Challenging wind and waves

Linking hydrodynamic research to the maritime industry

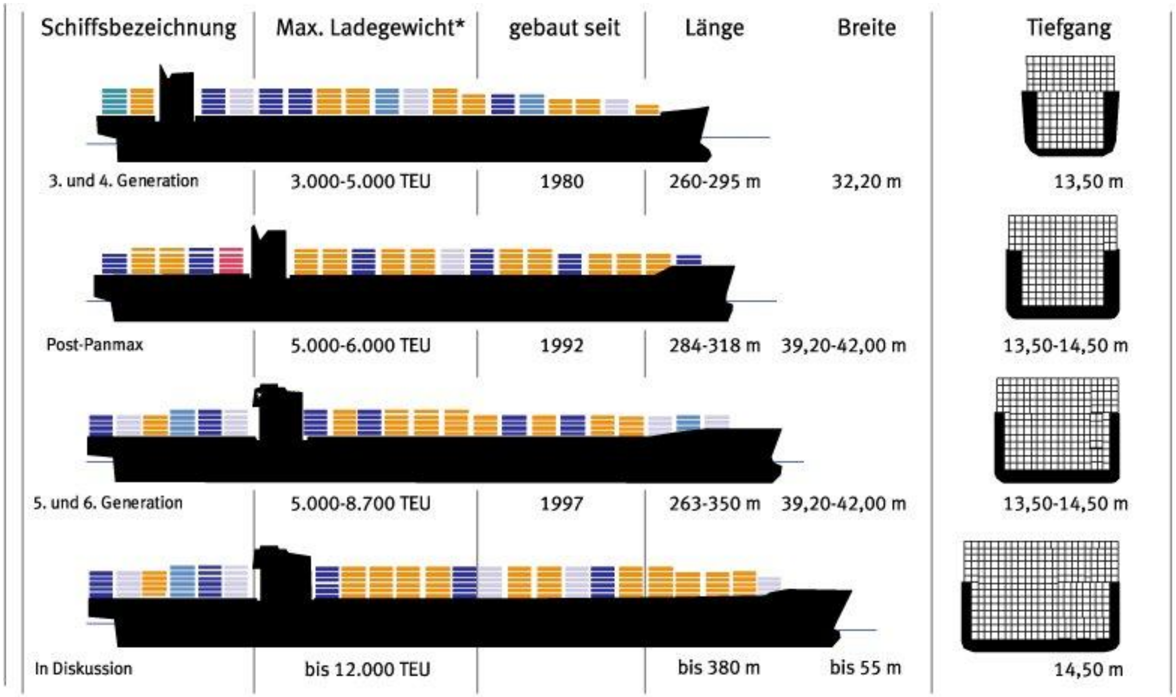
Structural reality @ scale

9 February 2012

Ingo Drummen (MARIN)

INTRODUCTION

- Ships are getting longer



Schiffsbezeichnung	Max. Ladegewicht*	gebaut seit	Länge	Breite	Tiefgang
3. und 4. Generation	3.000-5.000 TEU	1980	260-295 m	32,20 m	13,50 m
Post-Panmax	5.000-6.000 TEU	1992	284-318 m	39,20-42,00 m	13,50-14,50 m
5. und 6. Generation	5.000-8.700 TEU	1997	263-350 m	39,20-42,00 m	13,50-14,50 m
In Diskussion	bis 12.000 TEU		bis 380 m	bis 55 m	14,50 m

* in TEU = Transport Equivalent Unit, entsprechend einem 20-Fuß-Container

- Many large and ultra large container vessels have entered operation

INTRODUCTION

- Today's largest CS
  - 15200TEU
  - 397.71m long
  - 56m wide
  - 15.5m draft

15000 TEU MV Edith Maersk , Emma Maersk class
Source: wikipedia

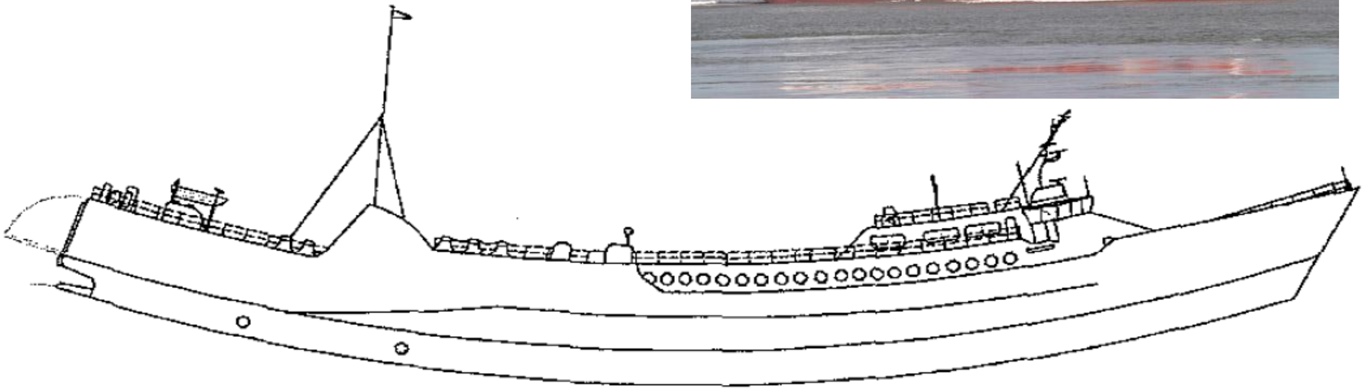
- An ultra-large CS of 20250TEU would measure 440m

HULL FLEXIBILITY

source: youtube

QUESTION 1

- What is the deflection amidships of a 300m containership in sagging conditions between two wave crests?
- A: 5mm
- B: 5cm
- C: 0.5m
- D: 5m



source: Faltinsen (2005)

ANSWER 1

- 75000tons
- 300m long
- Cross sectional moment of inertia 285m^4

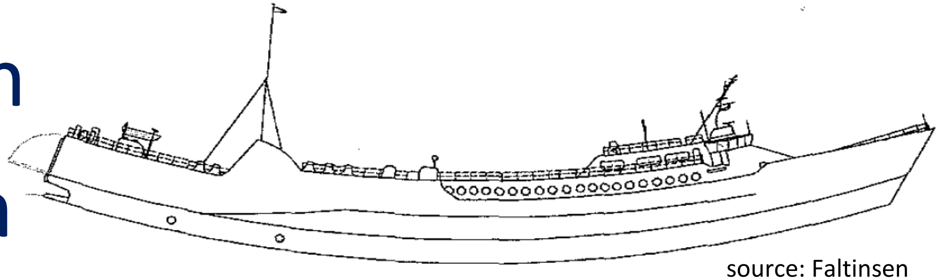
$$\frac{5}{384}\frac{qL^4}{EI} = \frac{5}{384}\frac{75\cdot10^6\cdot9.81\cdot300^3}{6\cdot10^{13}} = 4.3m$$

- Answer D

INCREASED EXTREME LOADS

- Larger size ships
  - implies increased hull flexibility
- Severe slamming can occur
  - if ships operate in harsh weather and/or
  - at high speed
- Combination of slamming and flexibility
  - increases the design load effects

INCREASED FATIGUE LOADS



source: Faltinsen (2005)

- Whipping contribution
- Springing contribution
- Springing is resonant vibration
  - of the two node mode
- Springing occurs for large ships
  - wave frequencies not high enough for resonance of small ships

SPRINGING

$$f = 3.57\sqrt{\frac{EI}{\rho A L^4}} \sim L^{-2}$$

- Increasing size
- Decreasing natural frequency
- Increasing springing probability

$$n\omega_e = \omega_s$$

ω_e = wave encounter frequency

ω_s = natural frequency
of the two node mode

SPRINGING EXAMPLE

- Containership with a length: 300m
- Natural frequency flexural two node: 0.5Hz
- Linear springing at 20kn:

$$\omega = \frac{-1 \pm \sqrt{1 + 4 \frac{U}{g} \omega_e}}{2 \frac{U}{g}} = \frac{-1 \pm \sqrt{1 + \frac{4 * 10.288}{9.81} * \pi}}{2 * \frac{10.288}{9.81}} = 1.45 rad/s$$

- Wave period: 4s
- Seconds order springing occurs for

$$\omega = \frac{-1 \pm \sqrt{1 + 4 \frac{U}{g} \omega_e}}{2 \frac{U}{g}} = \frac{-1 \pm \sqrt{1 + \frac{4 * 10.288}{9.81} * \pi / 2}}{2 * \frac{10.288}{9.81}} = 0.92 rad/s$$

- Wave period: 7s

QUESTION 2

- What is the relative importance of fatigue damage due to wave-induced vibrations of the lowest flexural modes of a 300m container ship?
- A: 0-25%
- B: 25-50%
- C: 50-100%
- D: >100%

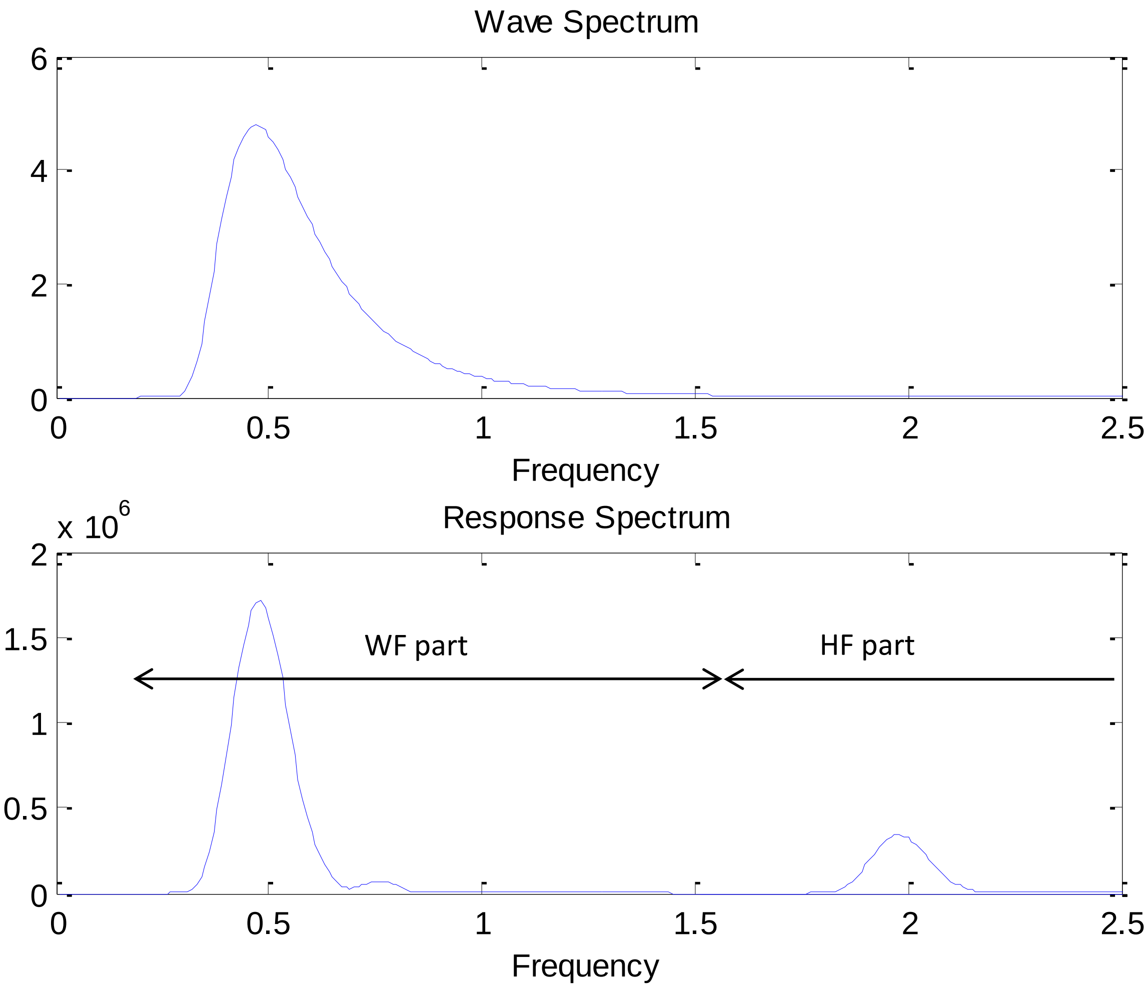
ANSWER 2

- Aalberts and Nieuwenhuijs (2006): 25% for a small container vessel
- Moe et al. (2005): 50% for a containership of 285m
- Drummen et al. (2008): 40% for a containership of 300m
- Answer B

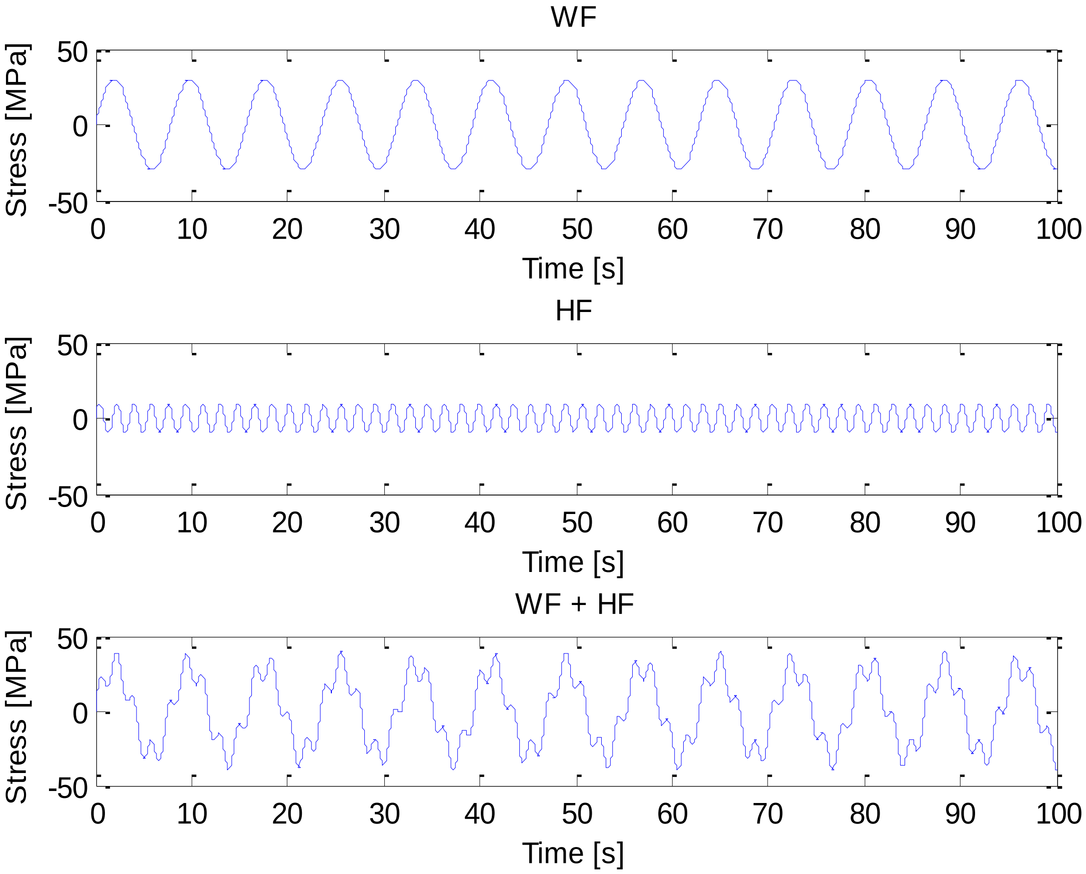
THREE CATEGORIES OF FATIGUE DAMAGE

- Total damage: damage due to the total stress history
- Wave frequency (WF) damage: damage due to the wave frequency stresses
- High frequency (HF) damage: difference between total damage and WF damage

THREE CATEGORIES OF FATIGUE DAMAGE



WHY IS HF DAMAGE IMPORTANT?



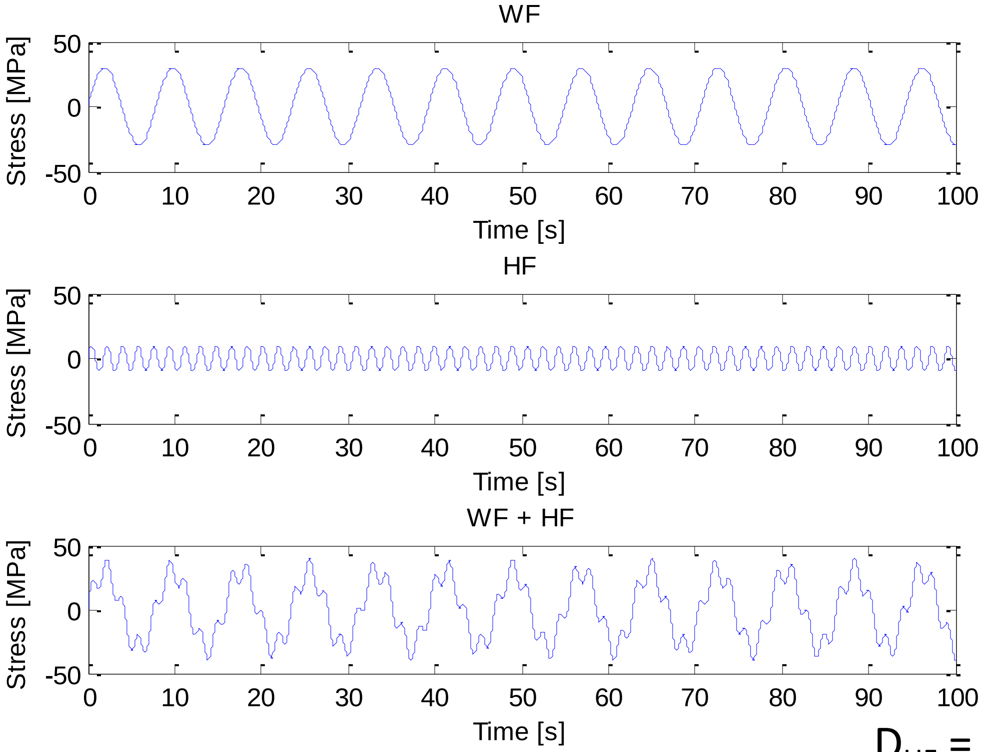
Damage = $0.356 \cdot 10^{-6}$

Damage = 1 ➔ failure

Damage = $0.007 \cdot 10^{-6}$

Damage = $1.102 \cdot 10^{-6}$

WHY IS HF DAMAGE IMPORTANT?



$D = 0.356 \cdot 10^{-6} \rightarrow D_{WF} = 0.356 \cdot 10^{-6}$

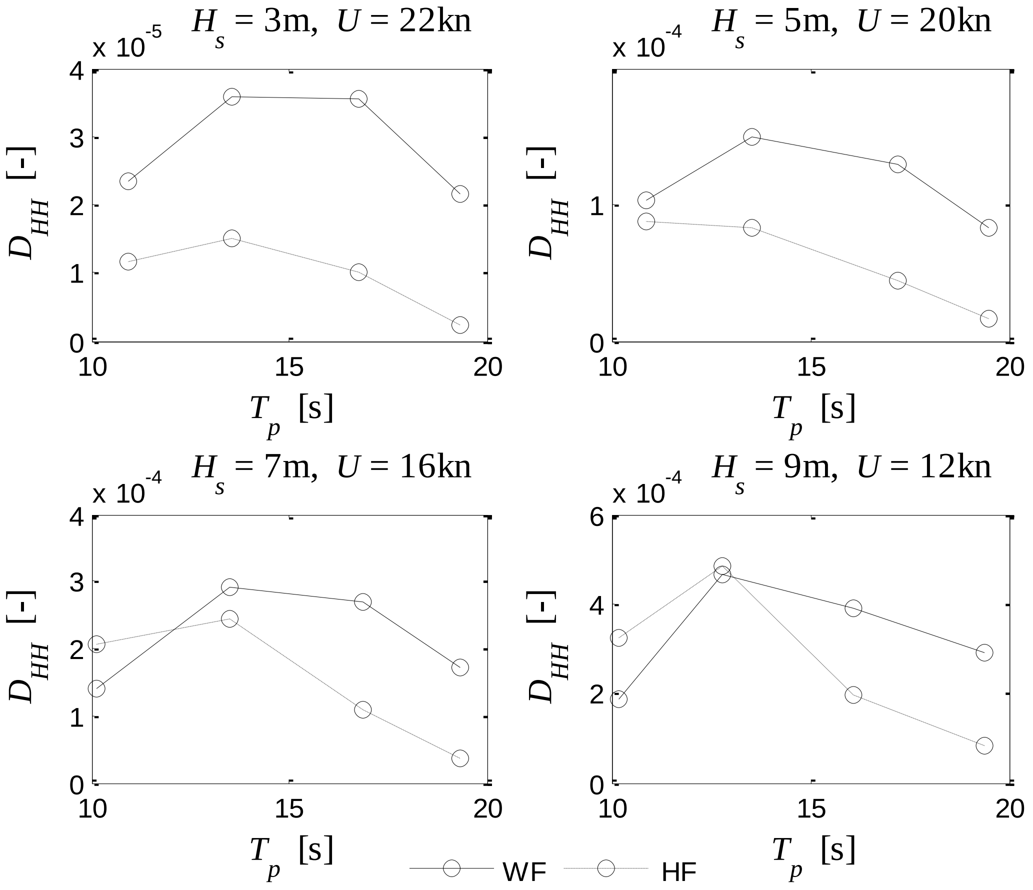
$D = 0.007 \cdot 10^{-6}$

$D = 1.102 \cdot 10^{-6} \rightarrow D_{T} = 1.102 \cdot 10^{-6}$

$D_{HF} = 1.102 \cdot 10^{-6} - 0.356 \cdot 10^{-6} = 0.746 \cdot 10^{-6}$

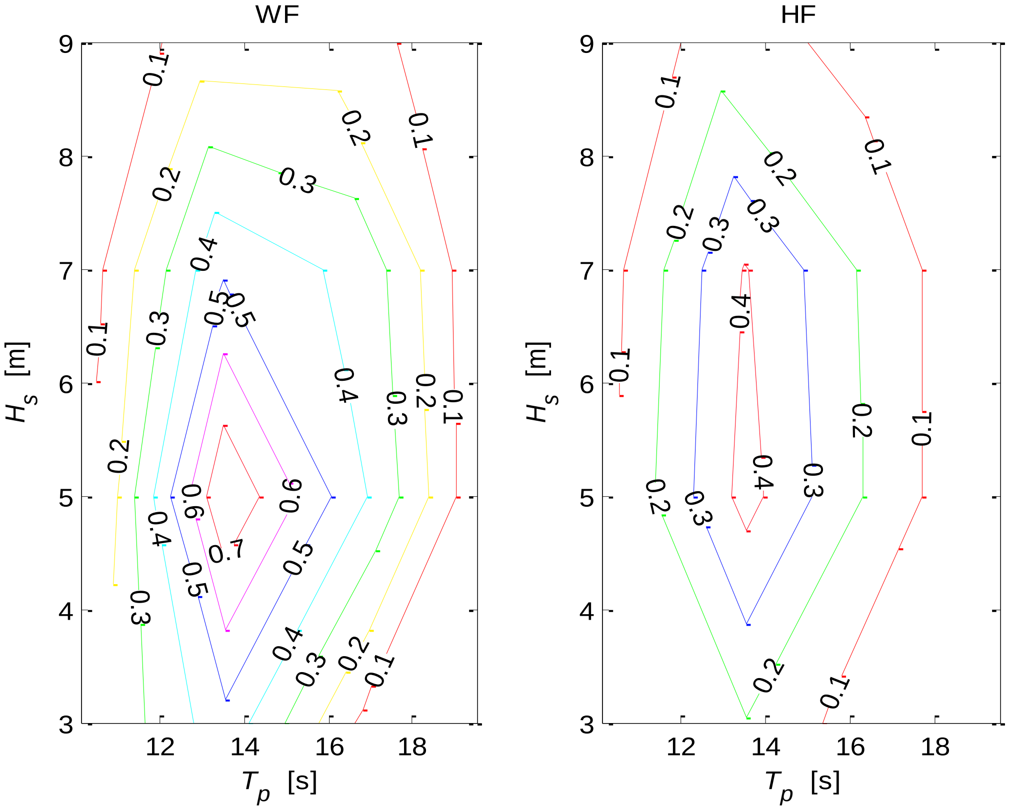
DRUMMEN ET AL. (2008)

- 16 sea states investigated (North Atlantic)



DRUMMEN ET AL. (2008)

- Results were combined
  - with total time in each sea state



Life time fatigue damage

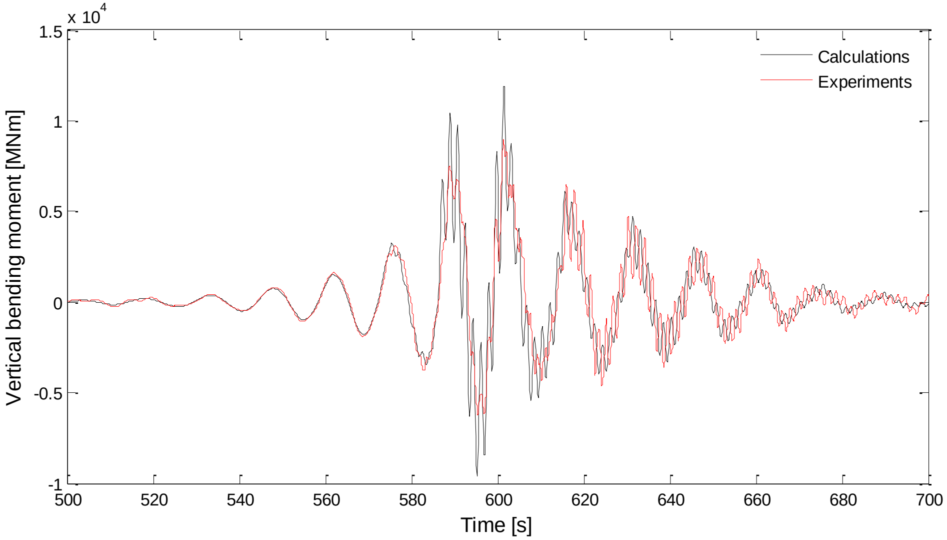
WF damage	63%
HF damage due to wave-induced vibrations	**37%**

CONCLUSION

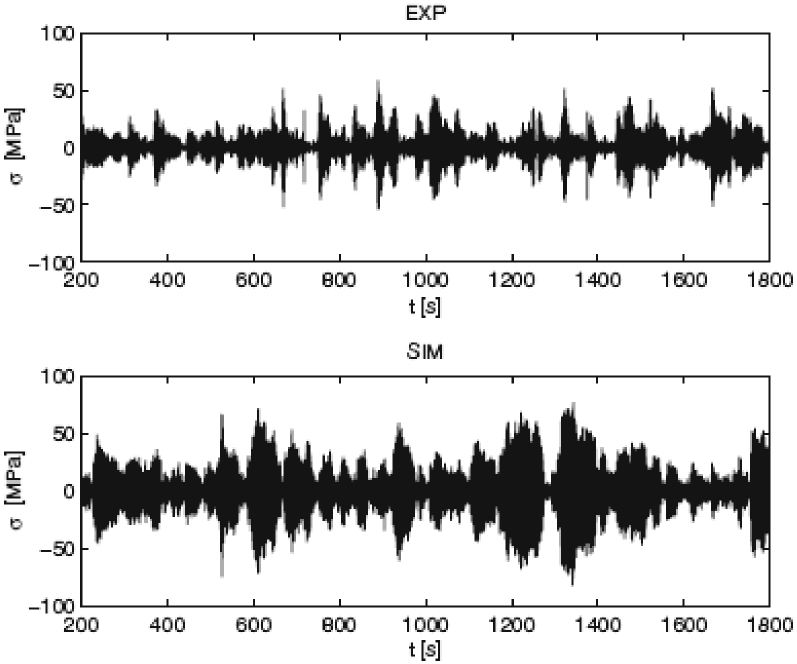
- Hull flexibility should be accounted for
  - in ship design
- Methods are available
  - for linear wave- and high-frequency stresses
- The challenge is to understand:
  - nonlinear hydrodynamic load mechanisms that cause high-frequency load effects
  - damping mechanisms

MODEL TESTS WITH 300M CONTAINERSHIP

whipping



springing



CONCLUSION

Model tests where flexibility is incorporated are necessary

THREE WAYS OF INCORPORATING FLEXIBILITY

- Fully flexible model
- Segmented model
- Combination between:
  - rigid physical model
  - finite element model

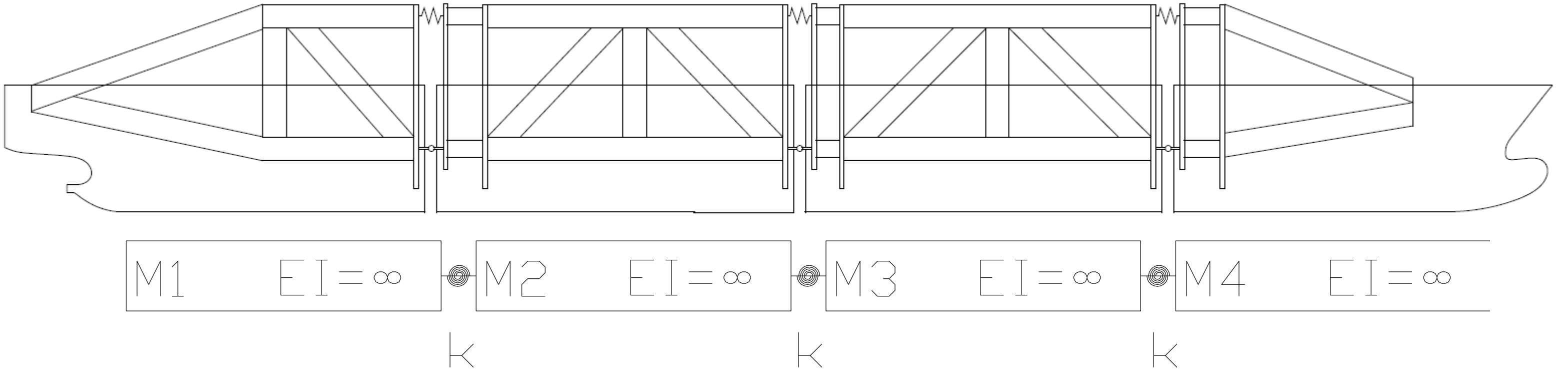
FULLY FLEXIBLE MODEL

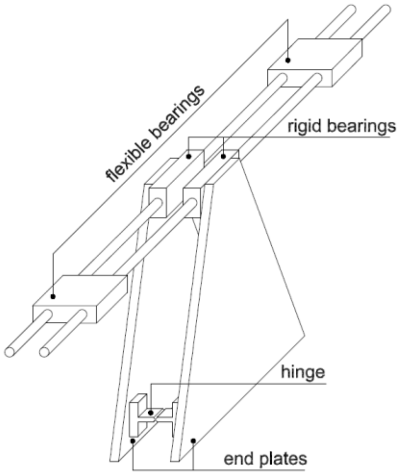
- Advantages
  - good representation of reality
- Disadvantages
  - difficult to built, flexibility needs to be scaled strength should be enough
  - expensive to built

SEGMENTED MODEL

- Flexible backbone
- Flexible joints

SEGMENTED MODEL WITH FLEXIBLE JOINTS





SEGMENTED MODEL WITH FLEXIBLE JOINTS

- Advantages
  - natural frequency can be tuned
- Disadvantages
  - difficult to built for VBM, HBM and torsion
  - cuts are expensive
  - discrete stiffness

COMBINATION PHYSICAL AND FE MODEL

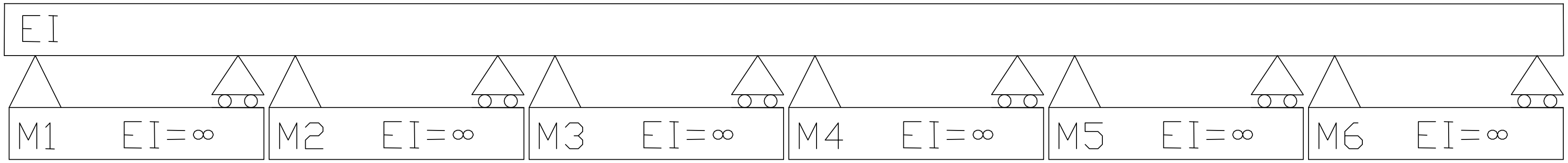
- Rigid model in basin
- Many pressure gauges
- Pressures on FE model
- Advantages
  - flexibility
- Disadvantages
  - sufficiently detailed FE model is needed

SEGMENTED MODEL WITH FLEXIBLE BACKBONE

- Goal is to build a model that has the same modal parameters as the ship
- Modal parameters
  - mode shapes
  - natural frequency
- Modeling up to
  - 2 and 3 node horizontal and vertical
  - torsion can also be modeled

SEGMENTED MODEL WITH FLEXIBLE BACKBONE

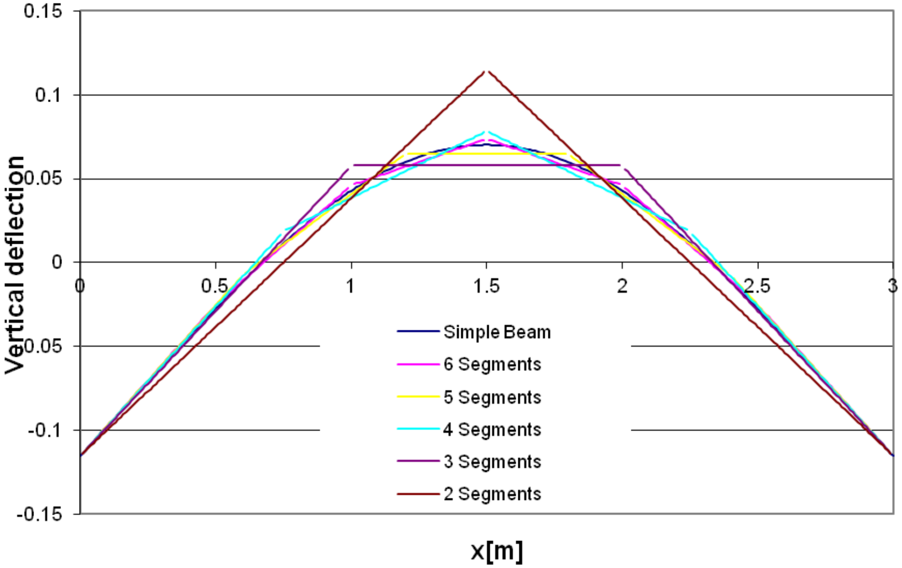
- Rigid segments connected to a flexible beam

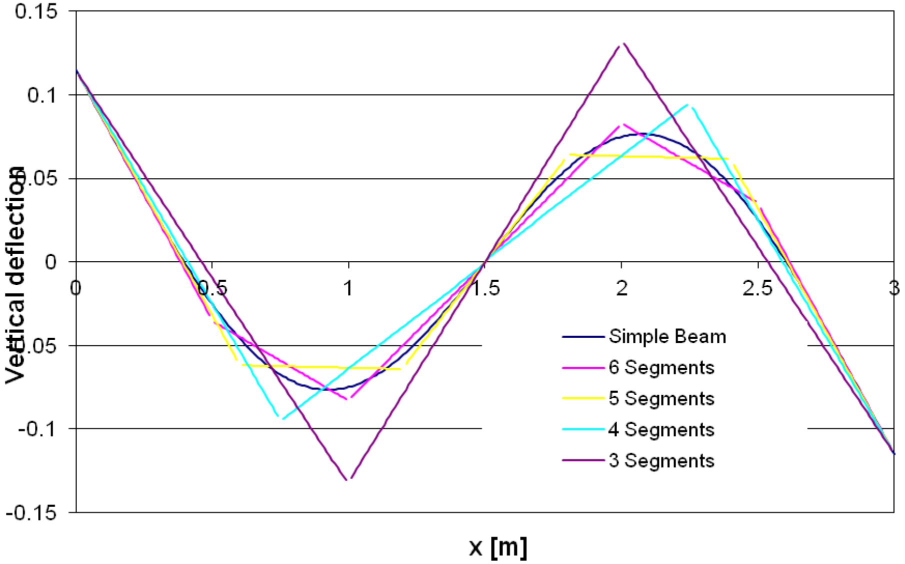


- Beam controls (noncontinuous) stiffness
- Segments control mass

SEGMENTED MODEL WITH FLEXIBLE BACKBONE

- Usually 5 or 6 segments





- Local weakening to fine tune natural frequencies

QUESTION 3

- How deep should the cut in a beam (d=110mm, t=2.5mm) be to achieve a local stiffness reduction of 30%?
- A: 2mm
- B: 5mm
- C: 20mm
- D: 30mm

ANSWER 3

- Answer A

SEGMENTED MODEL WITH FLEXIBLE BACKBONE

- Advantages
  - inexpensive solution
  - natural frequency can be tuned, but not as elegant as with flexible joints
- Disadvantages
  - difficult to incorporate torsion
  - measurements and weakening preferably at same location

TAILORED LIFETIME ASSURANCE

- MARIN goal: tailored lifetime assurance during the **design** and **operational** stages

TAILORED LIFETIME ASSURANCE – NUMERICAL DESIGN

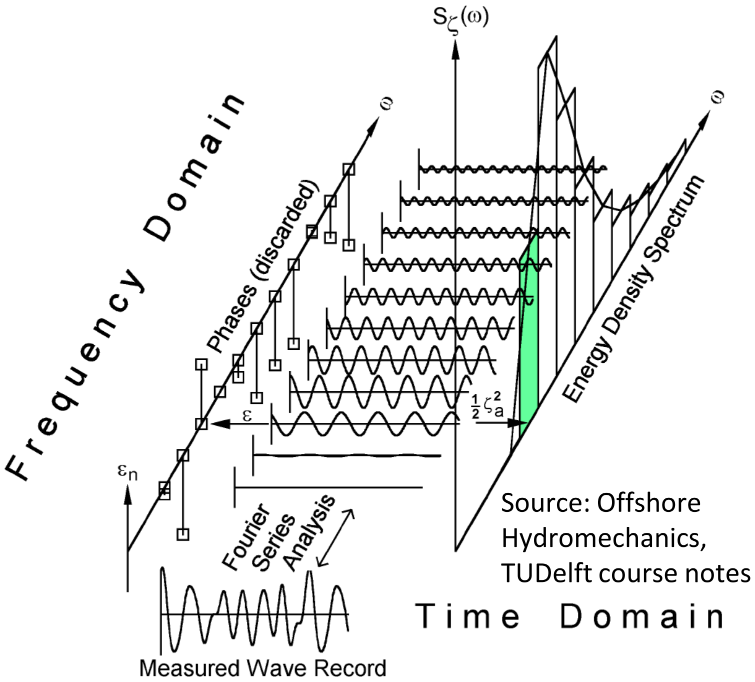
- Preliminary fatigue design method
  - length, breadth, draft, …
- Linear method I
  - geometry, mass dist., global structure, …
- Linear method II
  - geometry, mass dist., FE model, …

SPECTRAL FATIGUE CALCULATION

scatter diagram

Hs										
8.5	0	0	0	0	1	1	1	1	0	0
7.5	0	0	0	1	2	2	2	1	0	0
6.5	0	0	0	2	5	5	4	2	1	0
5.5	0	0	2	7	12	12	6	4	1	1
4.5	0	1	5	19	29	25	14	5	2	1
3.5	0	2	17	47	58	40	19	6	2	0
2.5	0	8	45	89	81	44	16	4	1	0
1.5	2	23	74	89	54	20	5	1	0	0
0.5	5	20	27	15	5	1	0	0	0	0
	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5

Tz

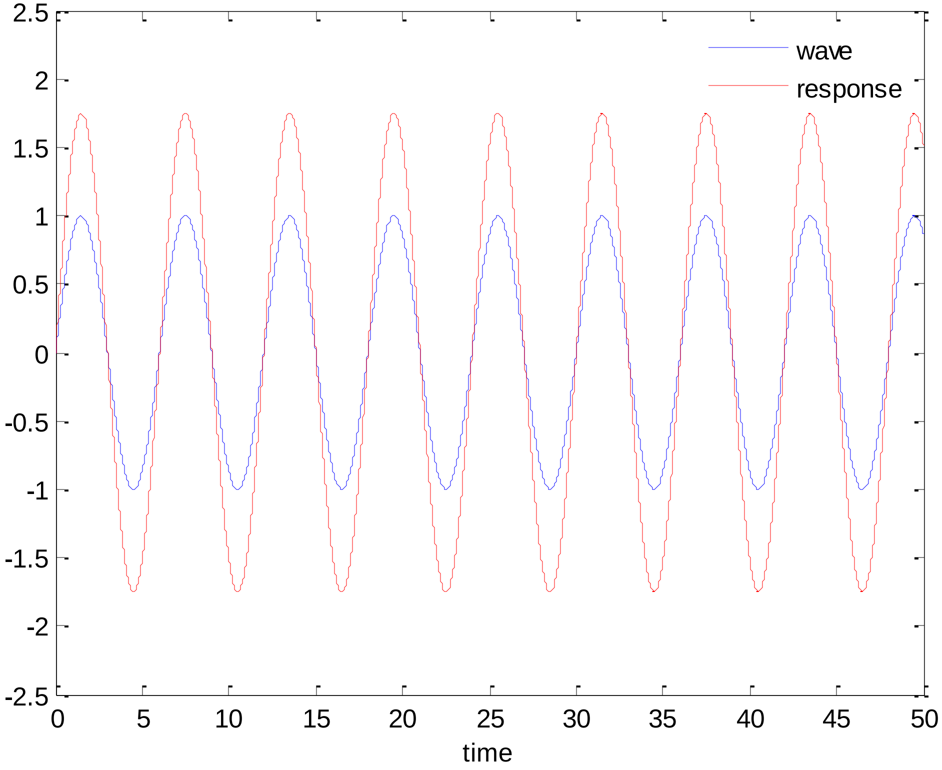


Source: Offshore Hydromechanics, TUDelft course notes

$$S_\zeta(\omega) = \frac{320 \cdot H_{1/3}^2}{T_p^4} \cdot \omega^{-5} \cdot \exp\left\{ \frac{-1950}{T_p^4} \cdot \omega^{-4} \right\}$$

SPECTRAL FATIGUE CALCULATION

$$S_R(\omega) = [\Phi_R(\omega)]^2 S_\zeta(\omega)$$



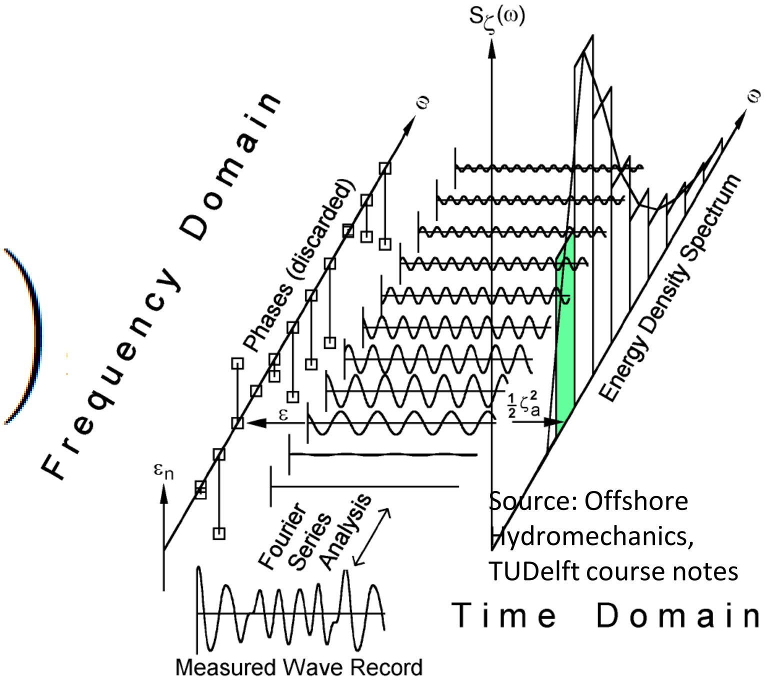
unit response per
unit wave height

$$\Phi_R(\omega) = \frac{A_{response}(\omega)}{A_{wave}(\omega)}$$

SPECTRAL FATIGUE CALCULATION

$$S_R(\omega) = [\Phi_R(\omega)]^2 S_\zeta(\omega)$$

$$\overline{D} = \frac{n_0}{a}(2\sqrt{2}\sigma)^m \Gamma\left(\frac{m}{2}+1\right)$$



Source: Offshore Hydromechanics, TUDelft course notes

scatter diagram

Hs

8.5	0	0	0	0	1	1	1	1	0	0
7.5	0	0	0	1	2	2	2	1	0	0
6.5	0	0	0	2	5	5	4	2	1	0
5.5	0	0	2	7	12	12	6	4	1	1
4.5	0	1	5	19	29	25	14	5	2	1
3.5	0	2	17	47	58	40	19	6	2	0
2.5	0	8	45	89	81	44	16	4	1	0
1.5	2	23	74	89	54	20	5	1	0	0
0.5	5	20	27	15	5	1	0	0	0	0
	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5

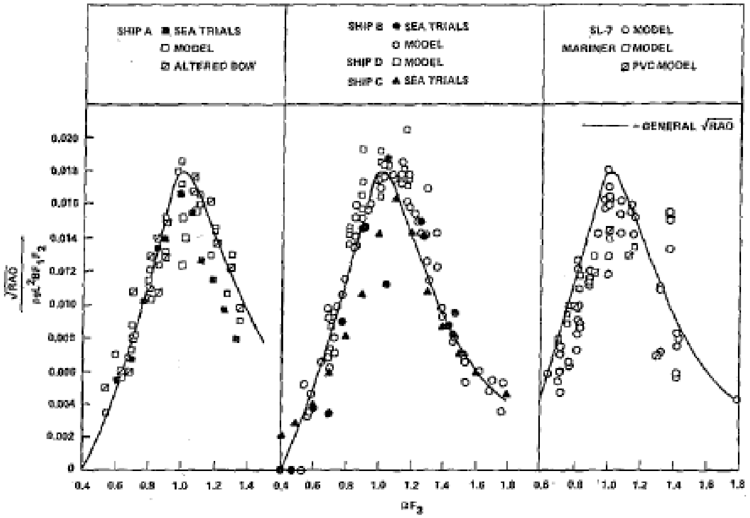
Tz

$$\overline{D_{LT}} = \int_{H_s}\int_{T_p}\int_{U}\int_{\beta} T_{LT}\overline{D_H}(h,t,u,\beta) f_{H_s,T_p,U,\beta}(h,t,u,\beta)\, dh\, dt\, du\, d\beta$$

PRELIMINARY FATIGUE DESIGN METHOD

$$S_R(\omega) = \sum_{\theta} [\Phi_R(\omega, \theta)]^2 S_\zeta(\omega, \theta)$$

- Linear theory
  - using universal RAO $\Phi_R(\omega, \theta)$
- Corrections
  - for weak and strong NLs
- Input
  - principle dimensions
  - empirical factors for NLs



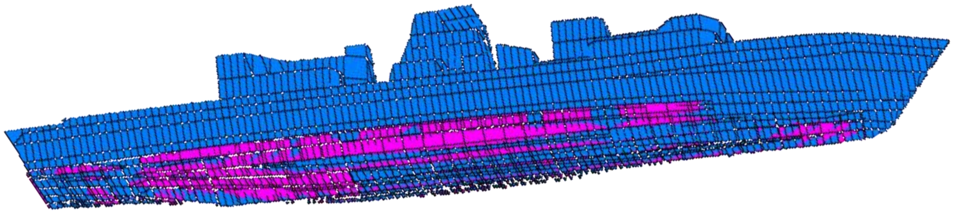
LINEAR METHOD I

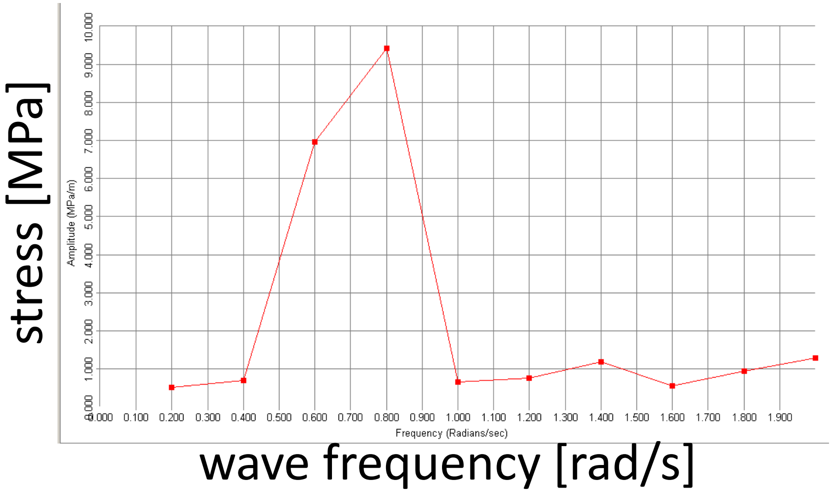
- Geometry and mass distribution
- Hydrodynamic calculations
  - can be done with 2D or 3D method
- Section modulus
  - estimated based on global ship structure
- $\Phi_R(\omega,\theta)$ = VBM divided by section modulus

$$S_R(\omega) = \sum_{\theta} [\Phi_R(\omega,\theta)]^2 S_\zeta(\omega,\theta)$$

LINEAR METHOD II

- Finite element model
  - also available
- $\Phi_R(\omega, \theta)$ = stress transfer functions





$$S_R(\omega) = \sum_{\theta} [\Phi_R(\omega, \theta)]^2 S_\zeta(\omega, \theta)$$

SCENARIO SIMULATIONS

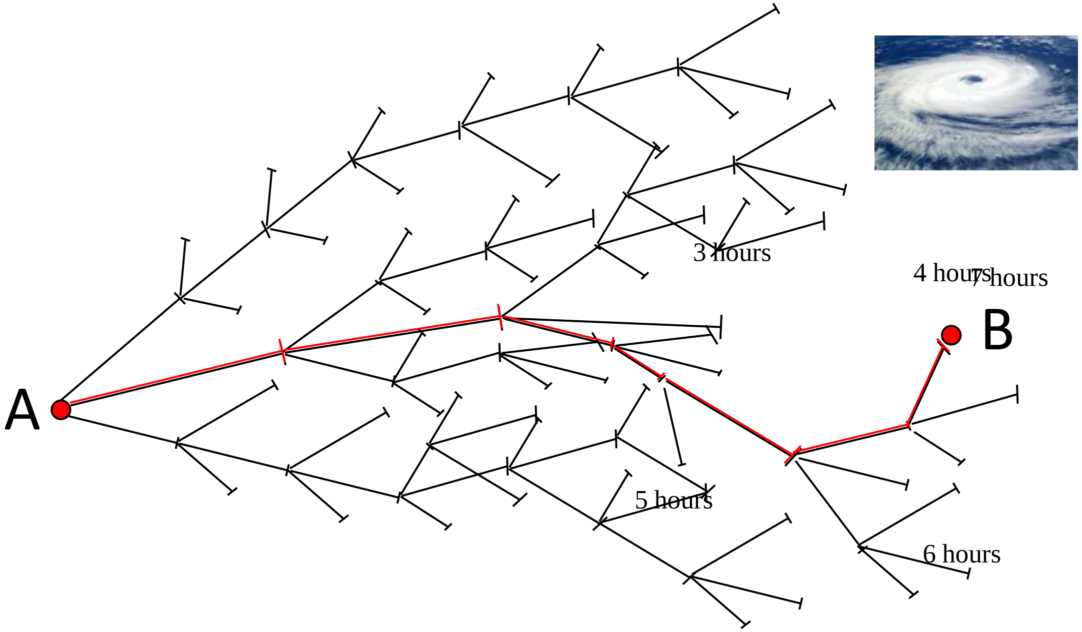
- Above methods rely on scatter diagram
- No scatter diagram used
  - in scenario simulations
- This has several potential merits

SCENARIO SIMULATIONS

- Accounting for the joint occurrence of wind sea, swell, current and wind velocity
- Accounting for the reaction of the master on:
  - changes in the adopted course and power
  - unexpected large delay
- Memory effects:
  - reaction of the master on past and anticipated conditions
  - effect of speed loss and delay on the duration

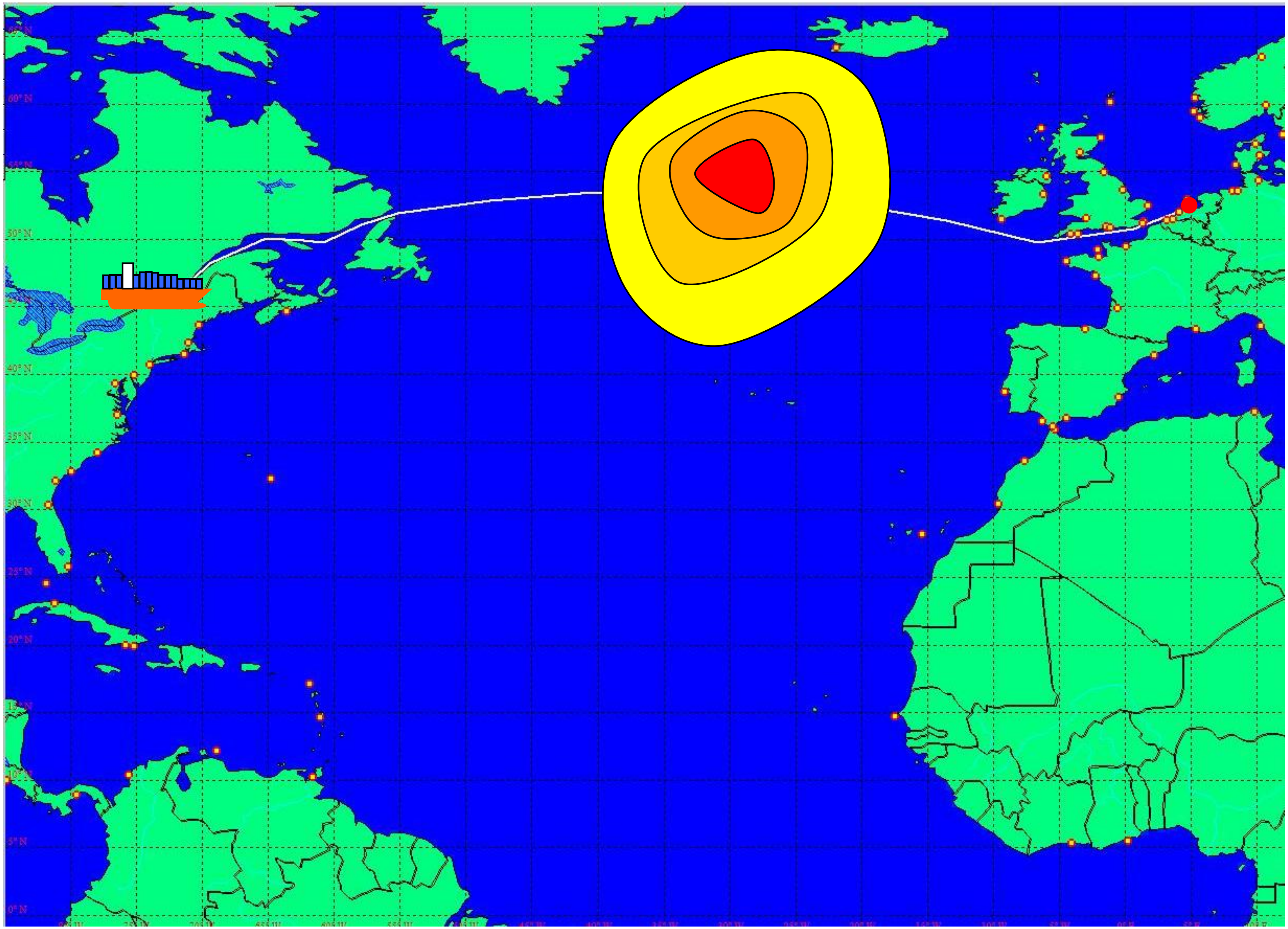
SCENARIO SIMULATIONS

$$S_R(\omega) = \sum_{\theta} [\Phi_R(\omega, \theta)]^2 S_\zeta(\omega, \theta)$$

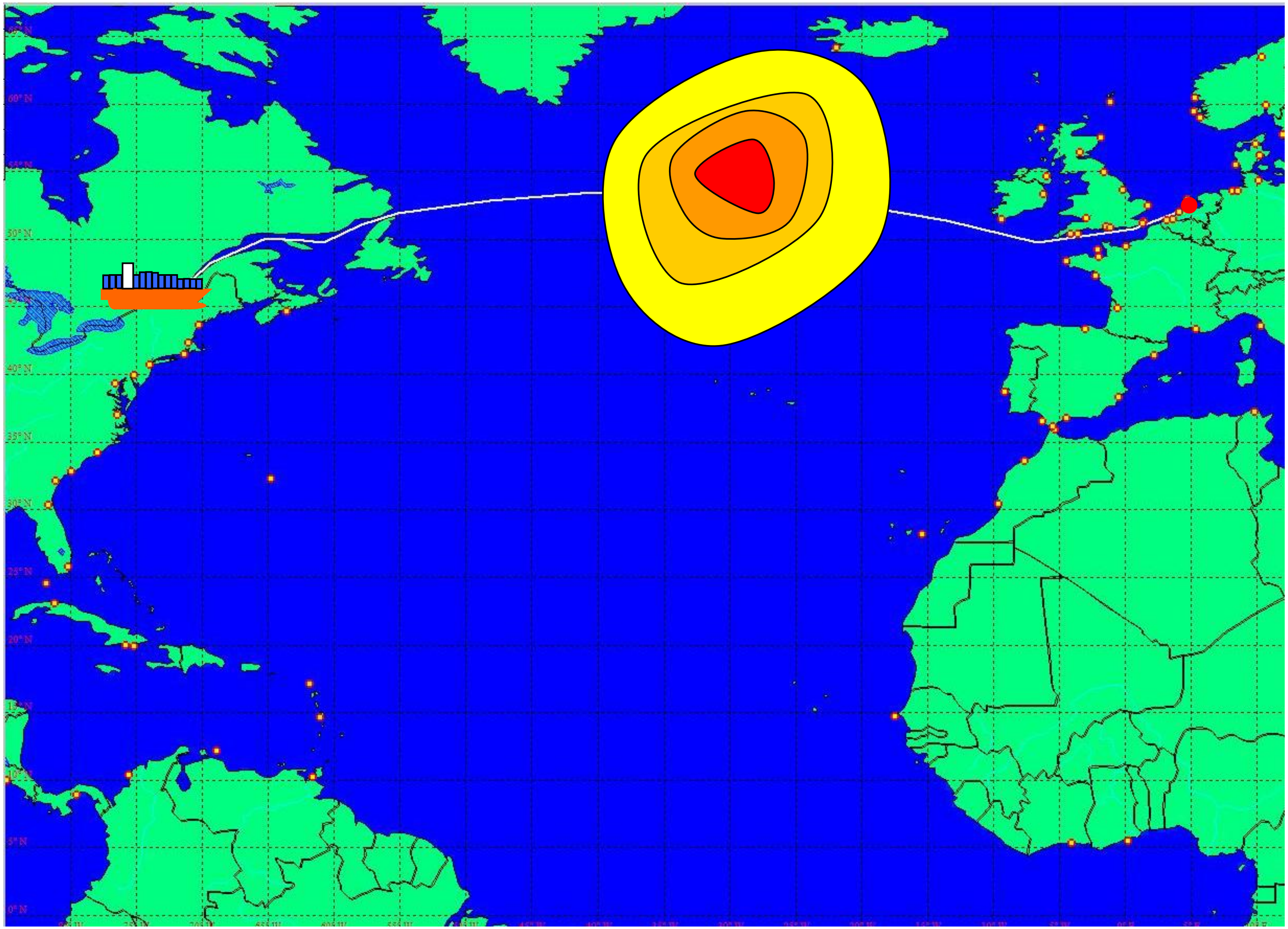


- Applicable for all three methods

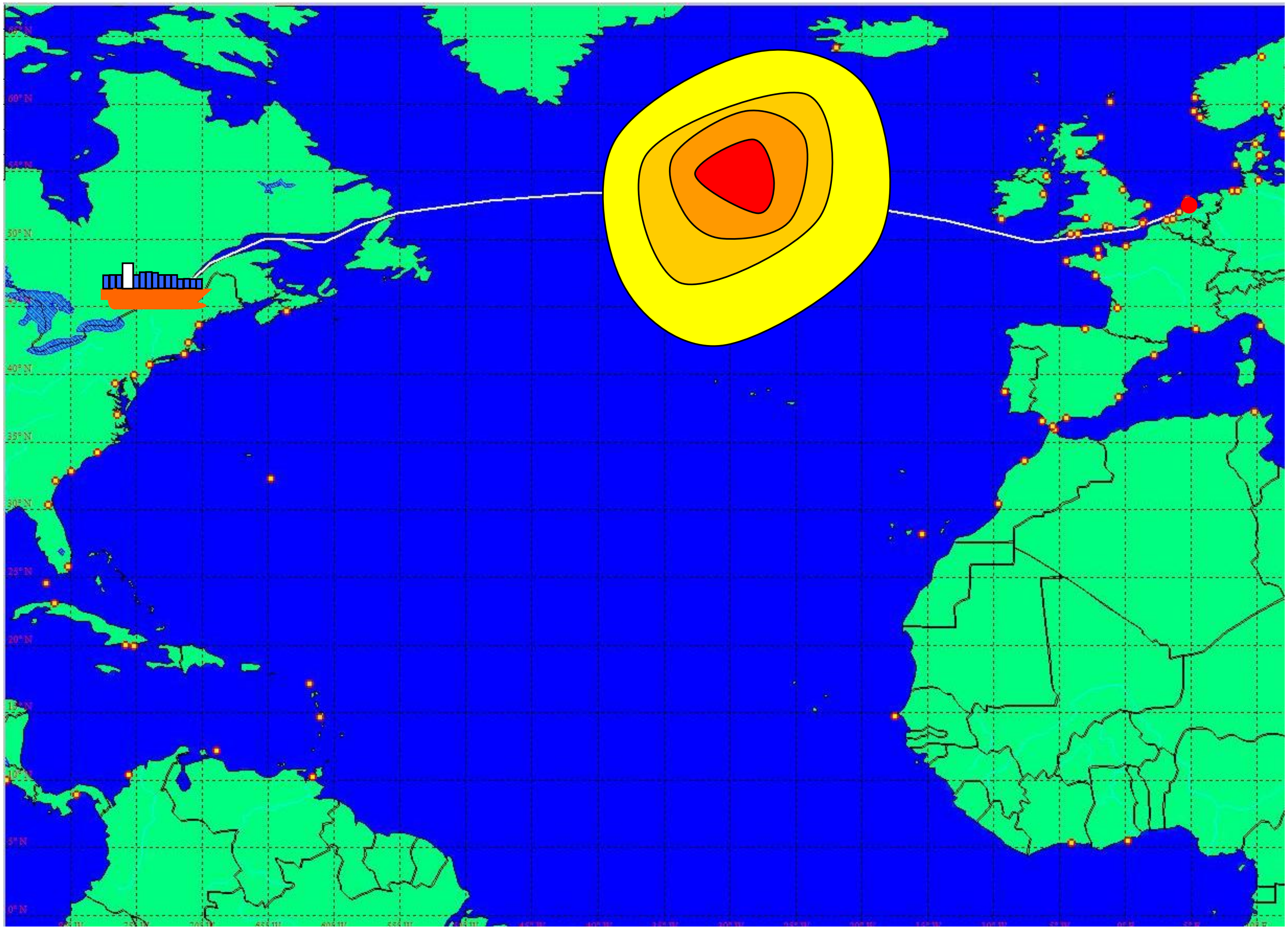
SCENARIO SIMULATIONS



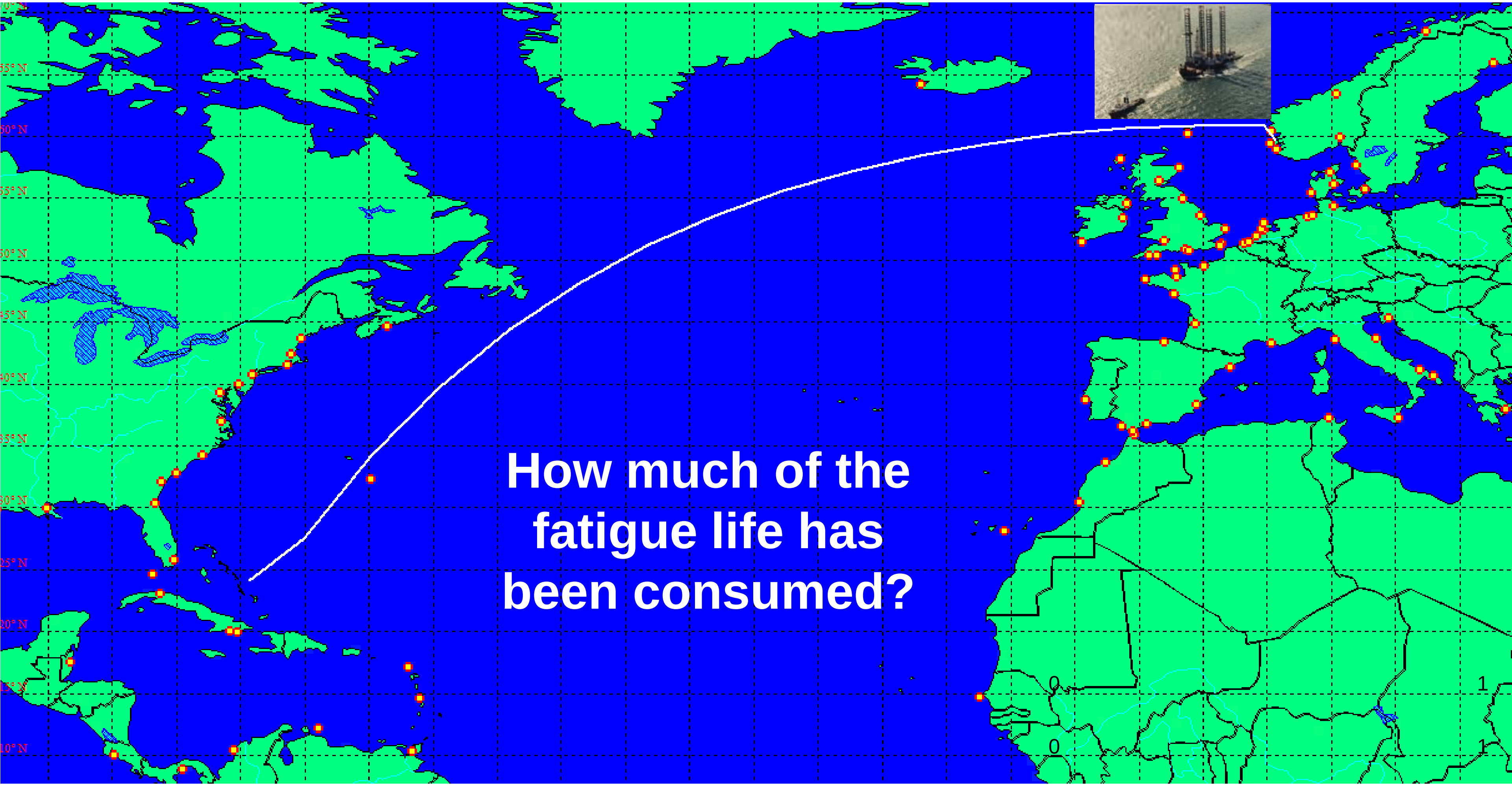
SCENARIO SIMULATIONS



SCENARIO SIMULATIONS

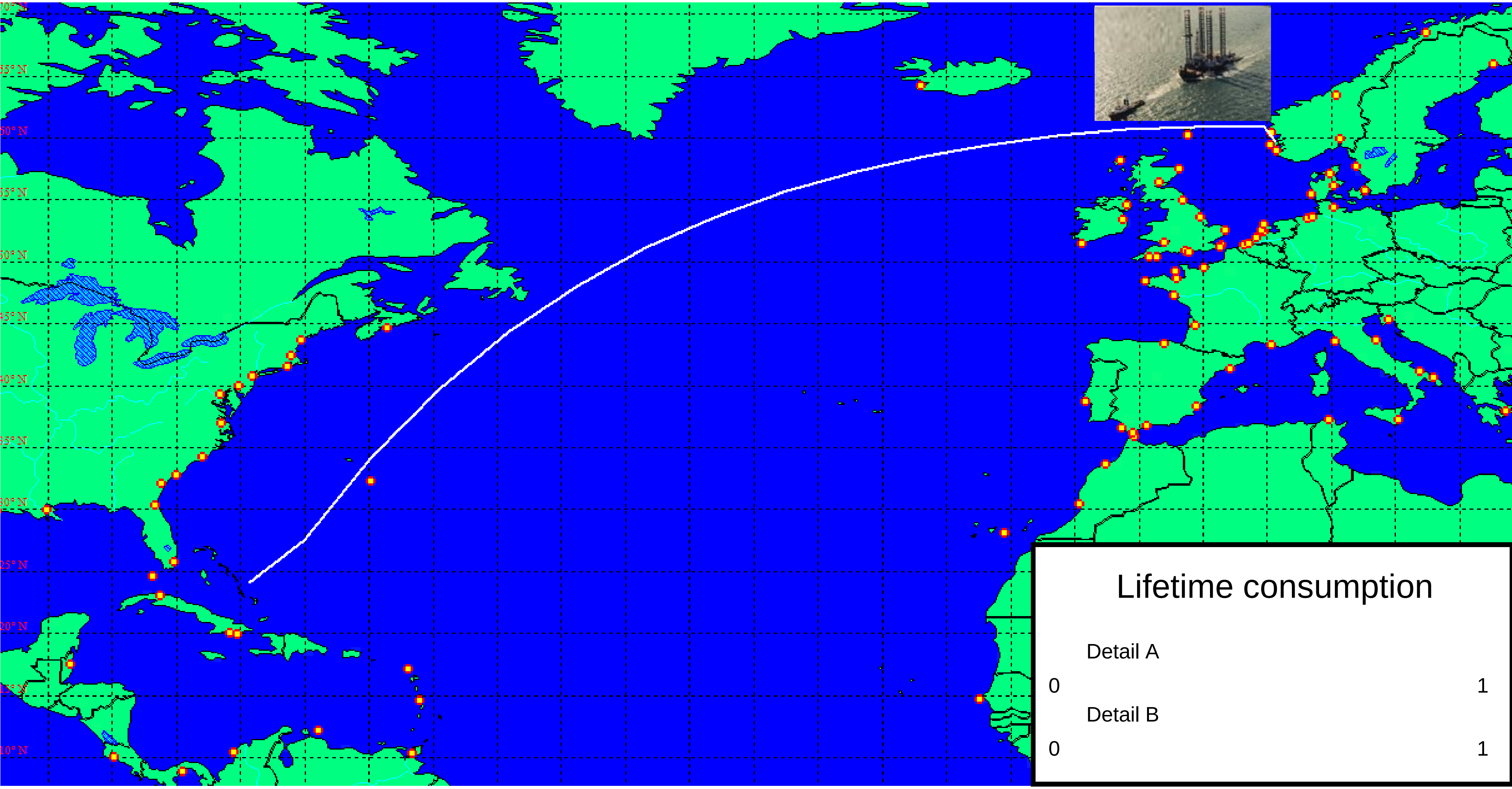


SCENARIO SIMULATIONS



How much of the fatigue life has been consumed?

SCENARIO SIMULATIONS



Lifetime consumption

Detail A

0 1

Detail B

0 1

OPERATIONAL STAGE

- Encountered conditions differ
  - from design conditions
- Advisory hull monitoring system installed to
  - determine lifetime during operations
  - explain differences with design

www.marin.nl