Summary of Support tube R&D

KEK  H. Yamaoka
Introduction

- Components at IR region
  - Supported by Tungsten tubes, CFRP tube
  - For high luminosity
  - Ground motion, culture Noise

- Analyses, Excitation tests

Study Items:
- Consistency of analyses
- How much is relative amplitude?
  - $|P1-P2| < 1\text{nm}$(Criteria)
- Is necessary CFRP tube?
- How much thickness?
Exciting Test

(1) 20t x 100w x 695L

(2) 20t x 100w x 1440L

(3) 20t x 100w x 695L

(4) 20t x 100w x 695L

(5) 80 x 10t x 200L (1/10 scale)

(6) 80 x 10t (1/10 scale)

Input exciting force
(Excitation table or Impact hammer)

Measurement: natural frequencies

Mode shape

Compare to the FEM results
Support structure should be modeled in FEM.

1st ~ 3rd mode: Good agreement with FEM.
Case-1

- 2.5mm thick plate
- Stiffness: 1:512
- 1st: 99Hz
- 2nd: 548Hz

Case-2

- 20mm thick plate
- 1st: 106Hz
- 2nd: 548Hz

By connecting very weak structure, Deviation can be absorbed. Correlation can be given. Relative amplitude can be estimated.
Calculation of relative amplitude

(Model-A)

\[ F_0 \cos(\omega t) = (m \cdot a) \sin(\omega t) \]

\[ \omega = 0 - 1000 \text{Hz} \]

QC-L: Tungsten (100mm)
QC-R: Tungsten (100mm)

CFRP

Difference of 1st mode of resonant frequency between QC-R and QC-L.

Get relative amplitude

(QC-R) - (QC-L)

(Model-B)

F_0 \cos(\omega t)

CFRP thick.

<table>
<thead>
<tr>
<th>CFRP thick.</th>
<th>3mm</th>
<th>5mm</th>
<th>10mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0Hz</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1Hz</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3Hz</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5Hz</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Estimation of Input Acc.

Data: Vertical @ATF(17:00 Feb. 10, 2004)

- Linear Spectrum
  - Input data
    - Input Acc. = 2x10^-7 m/s^2
    - Mass = 90 tons / 9.8 [m/s^2]
  - Self weight

- 2x10^-7 m/s^2
Relative amplitude between QC-R and QC-L

In case of 100mm-5mm(CFRP)-100mm, Δf=0Hz

1st mode

2nd mode

Amplitude: QC-R and QC-L

Same phase

Opposite phase

Relative amplitude between QC-R and QC-L
Other calculations

Diff: Relative amplitude between QC-R and QC-L.

<table>
<thead>
<tr>
<th>Δf</th>
<th>Mode</th>
<th>3-Point fixed(Both end+3.85m)</th>
<th>2-Point fixed(Both end)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>CFRP</td>
<td>Tungsten</td>
</tr>
<tr>
<td>0Hz</td>
<td>1st</td>
<td>Freq.(Hz)</td>
<td>75.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.098</td>
</tr>
<tr>
<td></td>
<td>2nd</td>
<td>Freq.(Hz)</td>
<td>77.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.213</td>
</tr>
<tr>
<td>1Hz</td>
<td>1st</td>
<td>Freq.(Hz)</td>
<td>75.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.129</td>
</tr>
<tr>
<td></td>
<td>2nd</td>
<td>Freq.(Hz)</td>
<td>77.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.174</td>
</tr>
<tr>
<td>3Hz</td>
<td>1st</td>
<td>Freq.(Hz)</td>
<td>73.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.173</td>
</tr>
<tr>
<td></td>
<td>2nd</td>
<td>Freq.(Hz)</td>
<td>76.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.094</td>
</tr>
<tr>
<td>5Hz</td>
<td>1st</td>
<td>Freq.(Hz)</td>
<td>71.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.204</td>
</tr>
<tr>
<td></td>
<td>2nd</td>
<td>Freq.(Hz)</td>
<td>76.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diff.(nm)</td>
<td>0.056</td>
</tr>
<tr>
<td>Canti</td>
<td>1st</td>
<td>Freq.(Hz)</td>
<td>75.6</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Amp(p-p)</td>
<td>0.224</td>
</tr>
</tbody>
</table>
Results

(Model-A)

Natural frequency: 76Hz, relative amp. : 0.1nm <1nm
This is ideal configuration.
In case of no CFRP tube(Cantilever): Amplitude=0.2nm <1nm
CFRP tube is not necessary because of less than 1nm.
However, it is difficult to amount on a very stiff base stand. So actual natural frequency must be lower than this value.

(Model-B)

Natural frequency: 17Hz, relative amp. : 2〜3nm
In case of no CFRP tube(Cantilever): Amplitude= 4nm

- CFRP tube:
  - No efficient to reduce amplitude.
  - Deviation of natural frequency between two tubes can be absorbed.
Optimization of CFRP tube thickness

Right side: 70Hz

Left side: 75Hz

CFRP: Changed!

<table>
<thead>
<tr>
<th>CFRP(mm)</th>
<th>1st mode Freq(Hz)</th>
<th>2nd mode Freq(Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>73.6</td>
<td>85.2</td>
</tr>
<tr>
<td>10</td>
<td>72.9</td>
<td>80.1</td>
</tr>
<tr>
<td>5</td>
<td>75.5</td>
<td>78.4</td>
</tr>
<tr>
<td>3</td>
<td>72.0</td>
<td>76.5</td>
</tr>
<tr>
<td>1</td>
<td>71.5</td>
<td>75.7</td>
</tr>
</tbody>
</table>

Less than this thickness, correlation and opposite phase doesn't appear at 2nd mode.

1st mode(CFRP: 1mm)

At least, thickness of CFRP: >3mm
Configuration of support system

- Both-ends supported structure with CFRP tube connection
  - Correlation is given to both-sides tubes in oscillating behavior.
    Tungsten tube: 100mm thick, CFRP: 5mm thick
    Support position: Both ends and 3.85m from I.P.

- Active vibration isolation system is necessary
  - Relative amplitude will be above 1nm. CFRP tube is not efficient to reduce amplitude less than 1nm.

- It is necessary to design the stiff support base as possible
  - Natural frequency becomes high.
  - Amplitude is decreased proportional to frequency.
Stiffness (CFRP tube)

- **Natural frequency**

\[ f_i = \frac{\lambda_i^2}{2\pi} \cdot \frac{1}{l^2} \sqrt{\frac{EI}{\rho A}} \]

- **Deformation**

\[ \omega = \frac{pl^3}{\alpha \cdot EI} \]

\( EI: \) Bending stiffness  
\( E: \) Young’s modulus  
\( I: \) Moment of Inertia

\[ I = \frac{\pi \cdot (d_1^4 - d_2^4)}{64} \]

**CFRP tube** (t=5mm)
- \( E = 150 \text{ GPa} \)
- \( I = \frac{(d_1^4 - d_2^4)}{64} \)
- \( = \frac{(800^4 - 790^4)}{64} \)
- \( = 9.9 \times 10^8 \text{ mm}^4 \)

**Tungsten** (t=100mm)
- \( E = 415 \text{ GPa} \)
- \( I = \frac{(d_1^4 - d_2^4)}{64} \)
- \( = \frac{(800^4 - 600^4)}{64} \)
- \( = 1.4 \times 10^{10} \text{ mm}^4 \)

**Ratio** = CFRP: Tungsten = **1 : 39**

**CFRP Tube**
- Not efficient to increase natural frequency and decrease amplitude.
Tests (Hammering test)

FRF (Frequency Response Function)

\[ H_{ij}(f) = \frac{X_i(f)}{F_j(f)} \]

\(X_i\): Output Acc.
\(F_j\): Input force

Input

Output

FFT

FRF Table
Results
(Taper flange, 12-M6)

A: 57Hz
B: 129Hz
C: 585Hz
D: 1216Hz
E: 1690Hz
F: 2500Hz
Comparison with FEM

MAC (Modal Assurance Criteria)

\[ MAC_{rr'} = \left| \frac{\psi_{r}^{test} \psi_{r'}^{test}}{\psi_{r}^{FE} \psi_{r'}^{FE}} \right|^2 \]

MAC = 1: Mode shape pairs is exactly match
MAC = 0: pairs that are completely independent

Damping ratio

\[ \zeta = \frac{f_1 - f_2}{2 \times f_n} \]

<table>
<thead>
<tr>
<th>Mode</th>
<th>Freq.</th>
<th>Damping(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>30.4Hz</td>
<td>1.68</td>
</tr>
<tr>
<td>2</td>
<td>188Hz</td>
<td>0.422</td>
</tr>
<tr>
<td>3</td>
<td>419Hz</td>
<td>0.303</td>
</tr>
<tr>
<td>4</td>
<td>584Hz</td>
<td>0.113</td>
</tr>
<tr>
<td>5</td>
<td>992Hz</td>
<td>8.02E-2</td>
</tr>
</tbody>
</table>

FEM Test

1st: 31Hz 1st: 30Hz
2nd: 192Hz 2nd: 188Hz
3rd: 457Hz 3rd: 419Hz
4th: 534Hz 4th: 584Hz
5th: 1022Hz 5th: 992Hz
**Harmonic analysis**

\[ F_0 \sin(\omega t) = (m \cdot a) \sin(\omega t) \]

- **F_0\sin(\omega t):** Excitation force
- **\omega = 0 – 1000Hz:** Sweep frequency

\[ m \ddot{x} + c \dot{x} + kx = F_0 \sin \omega t \]

\[
\frac{X}{X_{st}} = \frac{1}{\sqrt{1 - (\omega/\omega_n)^2} + (2\zeta \omega/\omega_n)^2}}
\]

- **F_0:** Input force \((F_0 = ma)\)
- **X:** Amplitude
- **X_{st}:** Static deformation
- **\zeta:** Damping ratio (2%)
- **\omega_n:** Resonant frequency
- **\omega:** Frequency

**If** \(\omega = \omega_n, \frac{X}{X_{st}} = 25\)**
<table>
<thead>
<tr>
<th></th>
<th>Model-1A</th>
<th>Model-1B</th>
<th>Model-2A</th>
<th>Model-2B</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Deformation(mm)</strong></td>
<td>1.6</td>
<td>-</td>
<td>0.09</td>
<td>-</td>
</tr>
<tr>
<td><strong>Stress(MPa)</strong></td>
<td>23</td>
<td>-</td>
<td>5</td>
<td>-</td>
</tr>
<tr>
<td><strong>Natural frequency(Hz)</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1st mode</td>
<td>17</td>
<td>15</td>
<td>71</td>
<td>15</td>
</tr>
<tr>
<td>2nd Mode</td>
<td>81</td>
<td>38</td>
<td>179</td>
<td>54</td>
</tr>
<tr>
<td>3rd mode</td>
<td>173</td>
<td>105</td>
<td>202</td>
<td>93</td>
</tr>
<tr>
<td><strong>Harmonic response(nm) @QC1</strong></td>
<td>8.0</td>
<td>8.0</td>
<td>0.2</td>
<td>6.0</td>
</tr>
<tr>
<td><strong>Spectrum analysis(nm) @QC1</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1st mode</td>
<td>6.5</td>
<td>2.0</td>
<td>4.3</td>
<td>2.7</td>
</tr>
<tr>
<td>2nd Mode</td>
<td>-1.7</td>
<td>1.1</td>
<td>0.2</td>
<td>0.2</td>
</tr>
<tr>
<td>3rd mode</td>
<td>-0.4</td>
<td>0.1</td>
<td>1.9</td>
<td>0.002</td>
</tr>
</tbody>
</table>
現実に考えられる固有振動数のずれについて

Young’s modulusの違い
Aluminumの場合種類によって
6.9GPa – 7.3GPaの範囲がある。
$\circ \ f_i$: 3% different
If 70Hz: 2Hz, 17Hz: 16.5Hz

寸法の誤差
寸法が多少ずれても断面2次モーメント$I$に大きな差は生まれない。
また、寸法を管理することができることから「寸法の誤差」による固有振動数のずれは小さいと考えられる。

組み立て及び設置誤差によるずれ
ネジ締めのトルク管理、寸法測定等で管理できるが完全にはできない。
3%以内の誤差にできるのではないだろうか。（根拠はすこしあいまい）

全体では5%ぐらいに抑えられるかもしれない。
$\circ$ どのくらいの誤差に収められるか試験。