Frequency Domain Analysis in LS-DYNA®

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Outline

1. Introduction
2. Frequency response functions
3. Steady state dynamics
4. Random vibration and fatigue
5. Response spectrum analysis
6. Acoustic analysis by BEM and FEM
7. Conclusion and future developments
1. INTRODUCTION
New keywords for frequency domain analysis

*FREQUENCY_DOMAIN_FRF
*FREQUENCY_DOMAIN_SSD
*FREQUENCY_DOMAINRANDOM_VIBRATION{OPTION}
*FREQUENCY_DOMAINACOUSTIC_BEM{OPTION}
*FREQUENCY_DOMAINACOUSTIC_FEM
*FREQUENCY_DOMAINRESPONSE_SPECTRUM
**Frequency domain vs. time domain**

- A time-domain graph shows how a signal changes over time.
- A frequency-domain graph shows the distribution of the energy (magnitude, etc.) of a signal over a range of frequencies.

**Frequency domain analysis**
- Harmonic (steady state vibration)
- Resonance
- Linear dynamics
- Long history (fatigue testing)
- Non-deterministic load (random analysis)

**Time domain analysis**
- Transient analysis (penetration)
- Impact (crash simulation)
- Large deformation (fracture)
- Non-linearity (contact)
A given function or signal can be converted between the time and frequency domains with a pair of mathematical operators called a transform.

Fourier Transform

Inverse Fourier Transform

\[ H(\omega) = \int_{-\infty}^{\infty} h(t) e^{i\omega t} \, dt \]

\[ h(t) = \frac{1}{2\pi} \int_{-\infty}^{\infty} H(\omega) e^{-i\omega t} \, d\omega \]
Applications of the frequency domain features

✓ Vehicle NVH
  • *Interior noise*
  • *Exterior radiated noise*
  • *Vibration*

✓ Vehicle Durability
  • *Cumulative damage ratio*
  • *Expected life (mileage)*

✓ Aircraft / rocket / spacecraft vibro-acoustics

✓ Durability analysis of machines and electronic devices

✓ Acoustic design of athletic products

✓ Civil Engineering
  • *Architectural acoustics (auditorium, concert hall)*
  • *Earthquake resistance*

✓ Off-shore platforms, wind turbine, etc.
  • *Random vibration*
  • *Random fatigue*
Application of LS-DYNA in automotive industry

- 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.
BINARY databases

Keyword *DATABASE_FREQUENCY_BINARY_{OPTION}*

<table>
<thead>
<tr>
<th>Database</th>
<th>Isrcode</th>
<th>used for</th>
</tr>
</thead>
<tbody>
<tr>
<td>D3SSD</td>
<td>21</td>
<td>Steady state dynamics</td>
</tr>
<tr>
<td>D3SPCM</td>
<td>22</td>
<td>Response spectrum analysis</td>
</tr>
<tr>
<td>D3PSD</td>
<td>23</td>
<td>Random vibration</td>
</tr>
<tr>
<td>D3RMS</td>
<td>24</td>
<td>Random vibration</td>
</tr>
<tr>
<td>D3FTG</td>
<td>25</td>
<td>Random fatigue</td>
</tr>
<tr>
<td>D3ACS</td>
<td>26</td>
<td>FEM acoustics</td>
</tr>
</tbody>
</table>

ASCII databases

- FRF: frf_amplitude, frf_angle, frf_real, frf_imag
- BEM acoustics: Press_Pa, Press_dB, bepres, fringe_*,
  panel_contribution_NID,
- SSD: elout_ssd, nodout_ssd, …
2

FREQUENCY RESPONSE FUNCTIONS
The concept of Frequency Response Function is the foundation of modern experimental system analysis and experimental modal analysis.

Keyword *FREQUENCY_DOMAIN_FRF

Express structural response due to unit load as a function of frequency.

Shows the property of the structure system.

Is a complex function, with real and imaginary components. They may also be represented in terms of magnitude and phase angle.

Result files: frf_amplitude, frf_angle

Support efficient restart.
## FRF formulations

<table>
<thead>
<tr>
<th>Accelerance, Inertance</th>
<th>Acceleration Force</th>
</tr>
</thead>
<tbody>
<tr>
<td>Effective Mass</td>
<td>Force Acceleration</td>
</tr>
<tr>
<td>Mobility</td>
<td>Velocity Force</td>
</tr>
<tr>
<td>Impedance</td>
<td>Force Velocity</td>
</tr>
<tr>
<td>Dynamic Compliance, Admittance, Receptance</td>
<td>Displacement Force</td>
</tr>
<tr>
<td>Dynamic Stiffness</td>
<td>Force Displacement</td>
</tr>
</tbody>
</table>
Example: Accelerance FRF for a plate

Harmonic point force excitation

Reference:
**Mode** | **Damping ratio (×0.01)**
--- | ---
1 | 0.450
2 | 0.713
3 | 0.386
4 | 0.328
5 | 0.340
6 | 0.624
7 | 0.072
8 | 0.083

---

**Graph Details:**
- **Point A:** Acceleration vs. Excitation Frequency (Hz)
- **Point B:** Acceleration vs. Excitation Frequency (Hz)

**Graph Annotations:**
- A. Experimental
- B. LS-DYNA (no damping)
- C. LS-DYNA (damping ratio 0.01)
- D. LS-DYNA (f-dept. damping ratio)
Example: nodal force/resultant force

Left end of the beam is fixed and subjected to z-directional unit acceleration.

Nodal force and resultant force FRF at the left end can be obtained.
Example: FRF for a trimmed BIW

Nodal force applied and displacement measured
3

STEADY STATE DYNAMICS
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 periodical 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.
- May be called as
  - Harmonic vibration
  - Steady state vibration
  - Steady state dynamics
- Keyword *FREQUENCY_DOMAIN_SSD
- Binary plot file d3ssd

\[ F(t) = F_0 \sin(\omega t + \phi) \]
Example: a rectangular plate under pressure

Benchmark example of rectangular plate
Freq = 6
Contours of X-stress
inner shell surface
min=-9.674e-07, at elem# 705
max=4.228e+04, at elem# 3160

Benchmark example of rectangular plate
Freq = 6
Contours of X-stress
outer shell surface
min=-4.2279.3, at elem# 3160
max=9.674e-07, at elem# 705

X-Stress
Enforced Motion

Relative displacement method

- Base acceleration loading is applied through inertial force.
- Total displacement (velocity, acceleration) is obtained by adding the relative displacement (velocity, acceleration) to the base displacement (velocity, acceleration).

\[
\begin{align*}
\mathbf{u} &= \mathbf{u}_{\text{relative}} + \mathbf{u}_{\text{base}} \\
\mathbf{\dot{u}} &= \mathbf{\dot{u}}_{\text{relative}} + \mathbf{\dot{u}}_{\text{base}} \\
\mathbf{\ddot{u}} &= \mathbf{\ddot{u}}_{\text{relative}} + \mathbf{\ddot{u}}_{\text{base}}
\end{align*}
\]

Large mass method

- A very large mass \( m_L \), which is usually \( 10^5-10^7 \) times of the mass of the entire structure, is attached to the nodes under excitation.
- A very large nodal force \( p_L \) is applied to the excitation dof to produce the desired enforced motion.

\[
\begin{align*}
p_L &= m_L \mathbf{\ddot{u}} \\
p_L &= i\omega m_L \mathbf{\dot{u}} \\
p_L &= -\omega^2 m_L \mathbf{u}
\end{align*}
\]
A rectangular plate subjected to nodal acceleration excitation

- The rectangular plate is excited by $z$-direction acceleration at one end.
- Steady state response at the other end is desired.

**Relative disp. method**

- The excitation points are fixed in loading dof.
- The resulted response is added with base motion to get total response.

**Large mass method**

- A large mass with $10^6$ times the original mass is added to excitation points using keyword *element_mass_node_set.
- To produce desired acceleration, nodal force is applied to the nodes in loading dof.
- The excitation points are free in loading dof.
Example: auto model subjected to base acceleration.

**Modal information**
- Number of nodes: \( \approx 290k \)
- Number of shells: \( \approx 532k \)
- Number of solids: \( \approx 3k \)

Base acceleration spectrum is applied to a shaker table. 500 modes up to 211 Hz are used in the simulation. The response is computed up to 100 Hz.
Freq = 15 Hz

Freq = 25 Hz

Freq = 35 Hz

Z-acceleration response

Frequency (Hz)
RANDOM VIBRATION AND FATIGUE
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)
- Provide input data for random fatigue and durability analysis

Examples:
- Fatigue
- 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

Based on Boeing’s N-FEARA package
Correlation of multiple excitations

The multiple excitations on structure can be

- uncorrelated
- partially correlated
- fully correlated

(cross PSD function needed)

Keyword

*FREQUENCY_DOMAIN_RANDOM_VIBRATION

<table>
<thead>
<tr>
<th>Card 5</th>
<th>SID</th>
<th>STYPE</th>
<th>DOF</th>
<th>LDPSD</th>
<th>LDVEL</th>
<th>LDFLW</th>
<th>LDSPN</th>
<th>CID</th>
</tr>
</thead>
<tbody>
<tr>
<td>Variable</td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td>7</td>
<td>8</td>
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<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
</tr>
<tr>
<td>Default</td>
<td></td>
<td></td>
<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
</tr>
</tbody>
</table>

When SID and STYPE are both < 0, they give the IDs of correlated excitations
The model is a simple pipe with 83700 elements and 105480 nodes. It is subjected to base acceleration PSD 1) in x, y and z-directions uncorrelated; 2) in x, y and z-directions correlated.

### RMS of Von Mises stress

(the pipe is subject to x, y, z excitations, uncorrelated)

<table>
<thead>
<tr>
<th>Freq (Hz)</th>
<th>Acceleration psd (g^2/Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.01</td>
</tr>
<tr>
<td>40</td>
<td>0.01</td>
</tr>
<tr>
<td>100</td>
<td>0.04</td>
</tr>
<tr>
<td>500</td>
<td>0.04</td>
</tr>
<tr>
<td>1000</td>
<td>0.0065</td>
</tr>
<tr>
<td>2000</td>
<td>0.001</td>
</tr>
</tbody>
</table>

**Max Von Mises RMS stress (MPa)**

<table>
<thead>
<tr>
<th></th>
<th>ANSYS</th>
<th>LS-DYNA</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>99.4</td>
<td>100.7</td>
</tr>
</tbody>
</table>
RMS of Von Mises stress
(the pipe is subject to x, y, z excitations, correlated)

Max Von Mises RMS stress (MPa)

<table>
<thead>
<tr>
<th></th>
<th>ANSYS</th>
<th>LS-DYNA</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>140.3</td>
<td>142.4</td>
</tr>
</tbody>
</table>
The tube was fixed to the shaker tables using aluminum blocks which surrounded the tube and were tightened using screws.

**Base acceleration PSD load**

<table>
<thead>
<tr>
<th>No.</th>
<th>Freq.</th>
<th>g²/Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5</td>
<td>0.004082</td>
</tr>
<tr>
<td>2</td>
<td>250</td>
<td>0.004082</td>
</tr>
</tbody>
</table>

*Courtesy of Rafael, Israel.*
PSD of acceleration - CH 3 - Test results vs. Ls-Dyna results with simple and improved fixture
A cluster server is analyzed by LS-DYNA to understand the location of vibration damage under standard random vibration condition.

<table>
<thead>
<tr>
<th>Hz</th>
<th>g^2/Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0.001</td>
</tr>
<tr>
<td>20</td>
<td>0.003</td>
</tr>
<tr>
<td>40</td>
<td>0.003</td>
</tr>
<tr>
<td>80</td>
<td>0.02</td>
</tr>
<tr>
<td>120</td>
<td>0.02</td>
</tr>
<tr>
<td>200</td>
<td>0.0015</td>
</tr>
<tr>
<td>500</td>
<td>0.0015</td>
</tr>
</tbody>
</table>
It is found that the $3\sigma$ Von-Mises stress is less than the yield stress of the material (176 Mpa).

<table>
<thead>
<tr>
<th></th>
<th>1σ</th>
<th>3σ</th>
</tr>
</thead>
<tbody>
<tr>
<td>$U$ (mm)</td>
<td>0.78</td>
<td>2.34</td>
</tr>
<tr>
<td>$S_{V-m}$ (MPa)</td>
<td>41.2</td>
<td>123.6</td>
</tr>
</tbody>
</table>
MPP Random Vibration Analysis

Nodes: 800k
Elements: 660k
Modes: 1000

Model courtesy of Predictive Engineering
Introduction: Random Fatigue

- 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

\[ R = \sum \frac{n_i}{N_i} \]

- Schemes:
  - Steinberg’s Three-band technique considering the number of stress cycles at the 1\(\sigma\), 2\(\sigma\), and 3\(\sigma\) levels.
  - Dirlik method based on the 4 Moments of PSD.
  - Narrow band method
  - Wirsching method
  - ...
S-N fatigue curve definition

- By *define_curve
- By equation

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

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

*N: number of cycles for fatigue failure
*S: stress

- 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.

Source of information: http://www.efunda.com
Steinberg’s Three Band Technique

<table>
<thead>
<tr>
<th>Standard Deviation</th>
<th>Percentage of Occurrence</th>
<th>Number of cycles for failure</th>
</tr>
</thead>
<tbody>
<tr>
<td>1σ stress</td>
<td>68.3%</td>
<td>$N_1$</td>
</tr>
<tr>
<td>2σ stress</td>
<td>27.1%</td>
<td>$N_2$</td>
</tr>
<tr>
<td>3σ stress</td>
<td>4.33%</td>
<td>$N_3$</td>
</tr>
</tbody>
</table>

$n_1 = E(0) \cdot T \cdot 0.683$

$n_2 = E(0) \cdot T \cdot 0.271$

$n_3 = E(0) \cdot T \cdot 0.0433$

$E[D] = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3}$

E(0): Zero-crossing frequency with positive slope

Reference

Example: Aluminum bracket

Aluminum 2014 T6
\( \rho = 2800 \text{ Kg/m}^3 \)
\( E = 72,400 \text{ MPa} \)
\( \nu = 0.33 \)

Time of exposure: 4 hours

Nodes constrained to shaker table

PSD = 2 \( g^2/\text{Hz} \)

Z-acceleration PSD

S-N fatigue curve
RMS of Von-Mises stress (unit: GPa)

Accumulative damage ratio
(by Steinberg’s method)
Example: beam with predefined notch

Aluminum alloy 5754
\[ \rho = 2700 \text{ Kg/m}^3 \]
\[ E = 70,000 \text{ MPa} \]
\[ \nu = 0.33 \]

Base acceleration is applied at the edge of the hole

Acceleration PSD
(exposure time: 1800 seconds)

Model courtesy of CIMES France
### Analysis method | Expected life | Damage ratio
---|---|---
Experiment | 7mn 25s | -
Steinberg | 4mn 10s | 7.19
Dirlik | 5mn 25s | 5.54
Narrow Band | 2mn 05s | 14.41
Wirsching | 5mn 45s | 5.08
Chaudhury & Dover | 6mn 03s | 6.03
Tunna | 4mn 06s | 7.31
Hancock | 22mn 18s | 1.35

### CODE | RMS S\(_{xx}\) |
---|---
ANSYS | 33.5 MPa
RADIOSS® BULK | 35.7 MPa
LS-DYNA | 35.0 MPa

RMS \(S_x\) = 35.0 MPa at critical point
Cumulative damage ratio by Steinberg’s method

Damage ratio = 7.188

Cumulative damage ratio by Dirlik method

Damage ratio = 5.540
Experiment setup

Failure at the notched point in experiment
A model from railway application

The structure is subjected to base acceleration defined by the International Standard IEC61373 to simulate the long-life test. This standard intends to highlight any weakness which may result in problem as a consequence of operation under environment where vibrations are known to occur in service on a railway vehicle.
Area 1 - damage ratio = 25.5

Area 2 - damage ratio = 3.45
## Initial Damage Ratio

### *FREQUENCY_DOMAIN_RANDOM_VIBRATION_FATIGUE*

<table>
<thead>
<tr>
<th>Card 1</th>
<th>Variable</th>
<th>MDMIN</th>
<th>MDMAX</th>
<th>FNMIN</th>
<th>FNMAX</th>
<th>RESTRT</th>
<th>MFTG</th>
<th>RESTRM</th>
<th>INFTG</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Type</td>
<td>I</td>
<td>I</td>
<td>F</td>
<td>F</td>
<td>I</td>
<td>I</td>
<td>I</td>
<td>I</td>
</tr>
<tr>
<td></td>
<td>Default</td>
<td>1</td>
<td>0.0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Define Card 7 if option FATIGUE is used and INFTG=1.

<table>
<thead>
<tr>
<th>Card 7</th>
<th>Variable</th>
<th>FILENAME</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Type</td>
<td>C</td>
</tr>
<tr>
<td></td>
<td>Default</td>
<td>d3ftg</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>VARIABLE</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>INFTG</td>
<td>Flag for including initial damage ratio. EQ.0: no initial damage ratio, EQ.1: read existing d3ftg file to get initial damage ratio.</td>
</tr>
<tr>
<td>FILENAME</td>
<td>Path and name of existing binary database (by default, D3FTG) for initial damage ratio.</td>
</tr>
</tbody>
</table>
Time of exposure: 4 hours

Edge fixed to shaker table.

Acceleration PSD = 1g²/Hz for 1-2000 Hz

S-N fatigue curve

RMS σ- von mises for x-ACL PSD

RMS σ-von mises for z-ACL PSD
Damage ratio for x-load only

1.152e-01

Damage ratio for z-load only

6.019e-01

Damage ratio for x+z load

7.171e-01

Dirlik method is used (MFTG = 2).
RESPONSE SPECTRUM ANALYSIS
*FREQUENCY_DOMAIN_RESPONSE_SPECTRUM

- 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 degree freedom system with different natural frequencies.
- The input spectrum is dependent on damping (using *DEFINE_TABLE to define the series of excitation spectrum corresponding to each damping ratio).
- Output binary database: d3spcm (accessible by LS-PREPOST).
- It is an approximate method.
- It has important application in earthquake engineering, nuclear power plants design etc.
Capabilities

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

Frequency interpolation
- Logarithmic
- Semi-logarithmic
- Linear

Results
- BINARY plot file: d3spcm
- ASCII files: nodout_spcm, elout_spcm

Applications
- Civil and hydraulic buildings
  - Hydraulic dams
  - Bridges
  - High buildings
- Nuclear power plants

Input spectrum
- Base velocity
- Base acceleration
- Base displacement
- Nodal force
- Pressure
Example: multi story tower

60 m /15 story tower

Loading conditions
Base acceleration input spectrum

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Acceleration (m/s²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>19.4</td>
</tr>
<tr>
<td>5.0</td>
<td>22.23</td>
</tr>
<tr>
<td>10.0</td>
<td>24.10</td>
</tr>
<tr>
<td>15.0</td>
<td>25.5</td>
</tr>
<tr>
<td>20.0</td>
<td>26.6</td>
</tr>
</tbody>
</table>

Mode combination method
SRSS

Damping
\( \zeta = 1\% \)
Z displacement results

NODAL SOLUTION
STEP=9999
UZ (AVG)
RSYS=SOLU
DMX = 3.35591
SMX = 0.05188

LS-DYNA keyword deck by LS-PrePost
Time = 1
Contours of Z-displacement
min=0, at node# 4
max=0.0518036, at node# 364

Fringe Levels
-5.180e-02
4.662e-02
4.144e-02
3.626e-02
3.108e-02
2.590e-02
2.072e-02
1.554e-02
1.036e-02
5.180e-03
0.000e+00
Von Mises stress results

LS-DYNA keyword deck by LS-PrePost

Time = 1
Contours of Effective Plastic Strain

max IR value
min=0, at elem# 1
max=1.04708e+08, at elem# 6357
Example: arch dam

- 464.88 feet high
- Rigid foundation
- Subjected to $x$-directional ground acceleration spectrum

The pseudo-acceleration spectrum of El Centro earthquake ground motion ($\zeta=5\%$)

The 1940 El Centro earthquake (or 1940 Imperial Valley earthquake) occurred on May 18 in the Imperial Valley in Southern California near the border of the United States and Mexico. It had a magnitude of 6.9.
ACOUSTIC ANALYSIS BY BEM/FEM
Introduction: BEM Acoustics

*FREQUENCY_DOMAIN_ACOUSTIC_BEM_{OPTION}

Available options include:

PANEL_CONTRIBUTION
HALF_SPACE

Structure loading

FEM transient analysis

SSD

V (P) in time domain

FFT

User data

V (P) in frequency domain

BEM acoustic analysis

Acoustic pressure and SPL (dB) at field points
BEM (accurate)

- Indirect variational boundary element method
- Collocation boundary element method

*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*
Acoustic Panel Contribution

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

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

A simplified tunnel model

Panel 1

Panel 2

Panel 3

Panel 4

Observation point

Projection in the direction of the total pressure

real

imaginary

Panel contribution at node 5401

Percentage (%)

Frequency (Hz)
Radiated Noise by a Car

Sound Pressure Level distribution (dB)

- The radius of the sphere is 3 m

Freq = 11 Hz

Freq = 101 Hz
The plate is 0.2 m away from the vehicle
Half-space Problem

Free space Green’s function

\[ G = \frac{e^{-ikr}}{4\pi r} \]

Half space Green’s function

\[ G_H = \frac{e^{-ikr}}{4\pi r} + R_H \frac{e^{-ikr'}}{4\pi r'} \]

Reflection plane equations:

\[ R_H = \begin{cases} 
1: \text{rigid reflection plane, zero velocity} \\
-1: \text{soft reflection plane, zero sound pressure (water-air interface in underwater acoustics)} 
\end{cases} \]

Helmholtz integral equation

\[ P(\omega) = -\int_S \left( i\rho \omega v_n(\omega) G_H + p(\omega) \frac{\partial G_H}{\partial n} \right) ds \]

The reflection plane is defined by *DEFINE_PLANE.
TL (Transmission loss) is the difference in the sound power level between the incident wave entering and the transmitted wave exiting the muffler when the muffler termination is anechoic (no reflection of sound).

\[ TL = 10 \log_{10} \frac{W_i}{W_t} \]
Three-point method

\[ p_i = \frac{1}{2i \sin[k(x_2 - x_1)]} \left( p_1 e^{ikx_2} - p_2 e^{ikx_1} \right) \]

\[ TL = 20 \log_{10} \left( \left| \frac{p_i}{p_o} \right| \right) + 10 \log_{10} \left( \frac{s_i}{s_o} \right) \]

Where, \( s_i \) and \( s_o \) are the inlet and outlet tube areas, respectively.

Four-pole method

\[
\begin{bmatrix}
  p_1 \\
  v_1
\end{bmatrix} =
\begin{bmatrix}
  A & B \\
  C & D
\end{bmatrix} 
\begin{bmatrix}
  p_2 \\
  -v_2
\end{bmatrix}
\]

The four pole parameters \( A, B, C, D \), can be obtained from:

\[
A = p_1 / p_2 \quad | \quad v_2 = 0, v_1 = 1
\]

\[
B = p_1 / -v_2 \quad | \quad p_2 = 0, v_1 = 1
\]

\[
C = v_1 / p_2 \quad | \quad v_2 = 0, v_1 = 1
\]

\[
D = v_1 / -v_2 \quad | \quad p_2 = 0, v_1 = 1
\]

\[
TL = 20 \log_{10} \left( \frac{1}{2} \left| A + B \left( \frac{1}{\rho c} \right) + C \rho c + D \right| \right) + 10 \log_{10} \left( \frac{s_i}{s_o} \right)
\]
Muffler transmission loss by different methods

Cutoff frequency for plane wave theory $f = 1119$ Hz

References

Double expansion chamber
References
**BOUNDARY_ACOUSTIC_MAPPING**

Purpose: Define a set of elements or segments on structure for mapping structural nodal velocity to boundary of acoustic volume.

<table>
<thead>
<tr>
<th>Card</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Variable</td>
<td>SSID</td>
<td>STYP</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Type</td>
<td>I</td>
<td>I</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Default</td>
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<td>0</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**VARIABLE**

<table>
<thead>
<tr>
<th>VARIABLE</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>SSID</td>
<td>Set or part ID</td>
</tr>
</tbody>
</table>
| STYP     | Set type:  
EQ.0: part set ID, see *SET_PART,  
EQ.1: part ID, see *PART,  
EQ.2: segment set ID, see *SET_SEGMENT. |
Mesh A: $20 \times 30$ (600)

Also mesh for structure surface

Mesh B: $15 \times 20$ (300)

Mesh C: $7 \times 11$ (77)

**CPU time** (Intel Xeon 1.6 GHz)

<table>
<thead>
<tr>
<th>Mesh</th>
<th>Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mesh A</td>
<td>16 min 34 sec</td>
</tr>
<tr>
<td>Mesh B</td>
<td>10 min 10 sec</td>
</tr>
<tr>
<td>Mesh C</td>
<td>6 min 49 sec</td>
</tr>
</tbody>
</table>

SPL at field point

- A: matching BEM mesh (mesh A)
- B: mismatching BEM mesh (mesh B)
- C: mismatching BEM mesh (mesh C)
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.
- When ATV option is included, the structure **does not need any external excitation**, and the curve IDs LC1 and LC2 are ignored.

**Output files**

- ✓ **ATV_DB_FIELD_PT_*** (decibel due to unit vn at each node)
- ✓ **ATV_FIELD_PT_*** (complex pressure due to unit vn at each node)
- ✓ **ATV_DB_FIELD_PT_FRINGE_*** (user fringe plot file for decibel due to unit vn at each node)
ATV is useful if the same structure needs to be studied under multiple load cases.

ATV at field points 1-m, due to unit normal velocity at node j

\[
\begin{bmatrix}
P_1 \\
P_2 \\
\vdots \\
P_i \\
P_m
\end{bmatrix} = \begin{bmatrix}
\Omega_{1,1} & \Omega_{1,2} & \cdots & \Omega_{1,j} & \cdots & \Omega_{1,n} \\
\Omega_{2,1} & \Omega_{2,2} & \cdots & \Omega_{2,j} & \cdots & \Omega_{2,n} \\
\vdots & \vdots & \ddots & \vdots & \ddots & \vdots \\
\Omega_{i,1} & \Omega_{i,2} & \cdots & \Omega_{i,j} & \cdots & \Omega_{i,n} \\
\vdots & \vdots & \ddots & \vdots & \ddots & \vdots \\
\Omega_{m,1} & \Omega_{m,2} & \cdots & \Omega_{m,j} & \cdots & \Omega_{m,n}
\end{bmatrix}_{m \times n}
\begin{bmatrix}
V_1 \\
V_2 \\
\vdots \\
V_j \\
\vdots \\
V_n
\end{bmatrix}
\]

Need to be computed only once  
Change from case to case
A simplified auto body model without any inside details

Analysis steps:
1. Modal analysis
2. Steady state dynamics
3. Boundary element acoustics

All done in one run
Introduction: FEM Acoustics

*FREQUENCY_DOMAIN_ACOUSTIC_FEM*

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

Hexahedron  Tetrahedron  Pentahedron
**Model information**

FEM: 2688 elements  
BEM: 1264 elements

Excitation of the compartment (1.4×0.5×0.6) m³ by a velocity of 7mm/s
Pressure distribution

Acoustic analysis of a simplified compa
Time = 100
Contours of Z-velocity
value=-0.00259e-08, at node 100514
min=-3.822e-09, at node 100530

f = 100 Hz

Acoustic analysis of a simplified compa
Time = 200
Contours of Z-velocity
value=-3.938e-08, at node 110523
min=-6.960e-08, at node 111154

f = 200 Hz

Acoustic analysis of a simplified compa
Time = 400
Contours of Z-velocity
value=-3.532e-08, at node 120530
min=-8.034e-09, at node 120514

f = 400 Hz

Acoustic analysis of a simplified compa
Time = 500
Contours of Z-velocity
value=-3.583e-08, at node 130523
min=-8.950e-08, at node 131154

f = 500 Hz
Introduction

To solve an interior acoustic problem by variational indirect BEM, collocation BEM and FEM. The cylinder duct is excited by harmonic nodal force at one end.

*Diameter 10mm, length 31.4mm*

This edge is fixed in x, y, z translation dof.

Nodal force 0.01N is applied for frequency range of 10-20000 Hz.

*FREQUENCY_DOMAIN_SSD
*FREQUENCY_DOMAIN_ACOUSTIC_BEM
or
*FREQUENCY_DOMAIN_ACOUSTIC_FEM
FEM Model

BEM Model

dB at Point 1

dB at Point 2
Acoustic pressure distribution
(by d3acs)

FEM acoustic analysis following SSD ana
Time = 5000
Contours of Z-velocity
min=0.00655163, at node# 108
max=62.7459, at node# 1183

Fringe Levels
5.275e+01
4.747e+01
4.220e+01
3.692e+01
3.165e+01
2.638e+01
2.110e+01
1.583e+01
1.055e+01
5.281e+00
6.592e-03

f = 5000 Hz

FEM acoustic analysis following SSD ana
Time = 10000
Contours of Z-velocity
min=0.119307, at node# 2538
max=166.84, at node# 1057

Fringe Levels
1.668e+02
1.502e+02
1.335e+02
1.168e+02
1.002e+02
8.348e+01
6.681e+01
5.014e+01
3.346e+01
1.679e+01
1.193e-01

f = 10000 Hz

FEM acoustic analysis following SSD ana
Time = 15000
Contours of Z-velocity
min=0.135217, at node# 2945
max=80.3927, at node# 2045

Fringe Levels
8.039e+01
7.237e+01
6.434e+01
5.631e+01
4.829e+01
4.026e+01
3.224e+01
2.421e+01
1.619e+01
8.159e+00
1.332e-01

f = 15000 Hz

FEM acoustic analysis following SSD ana
Time = 20000
Contours of Z-velocity
min=0.0380989, at node# 2552
max=63.3429, at node# 29

Fringe Levels
6.334e+01
5.701e+01
5.068e+01
4.435e+01
3.802e+01
3.169e+01
2.536e+01
1.903e+01
1.270e+01
6.369e+00
3.809e-02

f = 20000 Hz
CONCLUSION & FUTURE DEVELOPMENTS
A set of frequency domain features have been implemented, towards NVH, durability analysis of vehicles and other vibration and acoustic analysis

- Frequency Response Function
- Steady State Dynamics
- Random Vibration and Fatigue
- BEM & FEM Acoustics
- Response spectrum analysis

**Future work**

- SEA method for high frequency acoustics
- Fast multi-pole BEM for acoustics
- Infinite FEM
- Fatigue analysis with strains
- Feedbacks and suggestions from users

Thank you!