Estructuras de sistemas espaciales

Definición de cargas y análisis

Curso 2022-2023

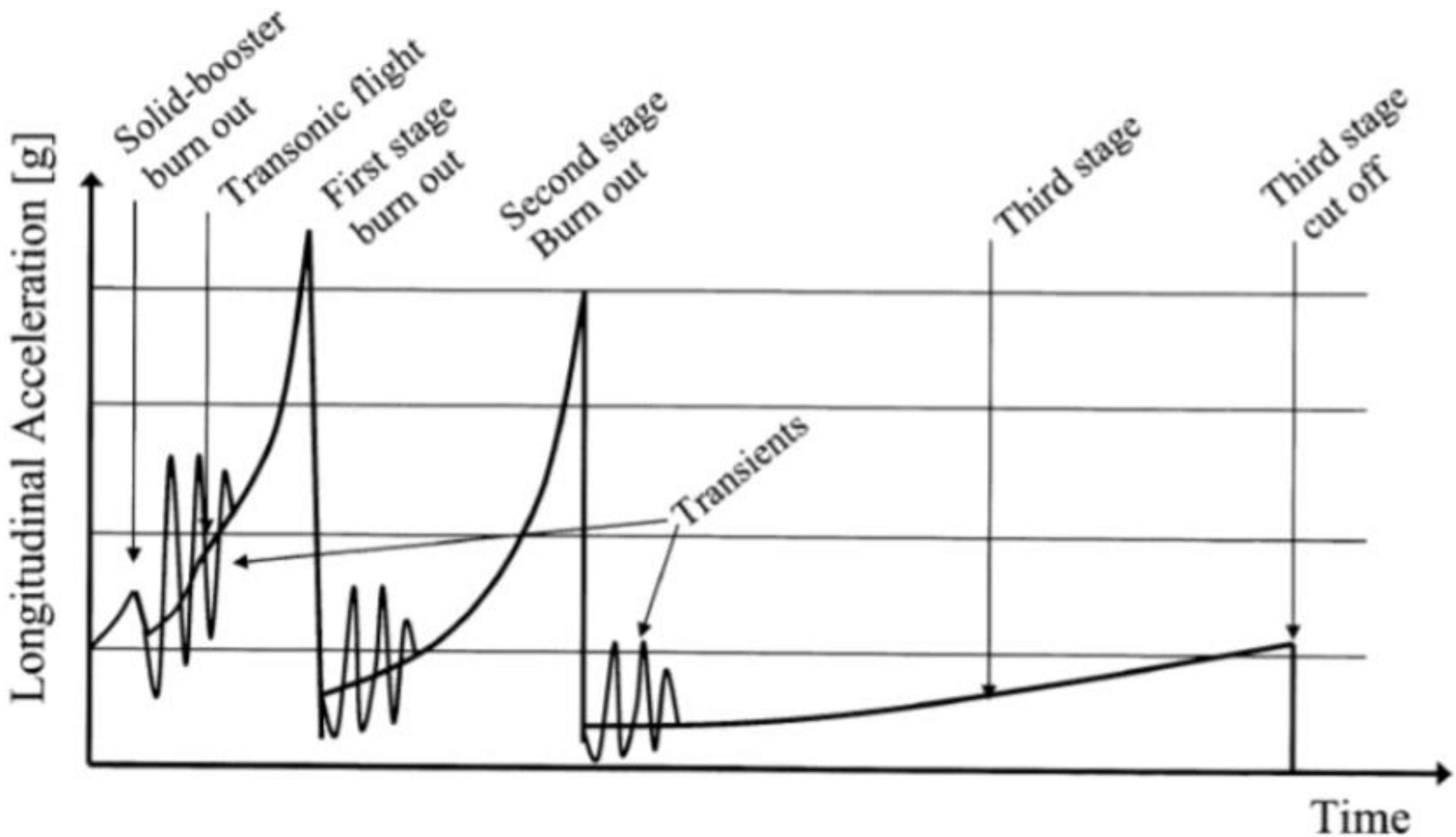
CARGAS DEBIDAS AL ENTORNO (LANZAMIENTO)

- Aceleraciones quasi-estáticas
- Cargas dinámicas de baja frecuencia
- Cargas dinámicas de ancho espectro:
  - Aleatorias (estructurales)
  - Acústicas
- Choque

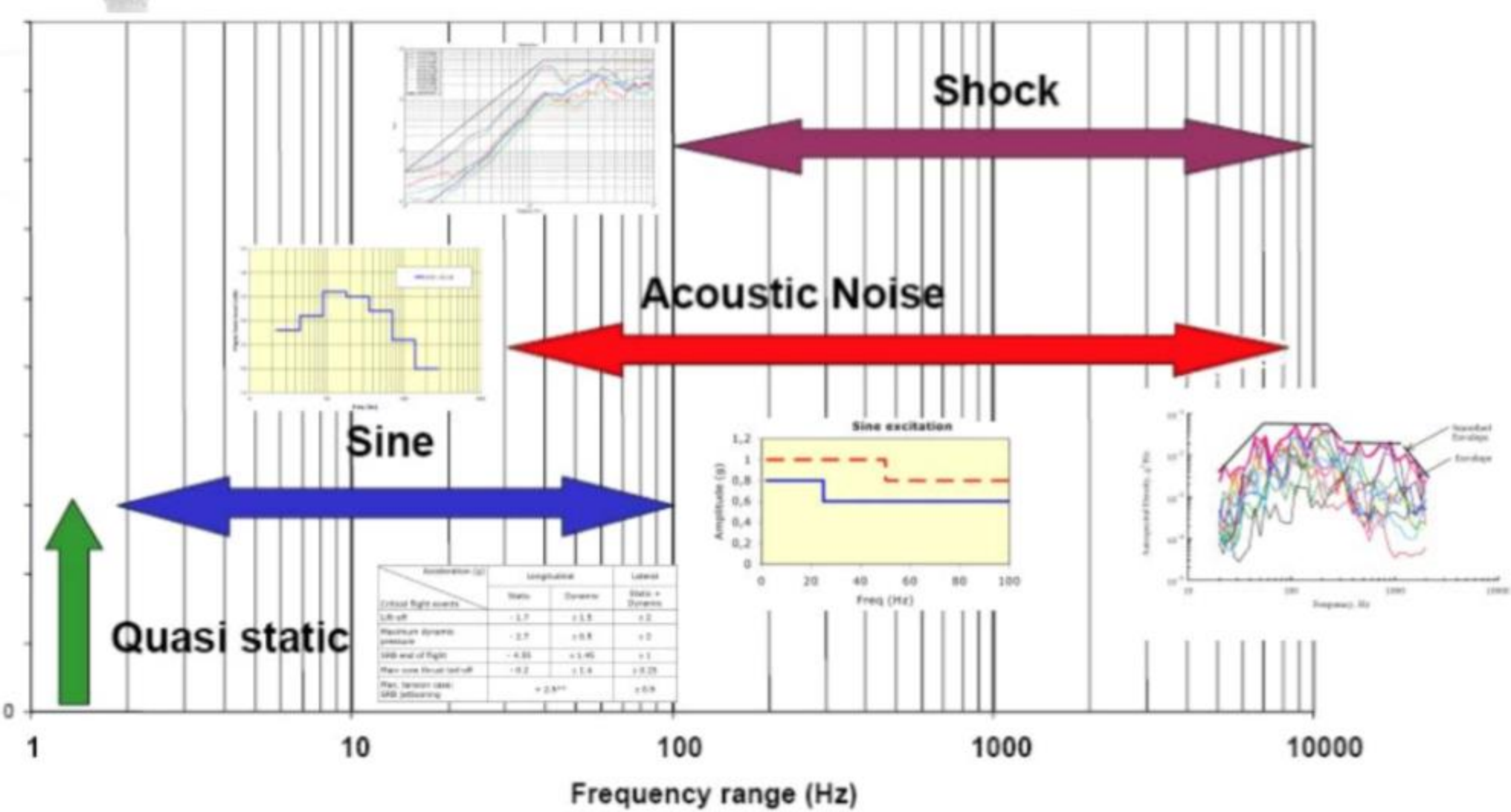
➢ Las cargas estructurales se transmiten a través de la interfaz S/C-lanzador

➢ Las cargas acústicas inciden directamente sobre la carga.

STEADY-STATE AND LOW FREQUENCY TRANSIENT ACCELERATIONS



STATIC AND DYNAMIC ENVIRONMENT SPECIFICATION



QUASI-STATIC LOADS (ACCELERATION)

- ‘Loads independent of time or which vary slowly, so that the dynamic response of the structure is not significant’ (ECSS-E-ST-32). Note: This is the definition of a quasi-static event.
- ‘Combination of static and low frequency loads into an equivalent static load specified for design purposes as C.o.G. acceleration’ (NASA RP-1403, NASA-HDBK-7004). Note: This definition is fully adequate for the design of the spacecraft primary structure. For the design of component the contribution of the high frequency loads, if relevant, is included as well.

- CONCLUSION: **quasi-static loading** means under **steady-state accelerations** (unchanging applied force balanced by inertia loads). For design purposes (e.g. derivation of design limit loads, selection of the fasteners, etc), the quasi-static loads are normally calculated by combining both static and dynamic load contribution. In this context the quasi-static loads are equivalent to (**or interpreted by the designer** as) **static loads**, typically **expressed as equivalent accelerations at the C.o.G**.

CARGAS DINÁMICAS DE BAJA FRECUENCIA

QM Sine Loads

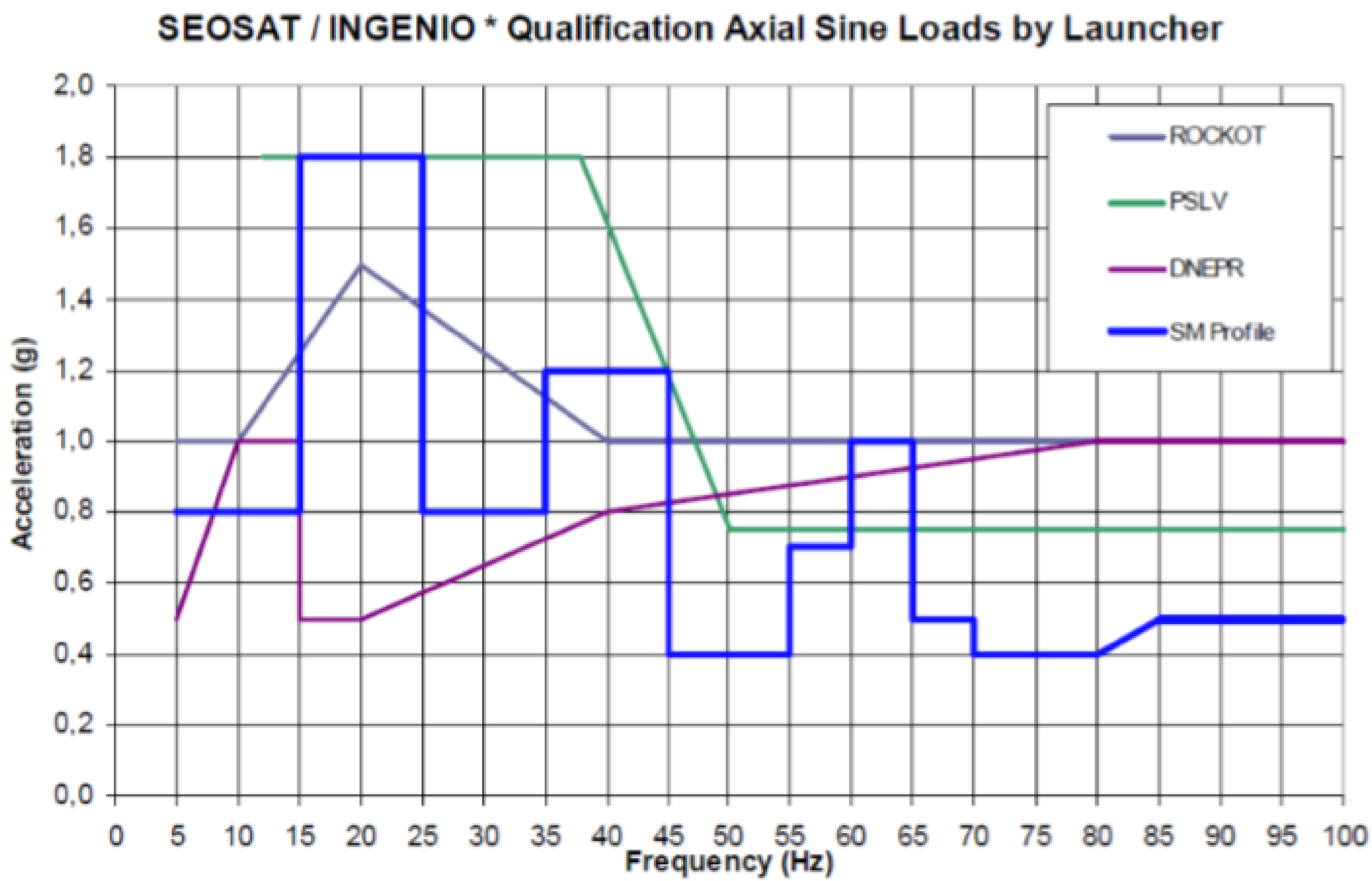


Figure 9.7.2.1: Sine levels from launcher candidates (PSLV, ROCKOT, DNPR). AXIAL excitation (Z AXIS)

CARGAS DINÁMICAS DE BAJA FRECUENCIA

QM Sine Loads

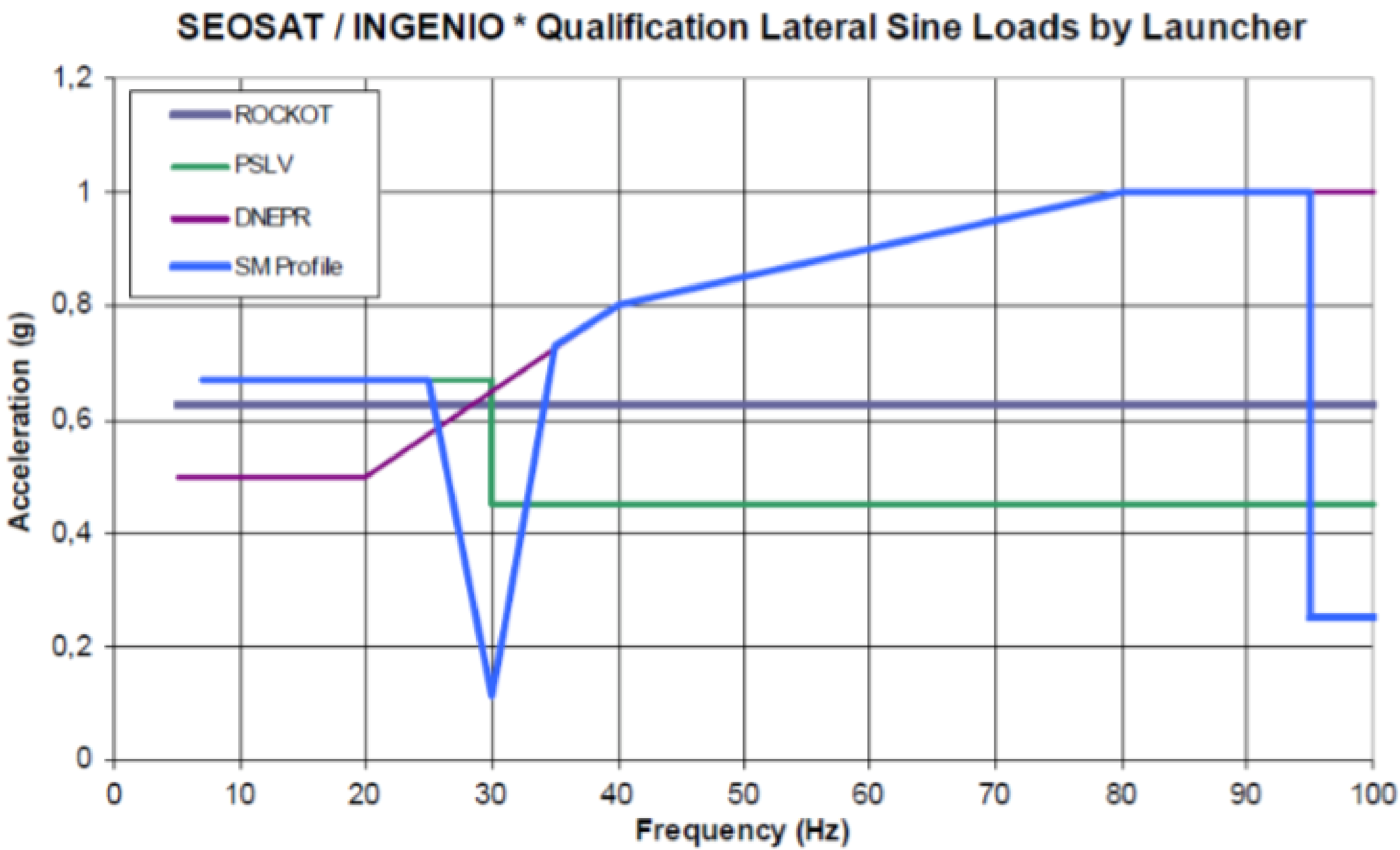
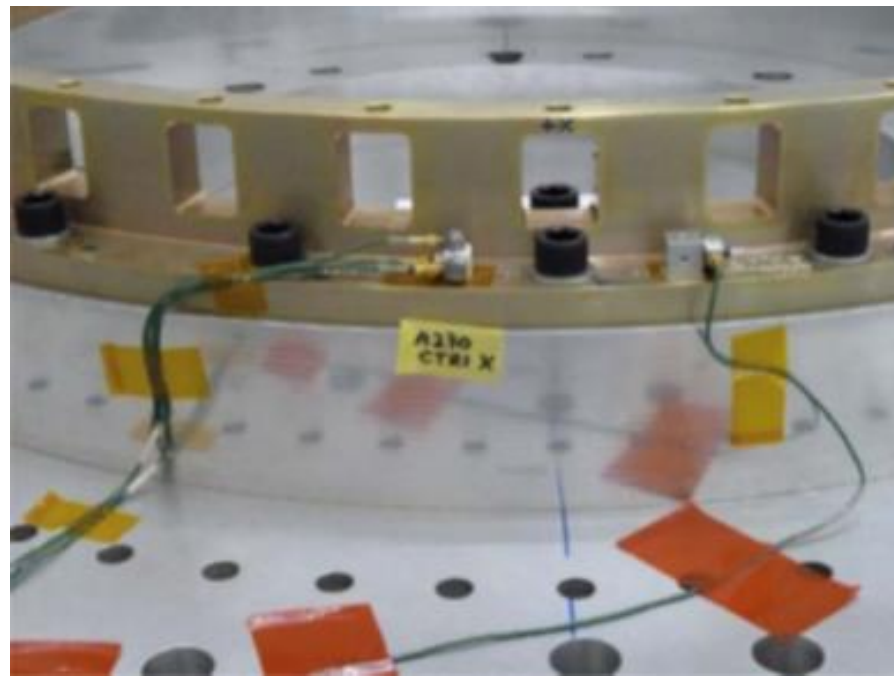


Figure 9.7.2.2: Sine levels from launcher candidates (PSLV, ROCKOT, DNPR). LATERAL excitation (X & Y AXIS)

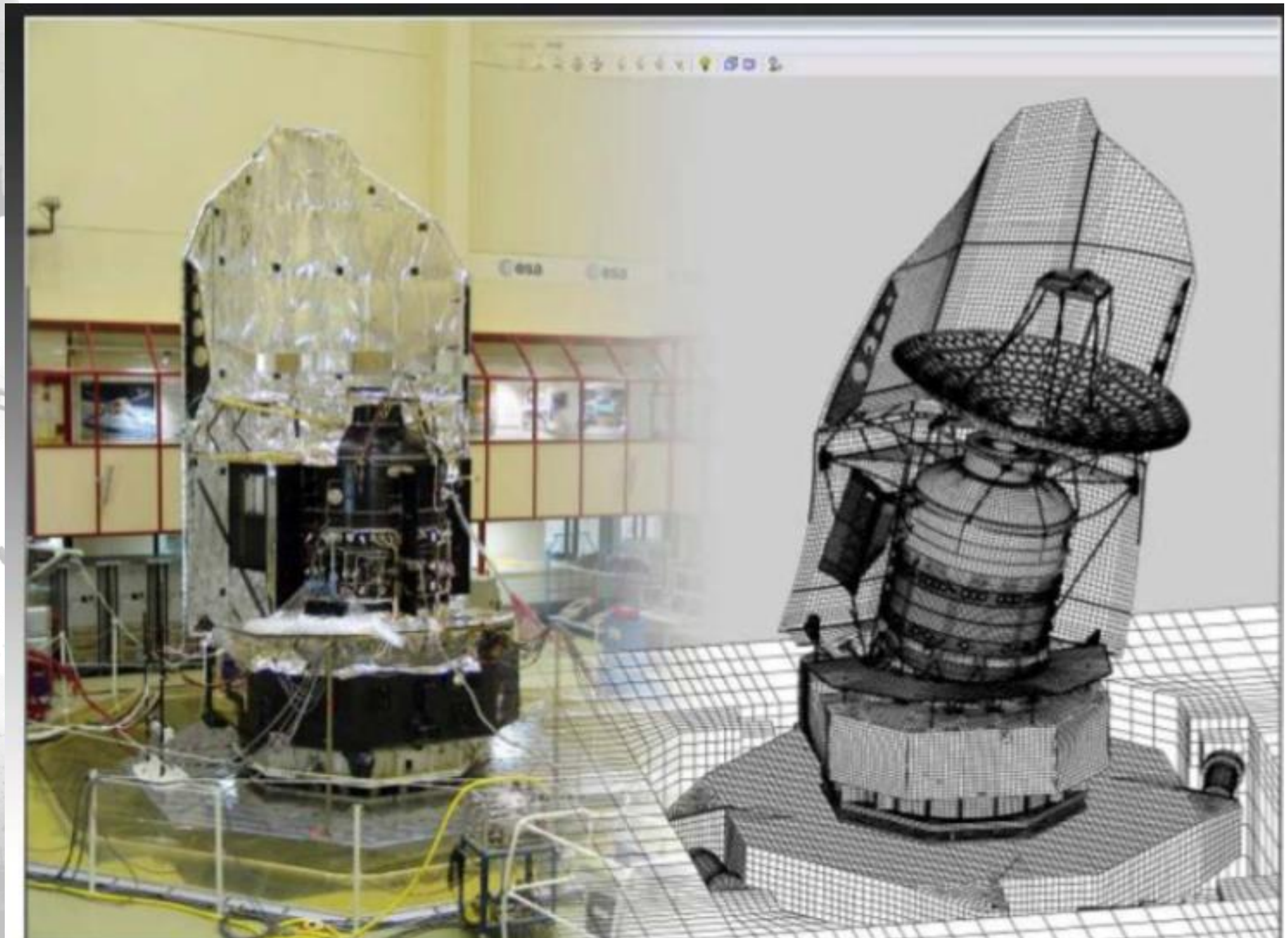
CARGAS DINÁMICAS DE BAJA FRECUENCIA

QM Sine Loads



CARGAS DINÁMICAS DE BAJA FRECUENCIA

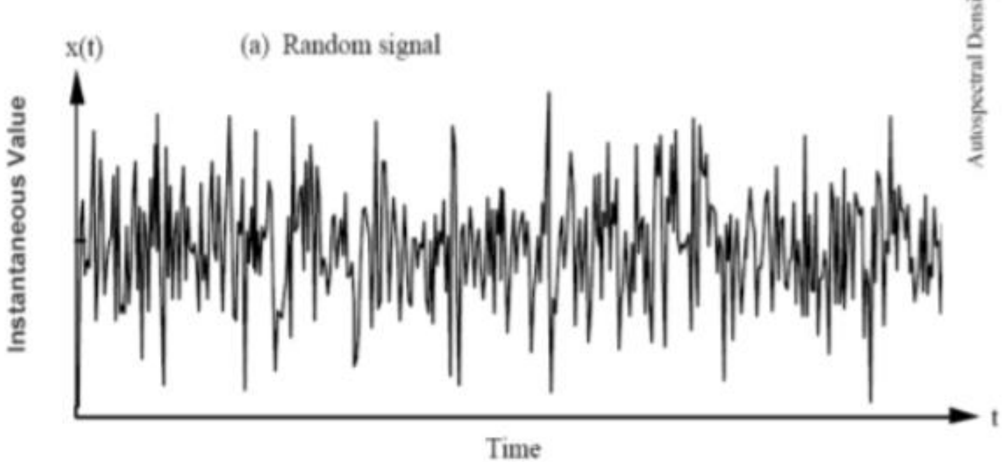
HYDRA shaker table (ESTEC test centre)

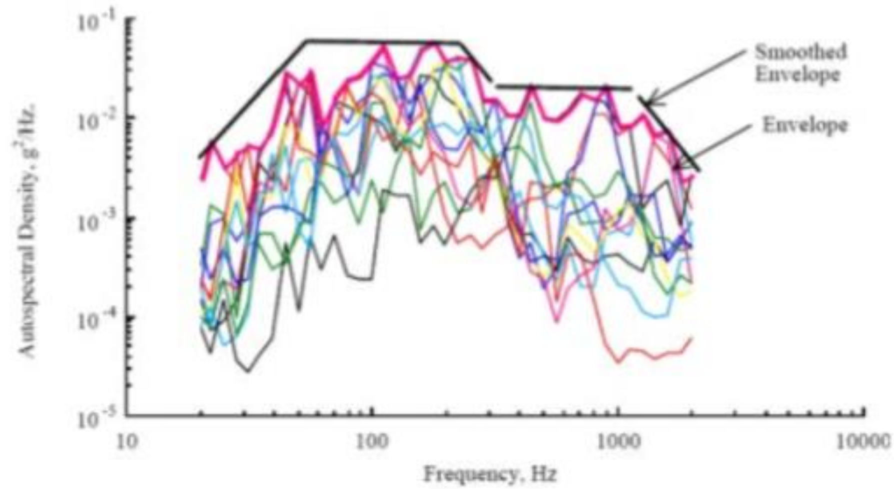


BROAD BAND AND HIGH FREQUENCY VIBRATIONS:

Broad band random vibrations are produced by:

- Engines functioning
- Structural response to broad band acoustic loads
- Aerodynamic turbulence



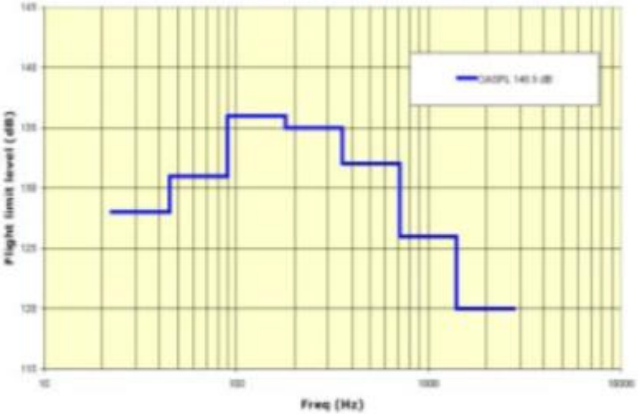


ACOUSTIC LOADS

- During the lift off and the early phases of the launch an extremely high level of acoustic noise sorrounds the payload

- The principal sources of noise are:
  - Engine functioning
  - Aerodynamic turbulence

- Acoustic noise (as pressure waves) impinging on light-weight panel-like structures produce high response.



ACOUSTIC LOADS

- Usually expresed in terms of the sound pressure level:

$$\text{SPL}=10\log\left(\frac{p^2}{{p_{ref}}^2}\right)$$

- Logarithmic frequency bands:

$$\left.\begin{aligned} \Delta f &= f_{max} - f_{min} \\ \frac{f_{max}}{f_{min}} &= 2^x \\ f_c &= \sqrt{f_{max} f_{min}} \end{aligned}\right\} \Delta f = \left(2^{\frac{x}{2}} - 2^{-\frac{x}{2}}\right) f_c$$

- Exact frequency bands vs Standard frequency bands

ACOUSTIC LOADS

TABLE 4.2

Comparison of 1-octave and ⅓-octave bands

1 Octave			⅓ Octave		
Lower cutoff frequency (Hz)	*Center frequency (Hz)*	*Upper cutoff frequency (Hz)*	*Lower cutoff frequency (Hz)*	*Center frequency (Hz)*	*Upper cutoff frequency (Hz)*
11	16	22	14.1	16	17.8
			17.8	20	22.4
			22.4	25	28.2
22	31.5	44	28.2	31.5	35.5
			35.5	40	44.7
			44.7	50	56.2
44	63	88	56.2	63	70.8
			70.8	80	89.1
			89.1	100	112
88	125	177	112	125	141
			141	160	178
			178	200	224
177	250	355	224	250	282
			282	315	355
			355	400	447
355	500	710	447	500	562
			562	630	708
			708	800	891
710	1,000	1,420	891	1,000	1,122
			1,122	1,250	1,413
			1,413	1,600	1,778
1,420	2,000	2,840	1,778	2,000	2,239
			2,239	2,500	2,818
			2,818	3,150	3,548
2,840	4,000	5,680	3,548	4,000	4,467
			4,467	5,000	5,623
			5,623	6,300	7,079
5,680	8,000	11,360	7,079	8,000	8,913
			8,913	10,000	11,220
			11,220	12,220	14,130
11,360	16,000	22,720	14,130	16,000	17,780
			17,780	20,000	22,390

ACOUSTIC LOADS

Octave band centre frequency (Hz)	Acceptance level (dB)	Qualification level (dB)	Test tolerance
	0 dB: (ref. 2 x 10^{-5} Pascal)		
31.5	128	132	-2, +4
63	130	134	-1, +3
125	135	139	-1, +3
250	139	143	-1, +3
500	134	138	-1, +3
1000	128	132	-1, +3
2000	124	128	-1, +3
OVERALL	142	146	-1, +3
Test duration	1 minute	2 minutes	

TABLE 4.3: ARIANE 5 acoustic tests acceptance and qualification levels.
(Source: ARIANE 5 User's Manual, Issue 3)

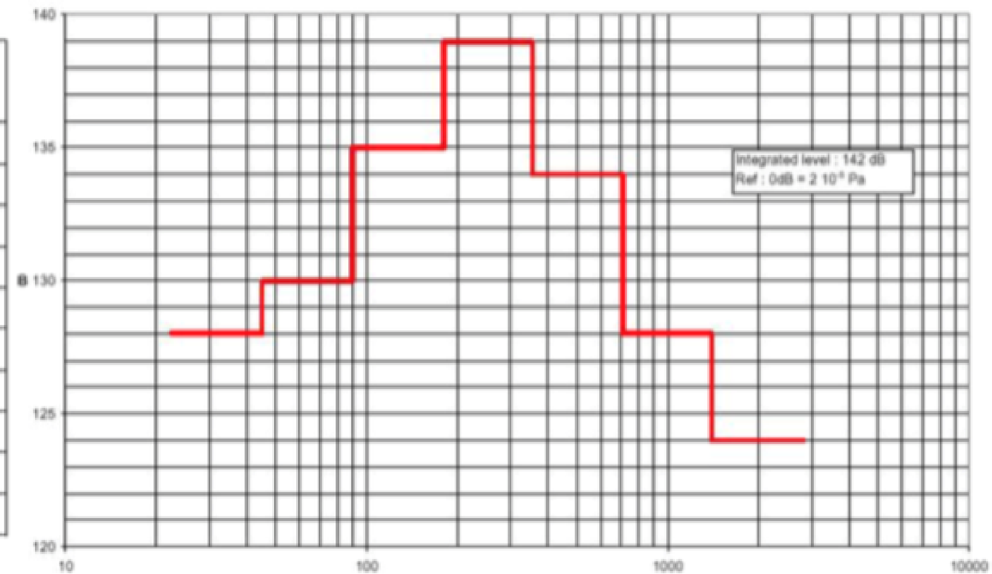
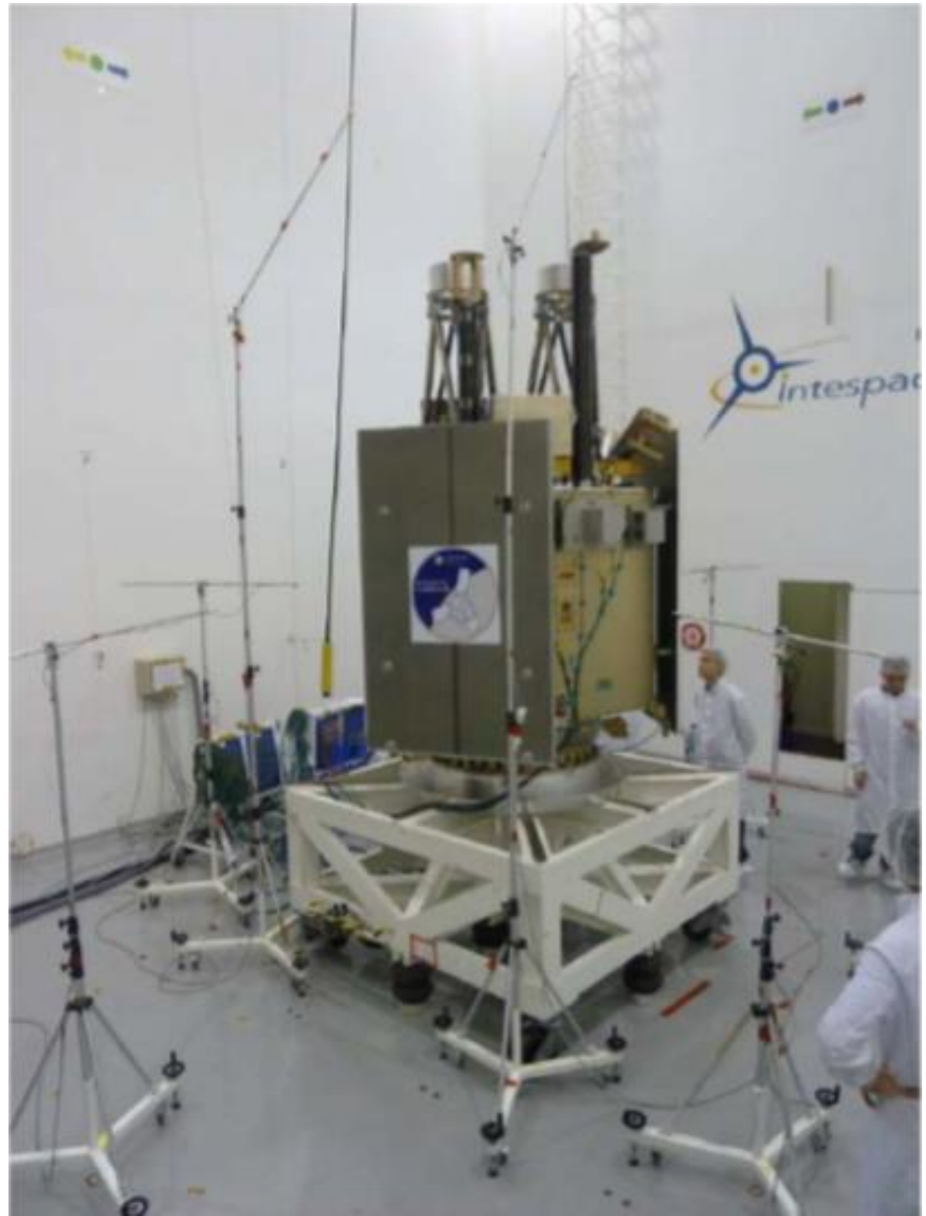


FIGURE 4.2: Acoustic environment inside the fairing for the ARIANE 5 launcher.
(Source: ARIANE 5 User's Manual, Issue 3)

EJEMPLOS:

QM Acoustic Loads



EJEMPLOS:

QM Acoustic Loads

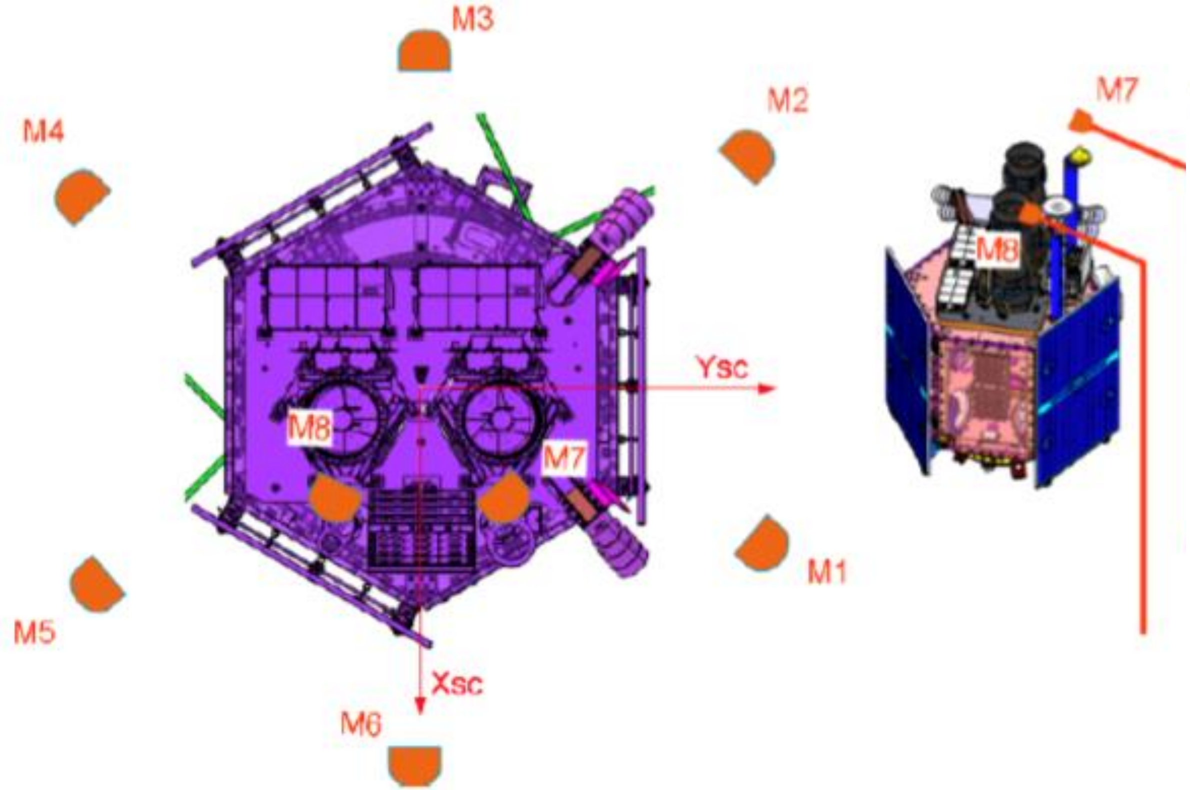
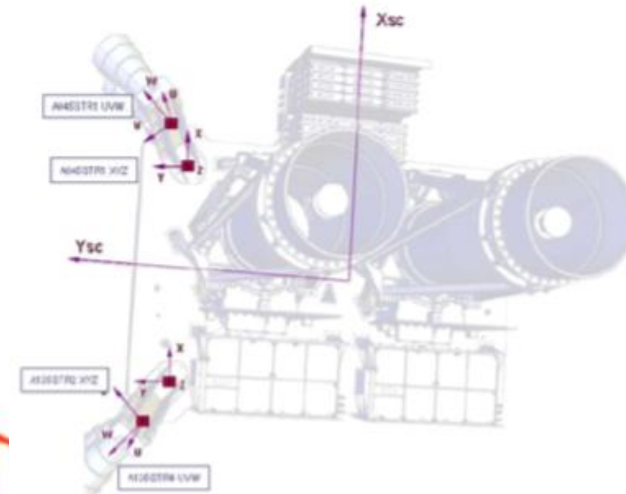
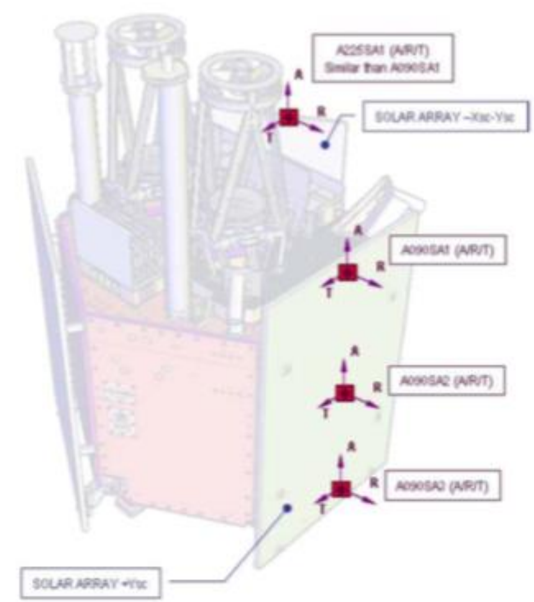


Figure 4 Microphones Disposition during Acoustic Test





EJEMPLOS:

QM Acoustic Loads

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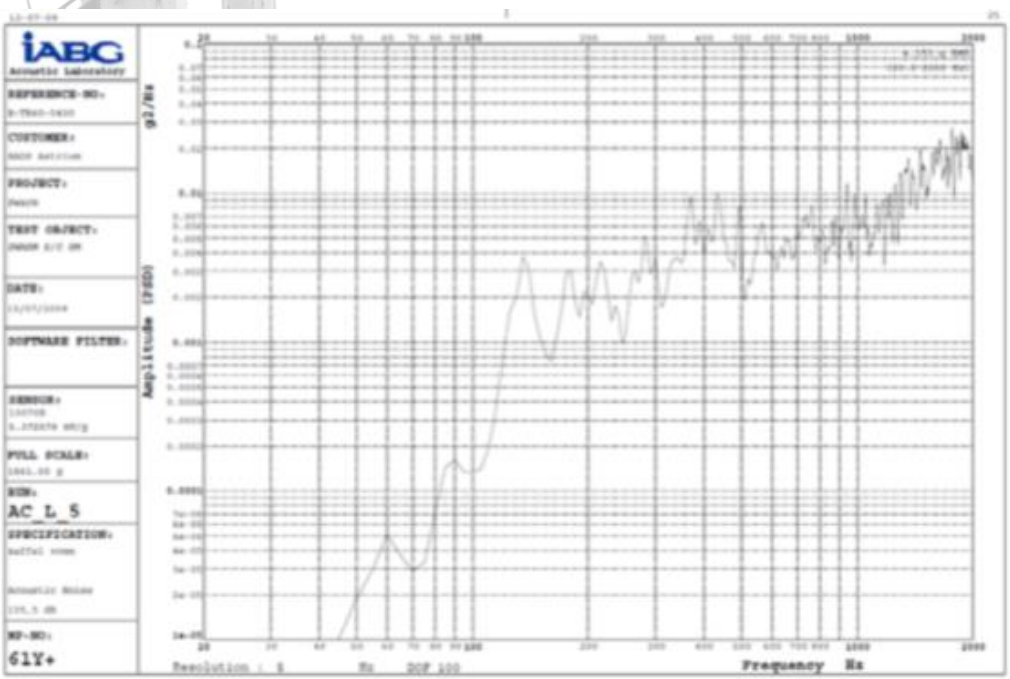
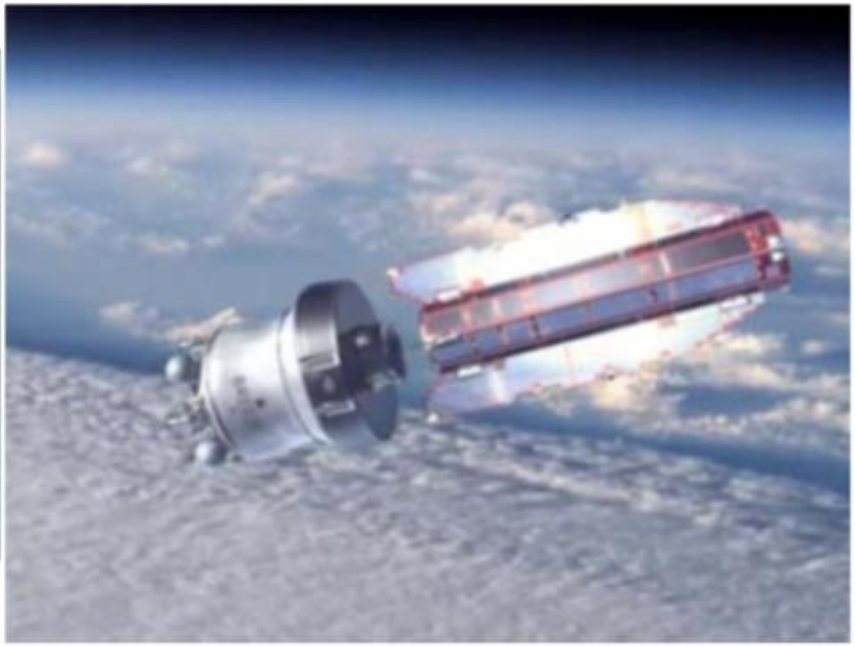


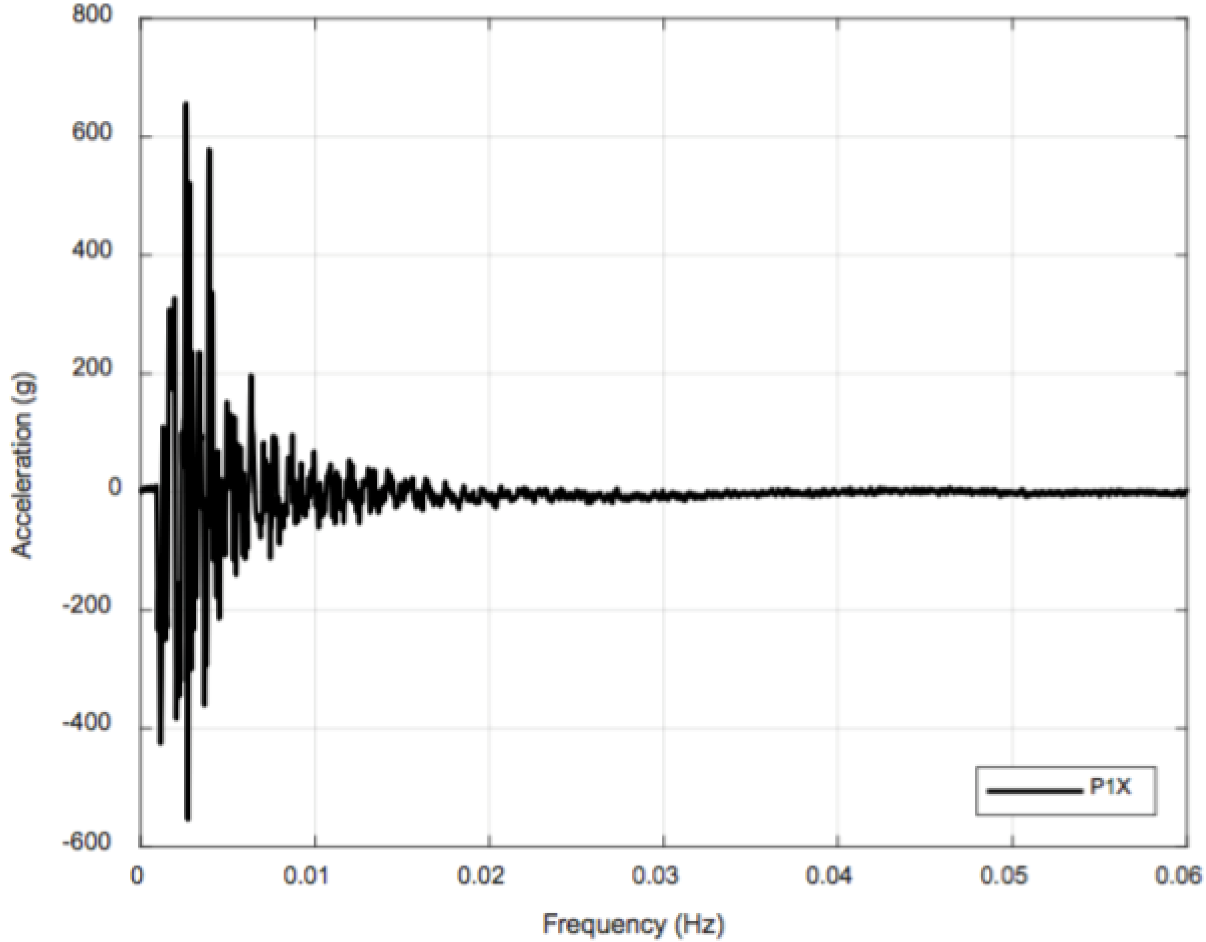
CHOQUE

Mainly caused by the actuation of pyrotechnic devices:

- Release mechanism for stage and satellite separation
- Deployable mechanism of solar arrays etc



CHOQUE



800
600
400
200
0
-200
-400
-600
Acceleration (g)
0
0.01
0.02
0.03
0.04
0.05
0.06
Frequency (Hz)
P1X

SHOCK LOADS ARIANE V

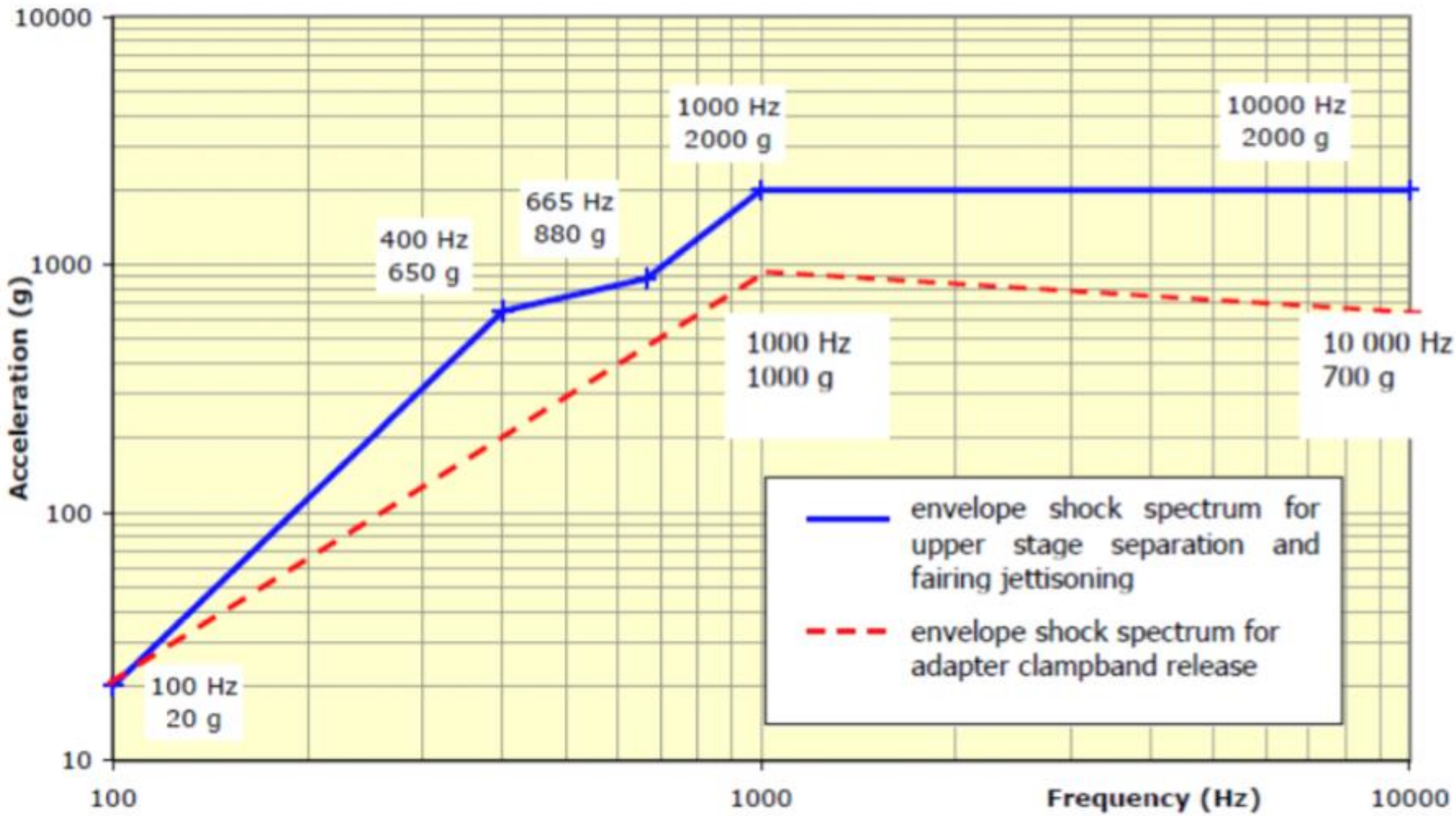


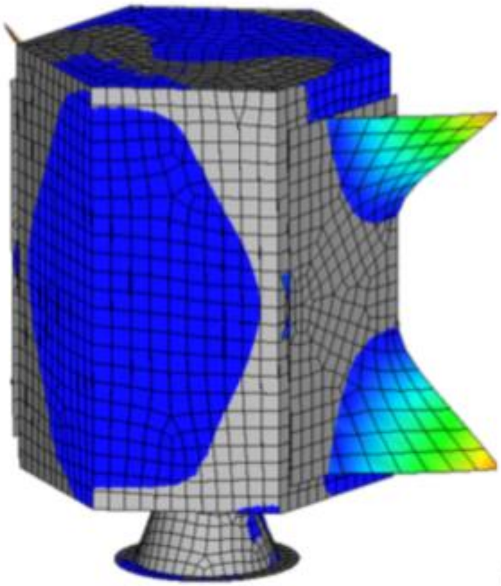
Figure 3.2.6.a – Envelope shock spectrum for the upper stage separation and fairing jettisoning and envelope shock spectrum for clampband release at spacecraft interface

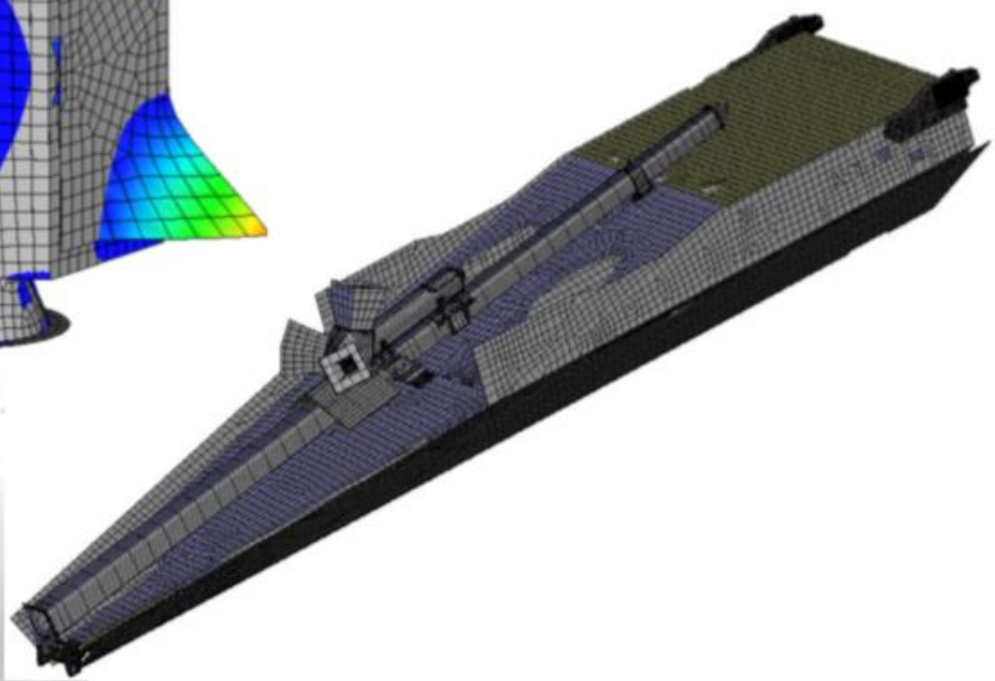
EJEMPLOS: Rockot dynamic specification

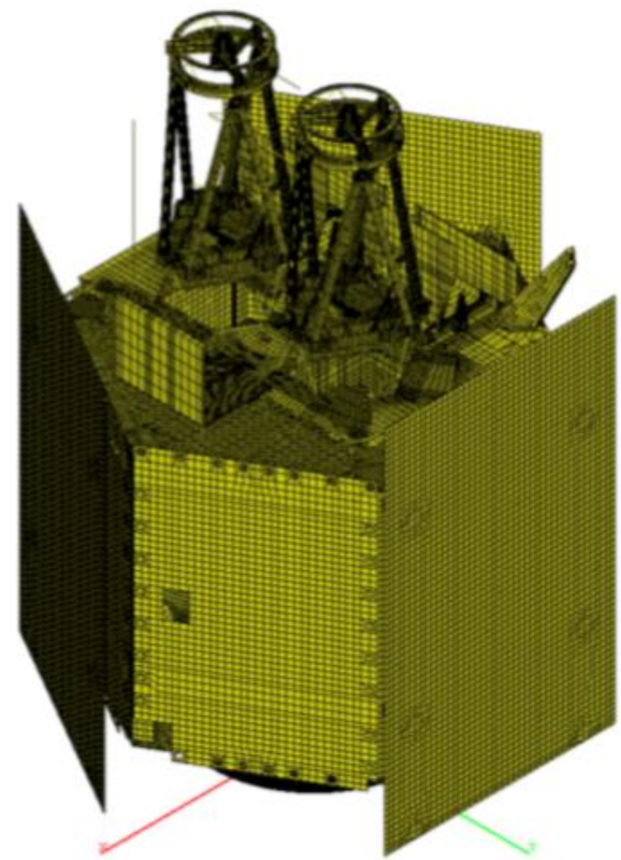
Environment	Level	
Sine vibration	**Longitudinal**= 1 g on [5-10] Hz 1.5 g at 20 Hz 1 g on [40-100] Hz	**Lateral** = 0.625 g on [5-100] Hz
Acoustic	31.5 Hz = 130.5 dB 63 Hz = 133.5 dB 125 Hz = 135.5 dB 250 Hz = 135.7 dB	500 Hz = 130.8 dB 1000 Hz = 126.4 dB 2000 Hz = 120.3 dB
Shock	100 Hz = 50 g 700 Hz = 800 g 1000 Hz → 1500 Hz = 2000 g	4000 Hz → 5000 Hz = 4000 g 10000 Hz = 2000 g

VERIFICACIÓN DEL DISEÑO MEDIANTE ANÁLISIS:

- Modelado por Elementos Finitos de la estructura







VERIFICACIÓN DEL DISEÑO MEDIANTE ANÁLISIS:

Problemas asociados a la verificación de modelos:

- Modelos dinámicos con un alto número de grados de libertad.
- Tiempos de cálculo y utilización de recursos muy elevado.
- Búsqueda de métodos de reducción de modelos.
- Habitualmente basados en la reducción modal.

REDUCCIÓN DE MODELOS:

- Reducción de las bases modales de las estructuras.
- Requisitos:
  - Desviación de las frecuencias naturales del modelo reducido ±3% respecto al modelo de referencia.
  - Desviación de las masas efectivas ±10%
- Métodos de reducción:
  - Static Condensation Method (Guyan)
  - Dynamic Condensation Method (Miller)
  - Improved Reduced System (IRS)
  - Craig-Bamption Reduction Method (CB)
  - Generalised Dynamic Reduction Method (GDR)
  - System Equivalent Reduction-Expansion Process (SEREP)
  - Ritz Vectors

MÉTODO DE REDUCCIÓN ESTÁTICA (GUYAN):

- Clasificación de grados de libertad en:
  - Grados de libertad MAESTROS $\{x_a\}$
  - Grados de libertad ESCLAVOS (eliminados) $\{x_e\}$
- Matrices del sistema (no amortiguado):

$$\begin{bmatrix} M_{aa} & M_{ae} \\ M_{ea} & M_{ee} \end{bmatrix} \begin{Bmatrix} \ddot{x}_a \\ \ddot{x}_e \end{Bmatrix} + \begin{bmatrix} K_{aa} & K_{ae} \\ K_{ea} & K_{ee} \end{bmatrix} \begin{Bmatrix} x_a \\ x_e \end{Bmatrix} = \begin{Bmatrix} F_a \\ F_e \end{Bmatrix}$$

- Hipótesis de aplicación:
  - Se desprecian efectos inerciales asociados a los GDL esclavos.
  - Se eliminan las fuerzas externas asociadas a los GDL esclavos:

$$\begin{bmatrix} M_{aa} & 0 \\ 0 & 0 \end{bmatrix} \begin{Bmatrix} \ddot{x}_a \\ \ddot{x}_e \end{Bmatrix} + \begin{bmatrix} K_{aa} & K_{ae} \\ K_{ea} & K_{ee} \end{bmatrix} \begin{Bmatrix} x_a \\ x_e \end{Bmatrix} = \begin{Bmatrix} F_a \\ 0 \end{Bmatrix}$$

MÉTODO DE REDUCCIÓN ESTÁTICA (GUYAN):

$$\begin{bmatrix} M_{aa} & 0 \\ 0 & 0 \end{bmatrix} \begin{Bmatrix} \ddot{x}_a \\ \ddot{x}_e \end{Bmatrix} + \begin{bmatrix} K_{aa} & K_{ae} \\ K_{ea} & K_{ee} \end{bmatrix} \begin{Bmatrix} x_a \\ x_e \end{Bmatrix} = \begin{Bmatrix} F_a \\ 0 \end{Bmatrix}$$

- Reducción de grados de libertad:

$$[K_{ea}]\{x_a\} + [K_{ea}]\{x_e\} = 0 \rightarrow \{x_e\} = -[K_{ee}]^{-1}[K_{ea}]\{x_a\}$$

$$\{x\} = \begin{Bmatrix} x_a \\ x_e \end{Bmatrix} = \begin{bmatrix} I \\ -[K_{ee}]^{-1}[K_{ea}] \end{bmatrix} \{x_a\} = \begin{bmatrix} I \\ G_{ea} \end{bmatrix} \{x_a\} = [T_{ea}]\{x_a\}$$

$$[\bar{M}_{aa}] = [T_{ea}]^T[M][T_{ea}] \;\; ; \;\; [\bar{K}_{aa}] = [T_{ea}]^T[K][T_{ea}]$$

MÉTODO DE REDUCCIÓN ESTÁTICA (GUYAN):

- Selección de los GDL maestros $\{x_a\}$

Criterio: Selección de los GDL asociados a mayores masas generalizadas.

$$\frac{1}{2p}\sqrt{\frac{k_{ii}}{m_{ii}}} \pounds 1.5 f_{\max}$$

donde:

k_{ii} son los términos diagonales de la matriz de rigidez (translación y rotación)

m_{ii} son los términos diagonales de la matriz de masas (translación y rotación)

$f_{\max}$ es la máxima frecuencia de interés

MÉTODO DE REDUCCIÓN ESTÁTICA (GUYAN):

- El problema de autovalores reducido quedaría:

$$\left\{\left[\bar{M}_{aa}\right]-\lambda_a\left[\bar{K}_{aa}\right]\right\}\left\{\phi_a\right\}=0$$

donde:

$\left\{\phi_a\right\}$ son los autovectores del problema de autovalores reducido

λ_a son los autovalores asociados a los autovectores (formas modales)

Las formas modales del sistema completo:

$$\left[\phi_{GR}\right]=\begin{bmatrix}\phi_a\\ \phi_e\end{bmatrix}=\begin{bmatrix}I\\ G_{ea}\end{bmatrix}\left[\phi_a\right]$$

MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):

- **Clasificación de grados de libertad en:**
  - Grados de libertad DE FRONTERA $\{q_f\}$
  - Grados de libertad INTERIORES $\{q_i\}$
- **Matrices del sistema (no amortiguado):**

$$\begin{bmatrix} M_{ff} & M_{fi} \\ M_{if} & M_{ii} \end{bmatrix} \begin{Bmatrix} \ddot{q}_f \\ \ddot{q}_i \end{Bmatrix} + \begin{bmatrix} K_{ff} & K_{fi} \\ K_{if} & K_{ii} \end{bmatrix} \begin{Bmatrix} q_f \\ q_i \end{Bmatrix} = \begin{Bmatrix} F_f \\ F_i \end{Bmatrix}$$

- **Hipótesis de aplicación:**

Se propone la clasificación de los desplazamientos en:

- Modos estáticos (*constraint modes*)
- Modos interiores elásticos con frontera fija

$$\{q\} = [\phi_s]\{\eta_f\} + [\phi_i]\{\eta_i\} = [\phi_s\,, \phi_i] \begin{Bmatrix} \eta_f \\ \eta_i \end{Bmatrix} = [\psi]\{\eta\}$$

MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):

- Cálculo de los modos estáticos (*constraint modes*)
  - Se desprecian efectos inerciales.
  - Se suponen nulas las fuerzas externas interiores.
  - Se impone un desplazamiento unitario en cada grado de libertad de frontera.

$$\begin{bmatrix} K_{ff} & K_{fi} \\ K_{if} & K_{ii} \end{bmatrix} \begin{Bmatrix} q_f \\ q_i \end{Bmatrix} = \begin{Bmatrix} R_f \\ 0 \end{Bmatrix}$$

$$\{q_i\} = -[K_{ii}]^{-1}[K_{if}]\{q_f\} \text{ con } \{q_f\} = [I]$$

Transformación estática:

$$\{q\} = \begin{Bmatrix} q_f \\ q_i \end{Bmatrix} = \begin{bmatrix} I \\ -[K_{ii}]^{-1}[K_{if}] \end{bmatrix} \{q_f\} = [\phi_s]\{q_f\}$$

MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):

- Cálculo de los modos elásticos interiores:
  - Se suponen fijos los grados de libertad de frontera.
  - Se calculan las formas modales:

$$|[K_{ii}] - \omega^2 [M_{ii}]| = \{0\}$$

$$([K_{ii}] - \omega^2 [M_{ii}])[\phi_{ii}] = \{0\}$$

Transformación modal:

$$\{q\} = \begin{Bmatrix} q_f \\ q_i \end{Bmatrix} = \begin{bmatrix} 0 \\ [\phi_{ii}] \end{bmatrix} \{\eta_i\} = [\phi_i]\{\eta_i\}$$

La matriz de transformación de Craig-Bampton será:

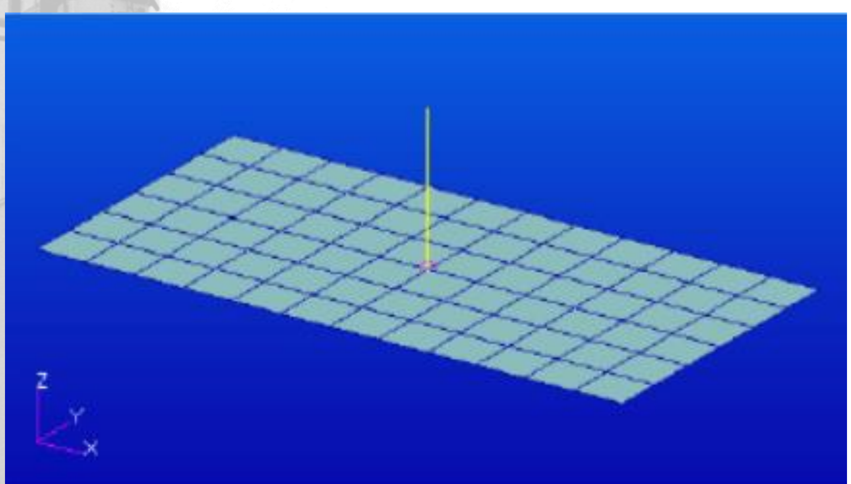
$$[\psi] = [\phi_s \, , \phi_i] = \begin{bmatrix} [I] & [0] \\ -[K_{ii}]^{-1}[K_{if}] & [\phi_{ii}] \end{bmatrix}$$

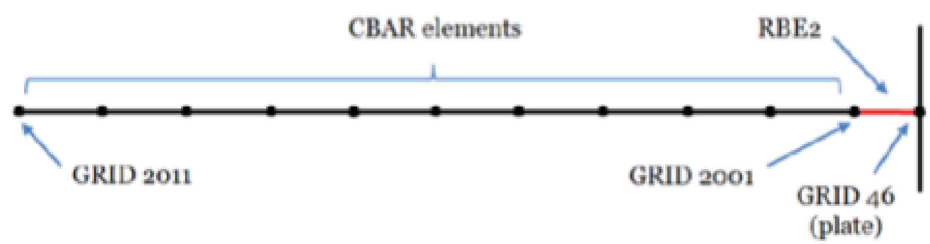
MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):

Placa de aluminio (1 x 0.5 m) y viga de aluminio (0.25)

Modelo FEM:

- Placa: 91 nodos, 546 DOF
- Viga: 11 nodos, 66 DOF





MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):

Modos interiores de la viga:

8 modos por debajo de 3 kHz

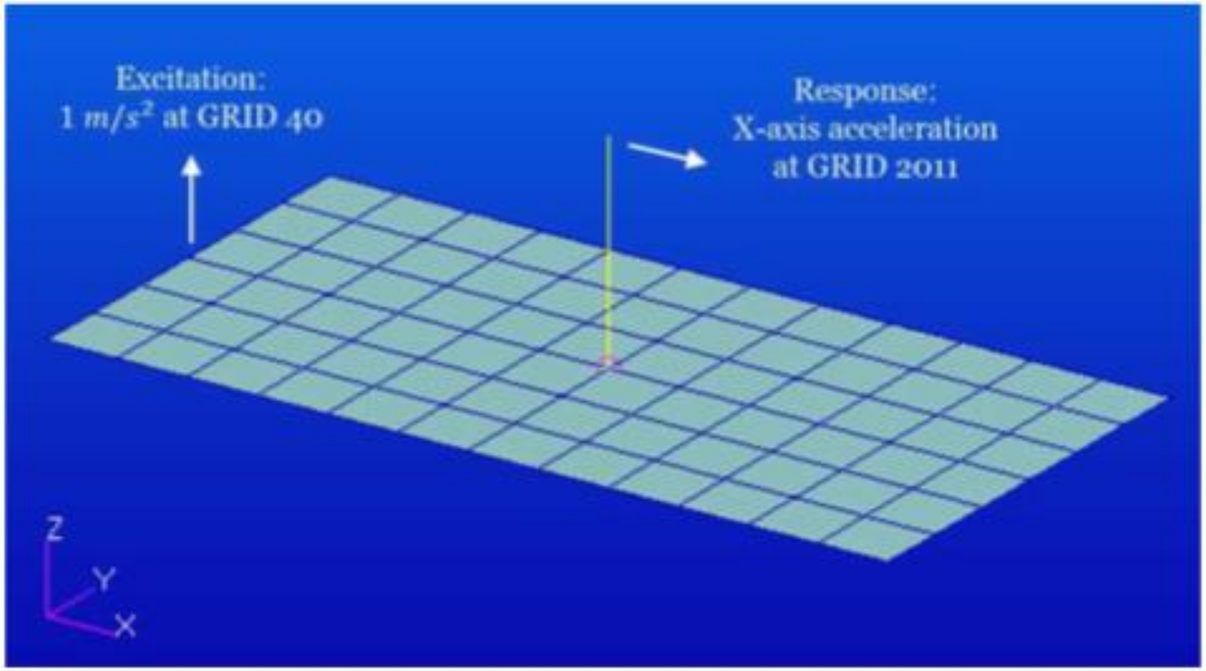
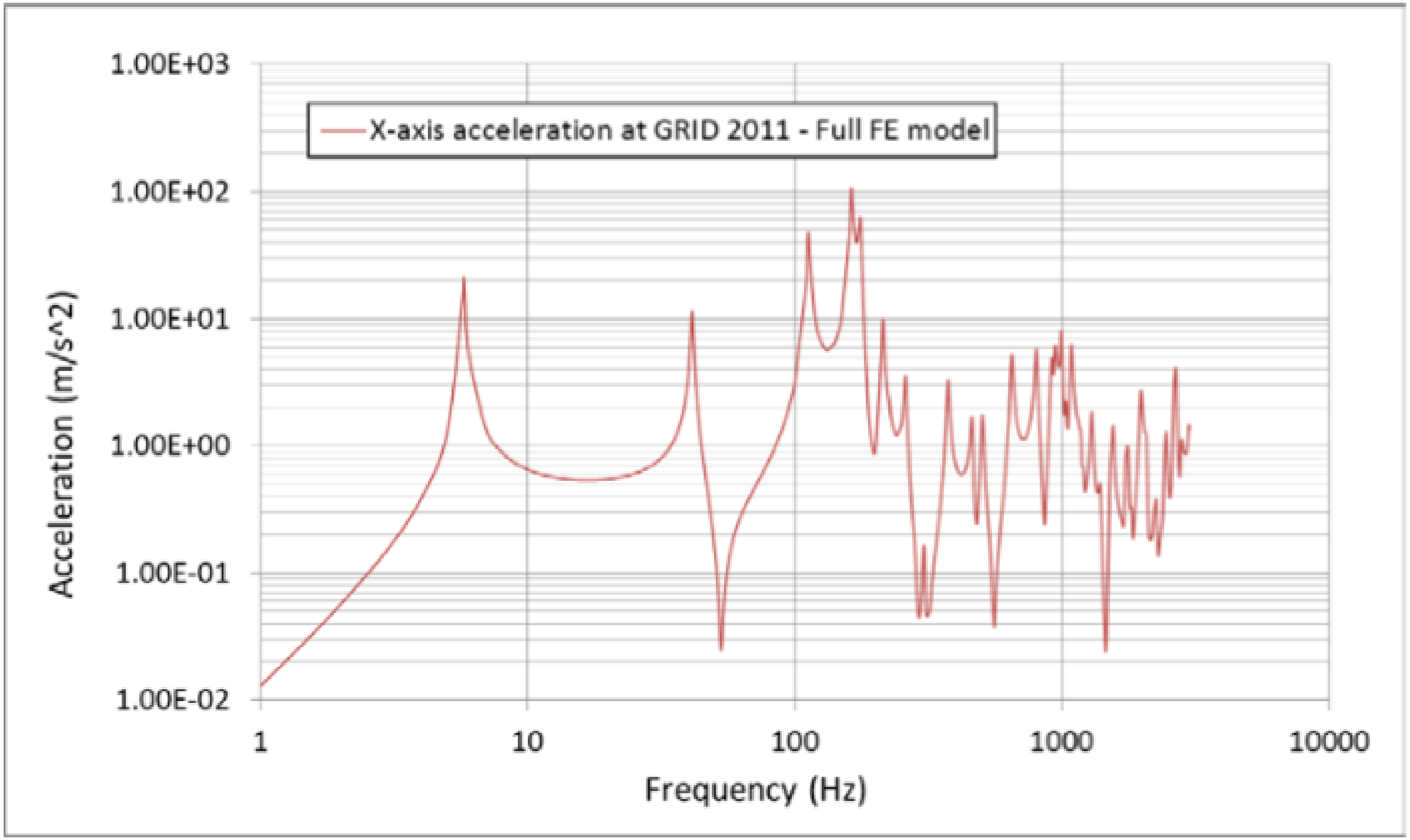


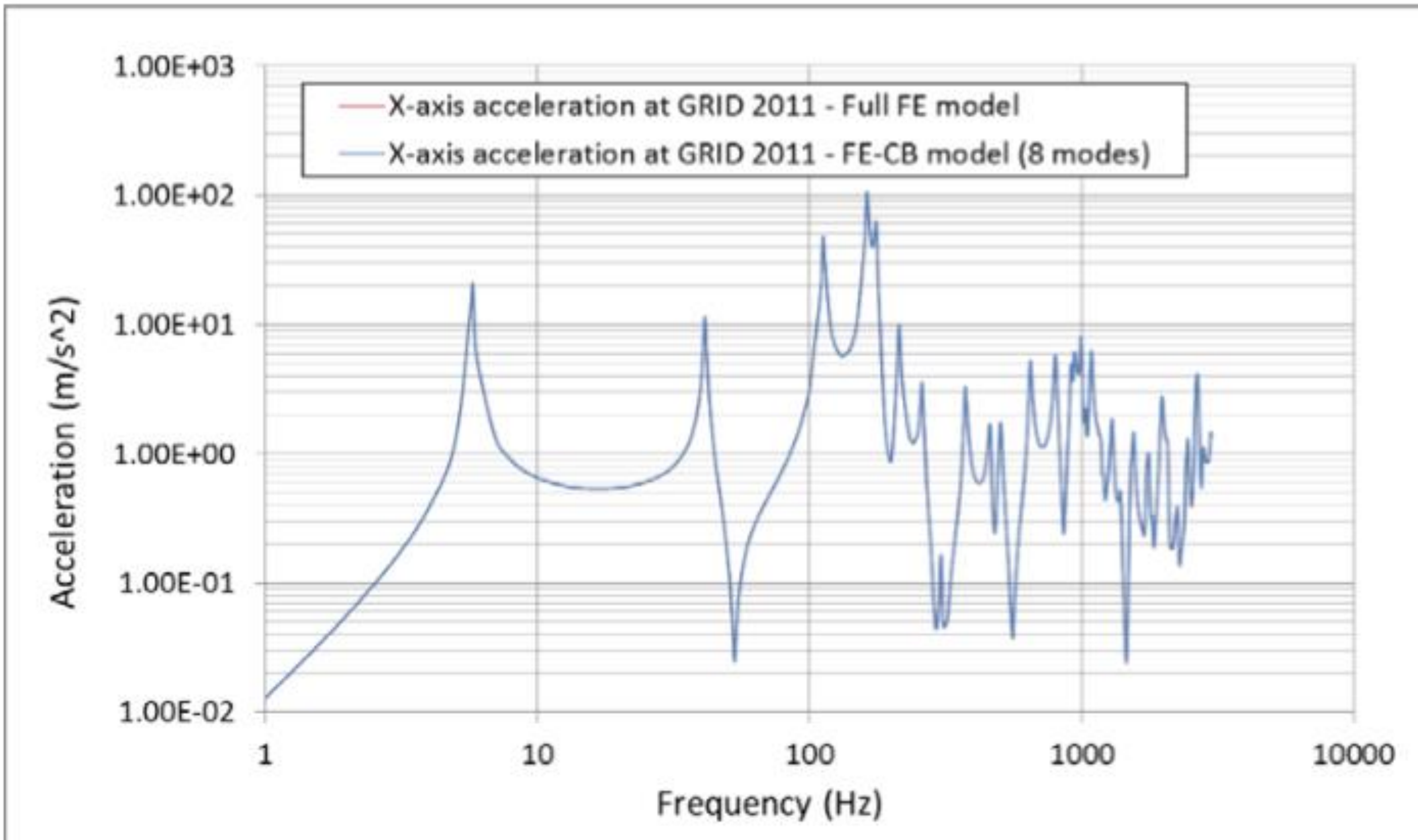
Table 2 – Beam fixed-interface normal modes

Mode No.	Frequency (Hz)
1	48.69
2	164.25
3	301.29
4	833.13
5	1000.21
6	1609.48
7	2616.94
8	2697.38
9	3828.42
10	4994.86
11	5026.63
12	5190.90

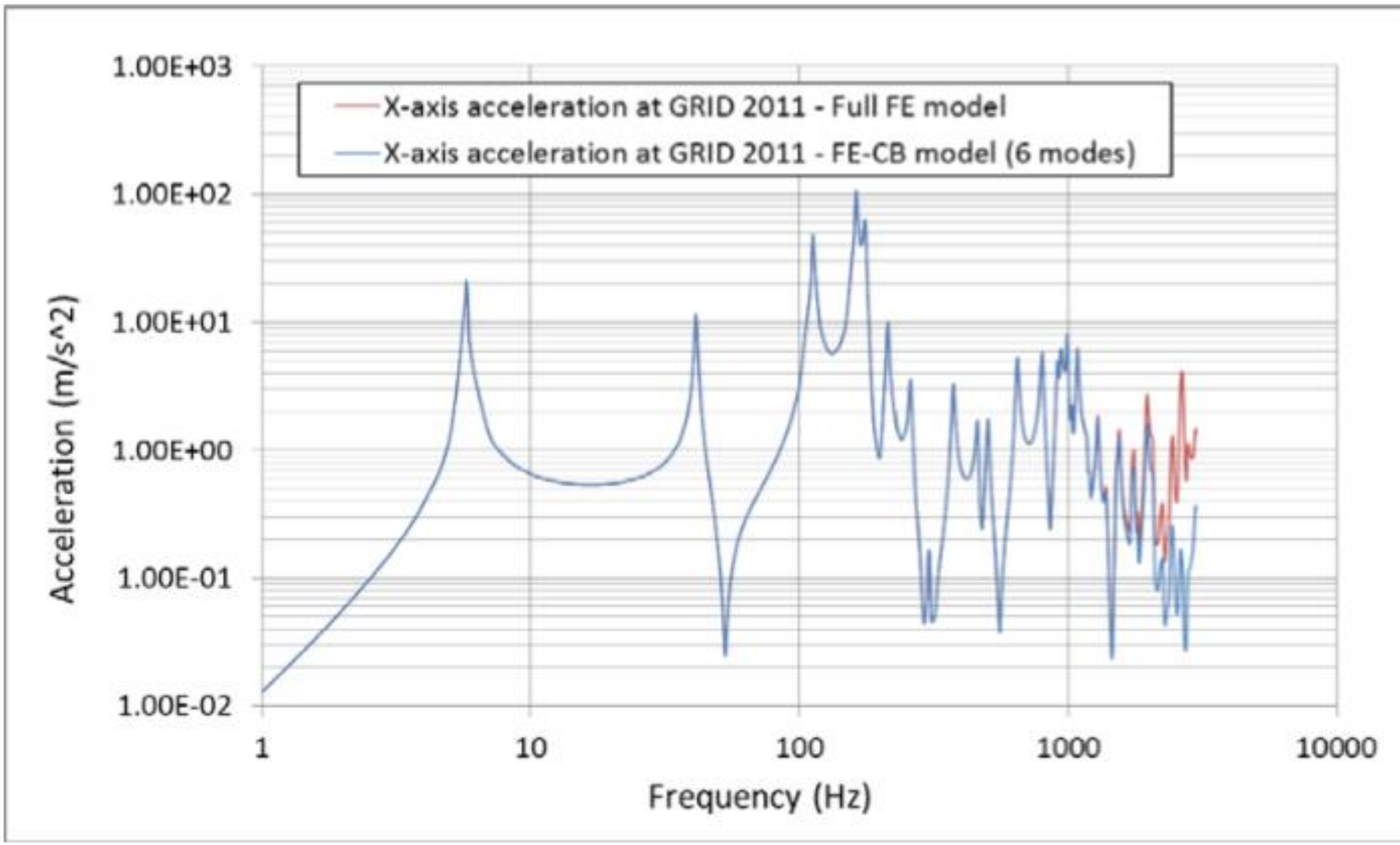
MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):



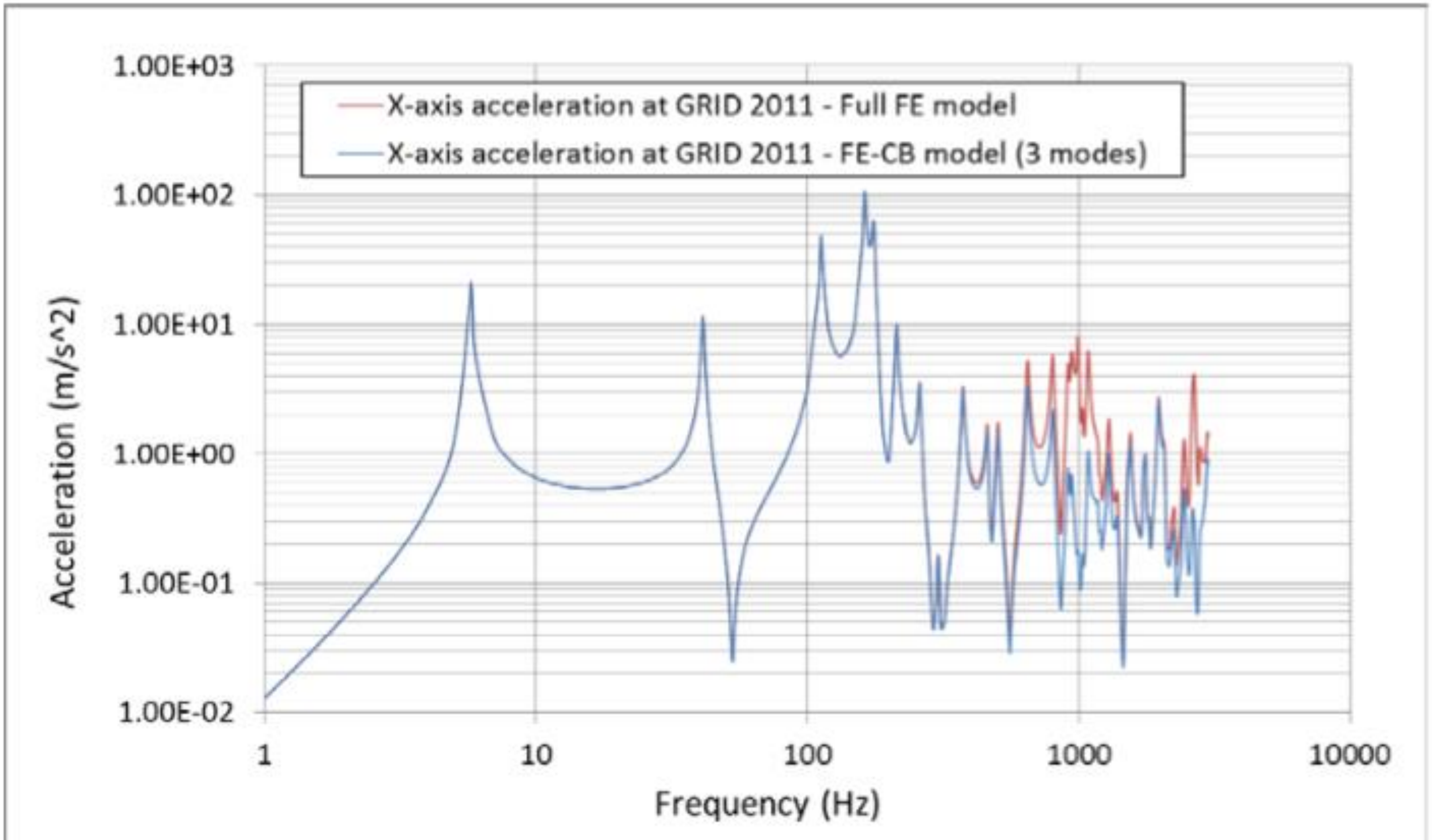
MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):



MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):

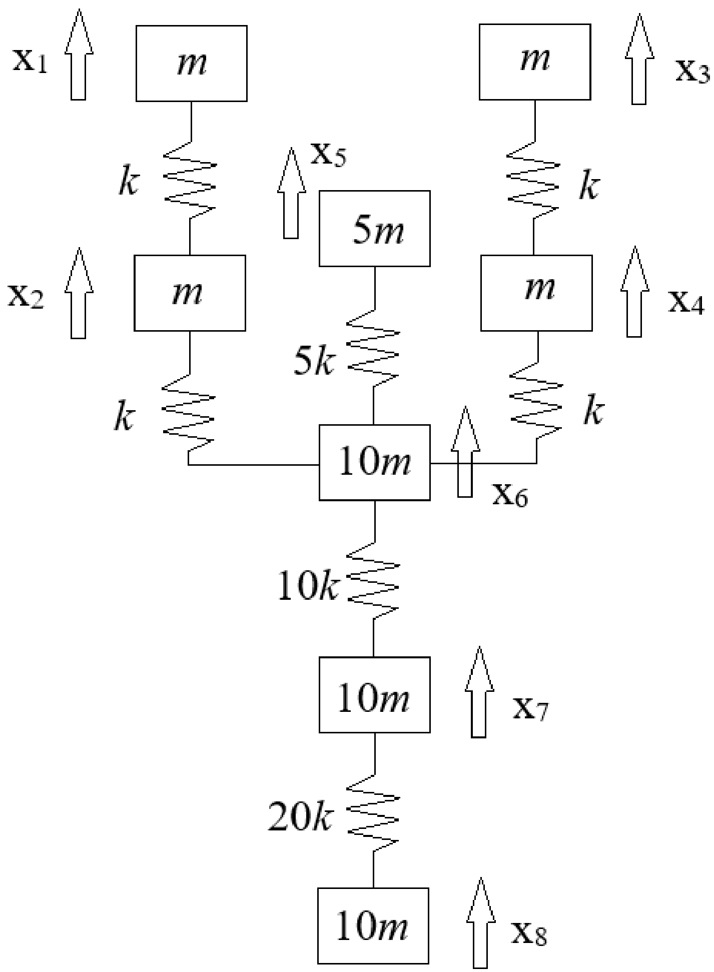


MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):



MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):

$m = 1$ kg $\qquad k = 1 \cdot 10^5$ N/m



- Matrices de masa y rigidez del sistema.
- Clasificación en grados de libertad interiores y de frontera.

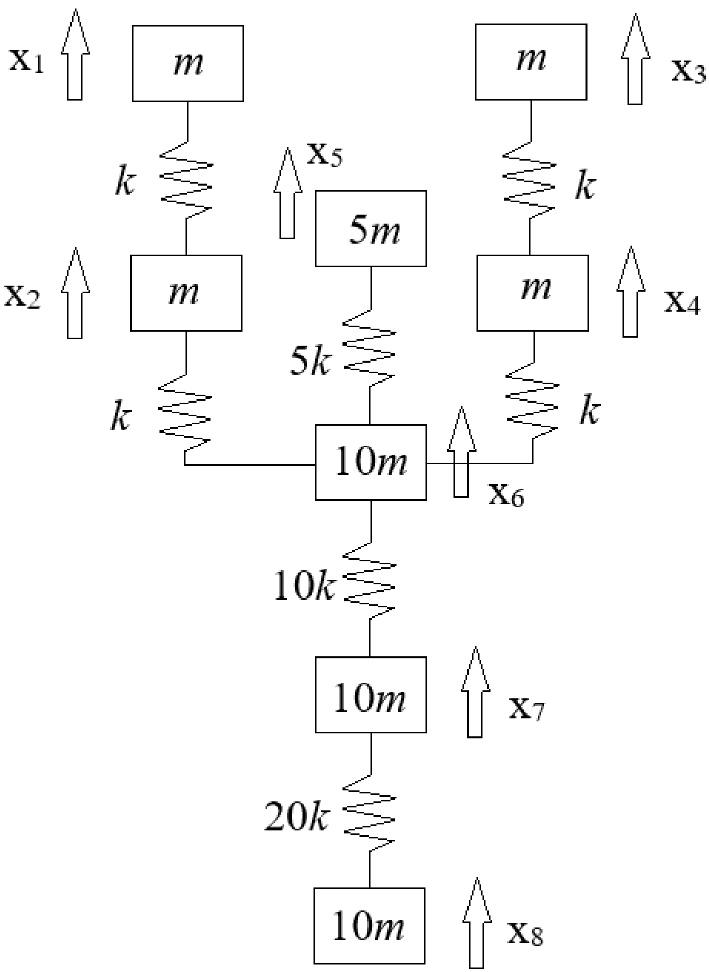
$$\{X\} = \begin{Bmatrix} X_i \\ X_f \end{Bmatrix} = \begin{Bmatrix} x_i \\ x_8 \end{Bmatrix}, \qquad i = 1, \dots .7$$

- Reordenamos las matrices:

$$[M] = \begin{bmatrix} M_{ff} & M_{fi} \\ M_{if} & M_{ii} \end{bmatrix}$$

$$[K] = \begin{bmatrix} K_{ff} & K_{fi} \\ K_{if} & K_{ii} \end{bmatrix}$$

MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):



- Cálculo de los modos estáticos (*constraint* modes)

$$[\phi_s] = \begin{Bmatrix} 1 \\ -[K_{ii}][K_{if}] \end{Bmatrix}$$

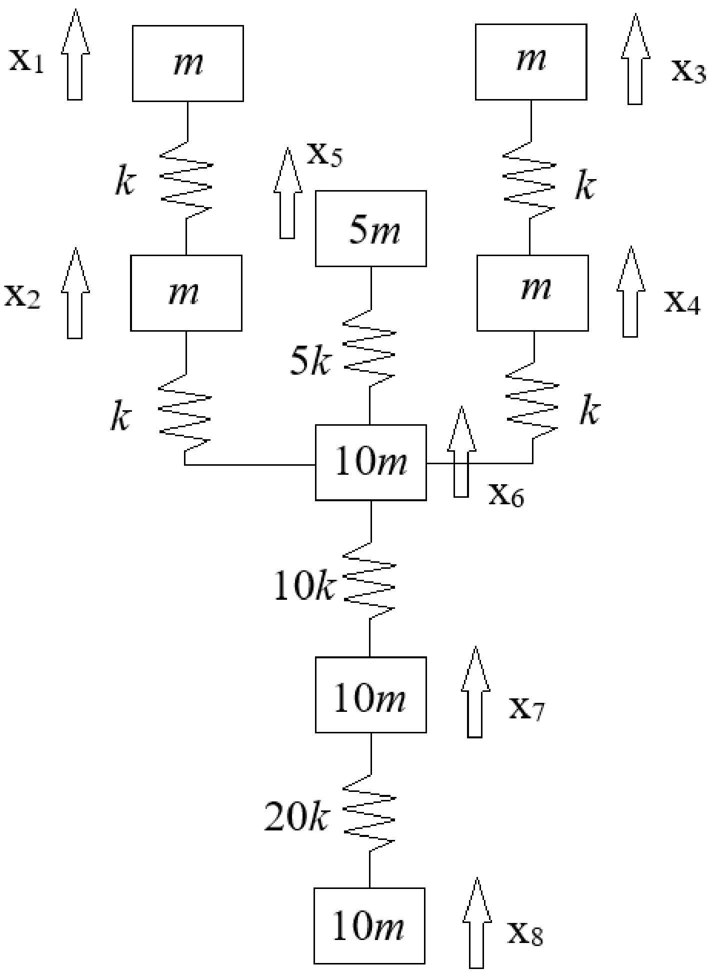
- Cálculo de los modos elásticos interiores

$$|[K_{ii}] - \omega^2 [M_{ii}]| = \{0\}$$

$$([K_{ii}] - \omega^2 [M_{ii}])[\phi_{ii}] = \{0\}$$

$$[\phi_i] = \begin{Bmatrix} 0 \\ [\phi_{ii}] \end{Bmatrix}$$

MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):



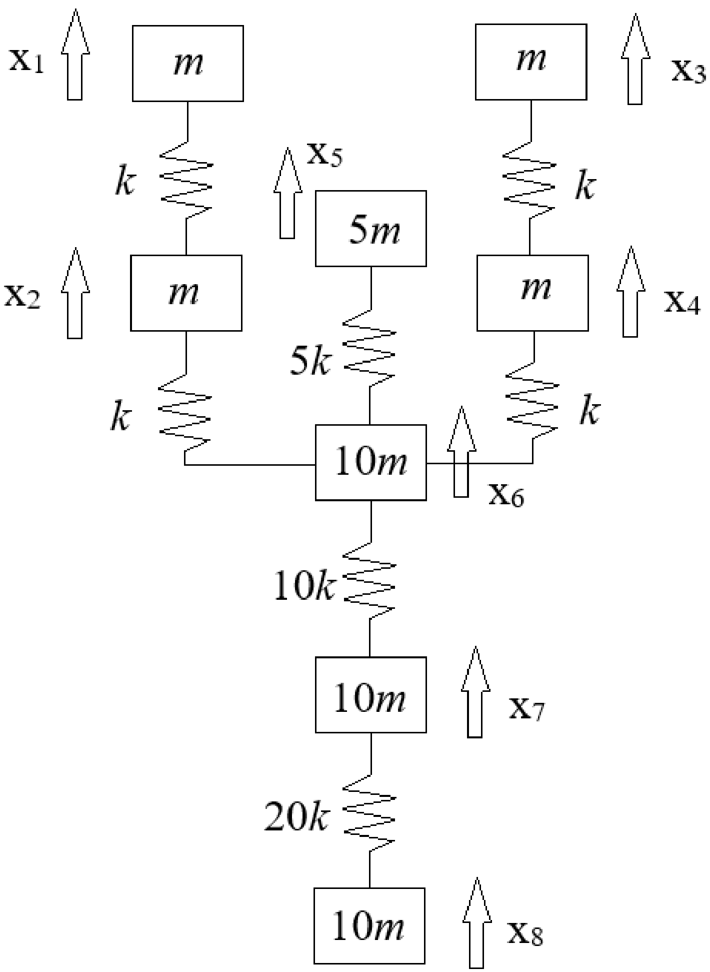
- Matriz de transformación CB del sistema

$$[\phi_{CB}] = [\phi_s \quad \phi_i]$$

$$[M_{CB}] = [\phi_{CB}]^T [M][\phi_{CB}]$$

$$[K_{CB}] = [\phi_{CB}]^T [K][\phi_{CB}]$$

MÉTODO DE SÍNTESIS DE COMPONENTES (CRAIG-BAMPTON):



- Para el cálculo de factores de participación modal y masas generalizadas:

$$[L_{ii}] = [M_{CB}(2{:}8, 1)]$$

$$[m_i] = [M_{CB}(2{:}8, 2{:}8)]$$

- Masas modales efectivas:

$$[M_{eff,i}] = \frac{[L_{ii}]^T [L_{ii}]}{m_i}$$