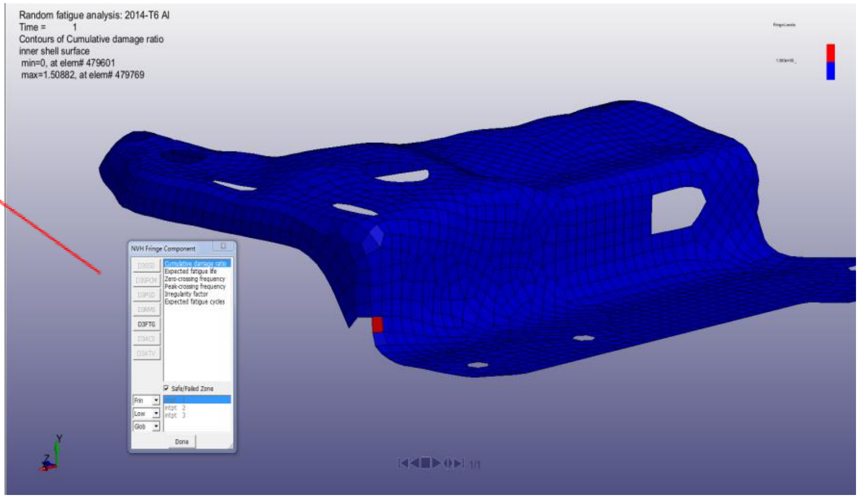
Show the fatigue Safe/Failed zone.



The Safe/Failed zone function can help user to locate the fatigue failed zone quickly.

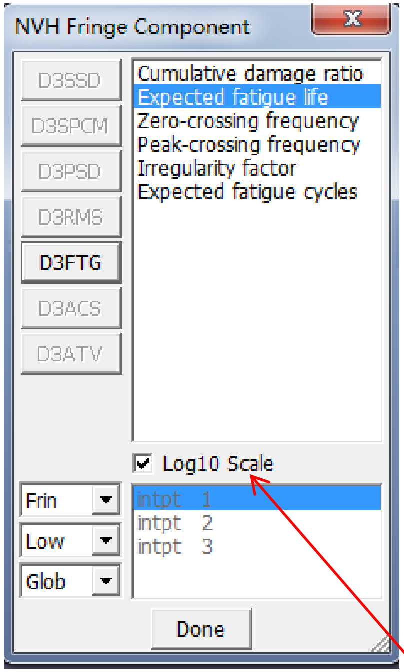


Random fatigue analysis: 2014-T6 Al
Time = 1
Contours of Cumulative damage ratio
inner shell surface
min=0, at elem# 479601
max=1.50882, at elem# 479769



Random fatigue analysis: 2014-T6 Al
Time = 1
Contours of Cumulative damage ratio
inner shell surface
min=0, at elem# 479601
max=1.50882, at elem# 479769

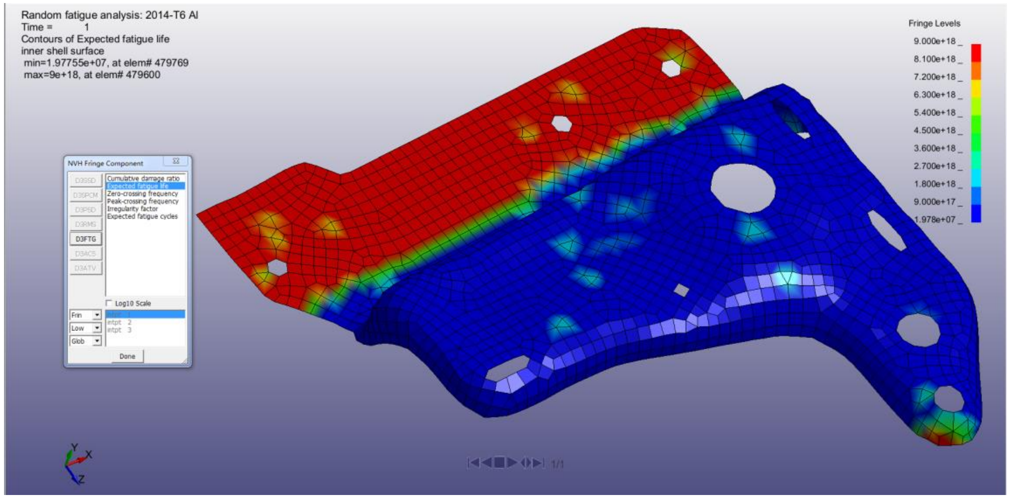
Show log scaling on expected fatigue life.



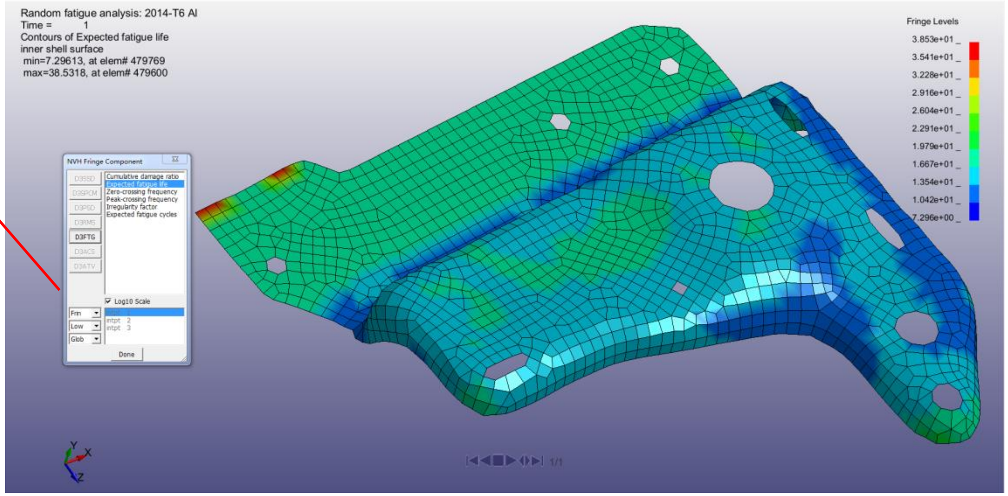
Cumulative damage ratio
Expected fatigue life
Zero-crossing frequency
Peak-crossing frequency
Irregularity factor
Expected fatigue cycles

☑ Log10 Scale

intpt 1
intpt 2
intpt 3



Random fatigue analysis: 2014-T6 Al
Time = 1
Contours of Expected fatigue life
inner shell surface
min=1.97755e+07, at elem# 479769
max=9e+18, at elem# 479600



Random fatigue analysis: 2014-T6 Al
Time = 1
Contours of Expected fatigue life
inner shell surface
min=7.29613, at elem# 479769
max=38.5318, at elem# 479600

The log10 scale can be helpful to show the fringe of expected fatigue life, which may have a huge span of values.

5) CONCLUSION & FUTURE WORK

- Framework for vibration / acoustic / fatigue solvers created
- Basic capabilities / functionalities implemented
- Continue to work on direct SSD, damping, frequency dependent material properties, auto / aircraft seats
- DDAM: more criteria / standards on Shock Design Values
- Aero-acoustics
- Thermo-acoustic
- Pre- and post-processing for SEA
- Multi-axial fatigue analysis
- Strain based fatigue analysis using elastic FEM (Neuber' equation is needed)
- Progressive fatigue database to show the fatigue failure evolution in time domain (like d3plot)

Thank you !