Modal analysis of heat exchangers: experimental, analytical and numerical approaches

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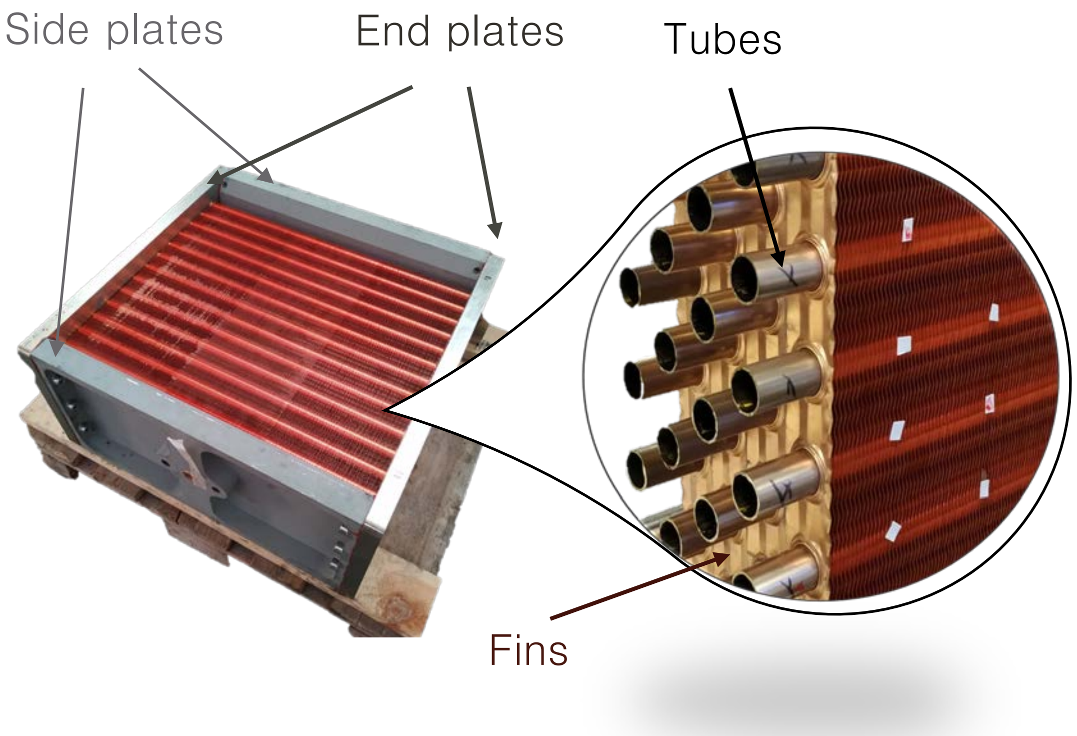
Overview

Research project and motivation

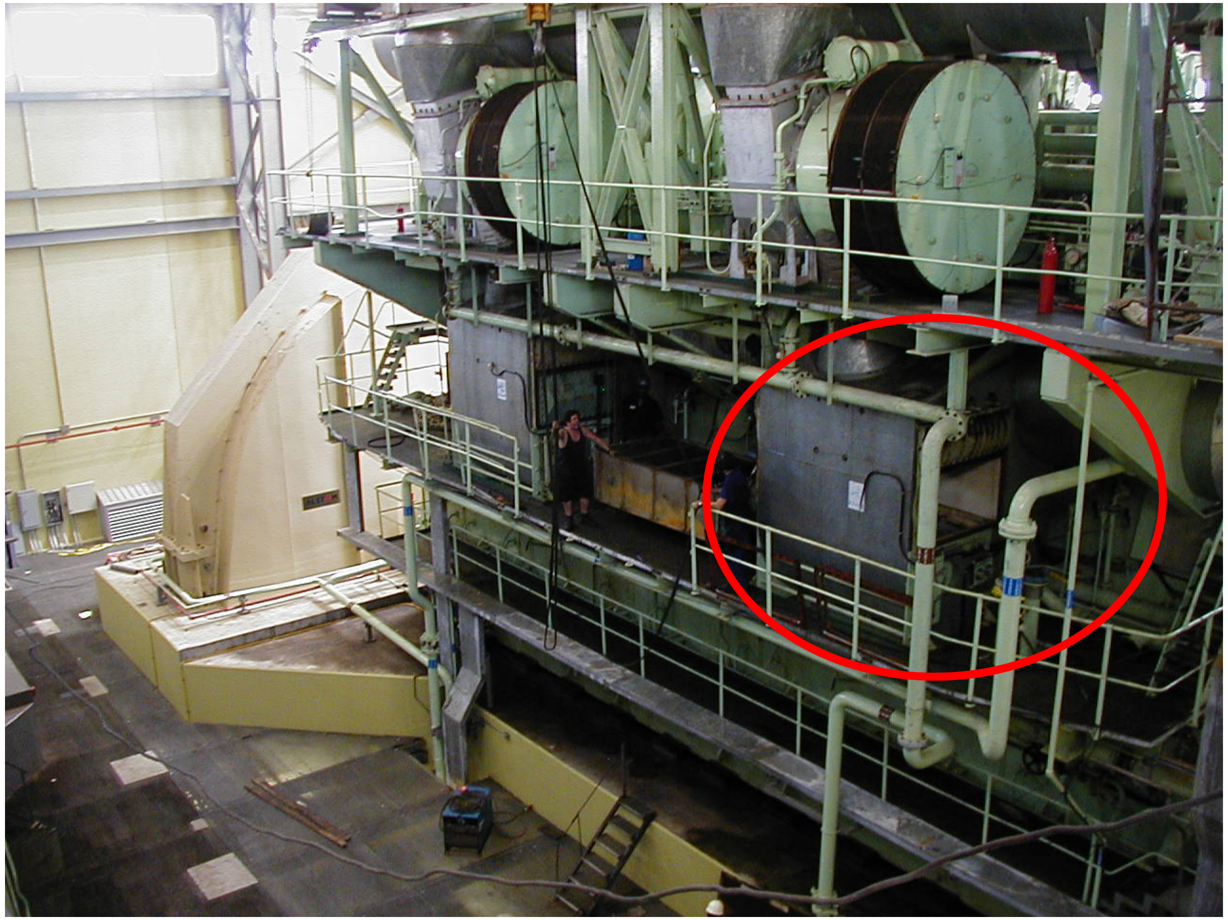
Analytical model of heat exchanger

Experiments and validation

Summary and conclusions

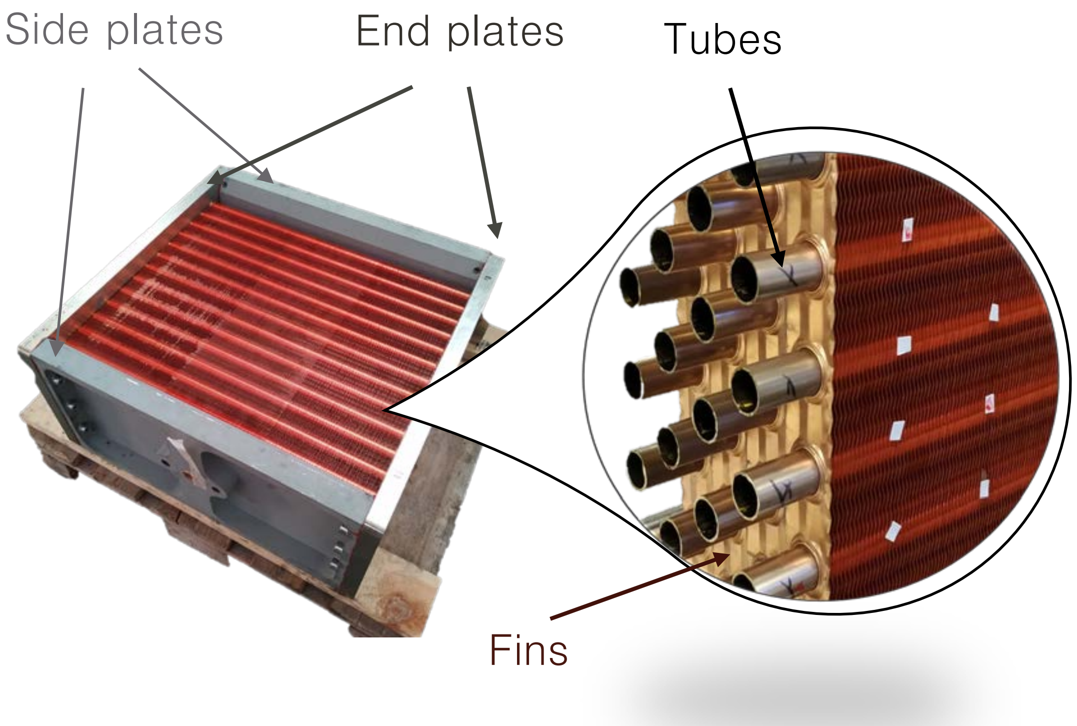


Heat exchanger components



Location of heat exchanger in a ship engine

Research project and motivation



Heat exchanger components

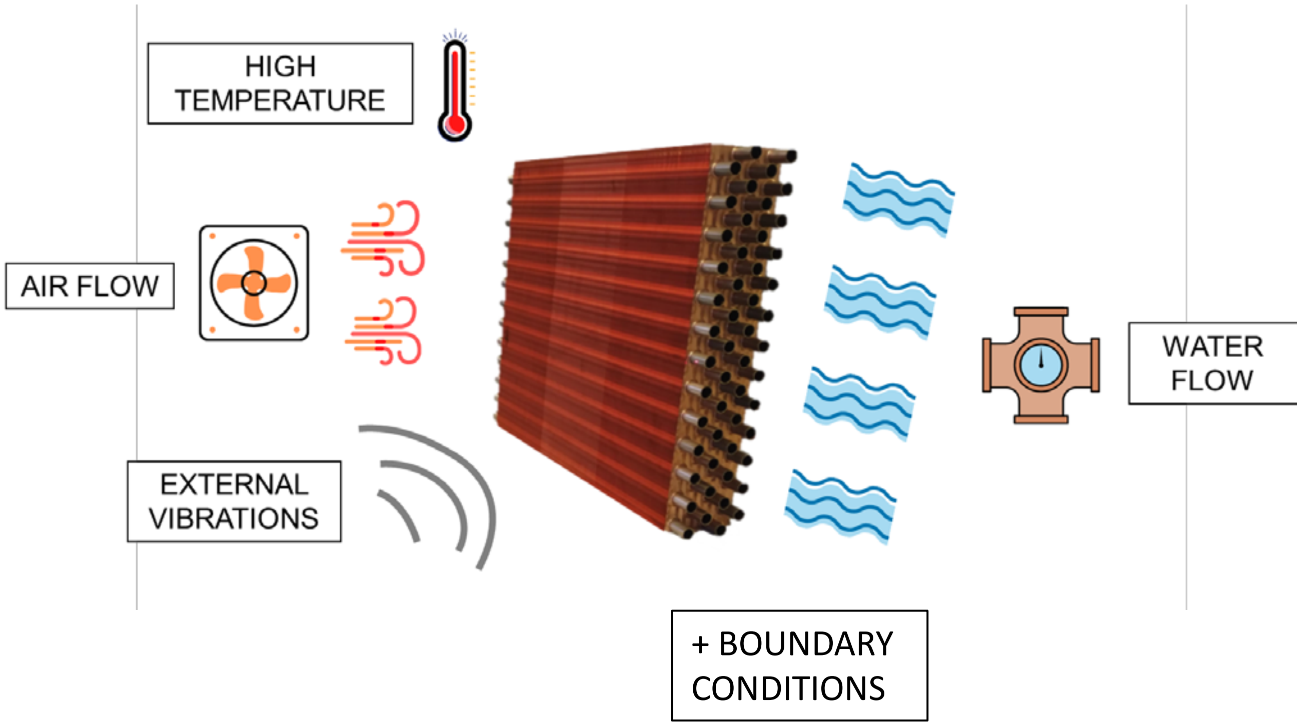


Illustration of heat exchanger in a dynamically loaded environment

Motivation

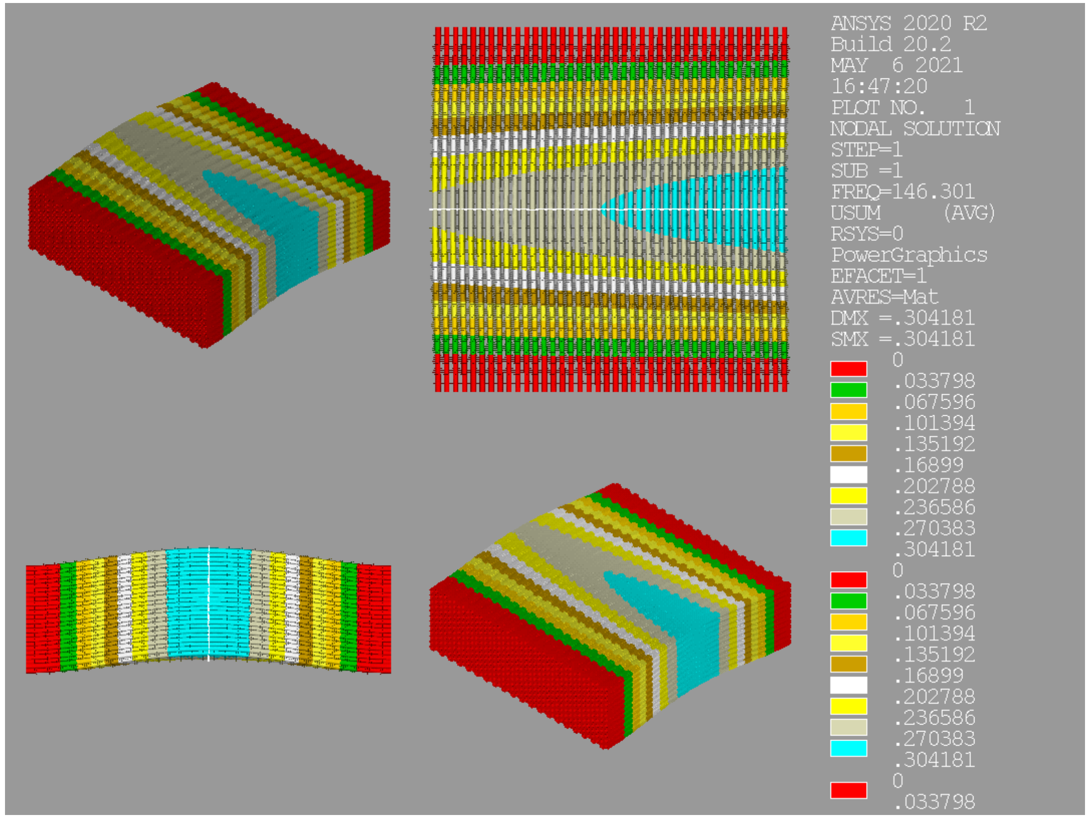
- HE modes excited by dynamic loading from the engine can cause accelerated fatigue failure
- Need to understand behaviour of lower vibrational modes of HE
- FE based analysis may be infeasible in early design phases due to *curse of dimensionality*

Introduction

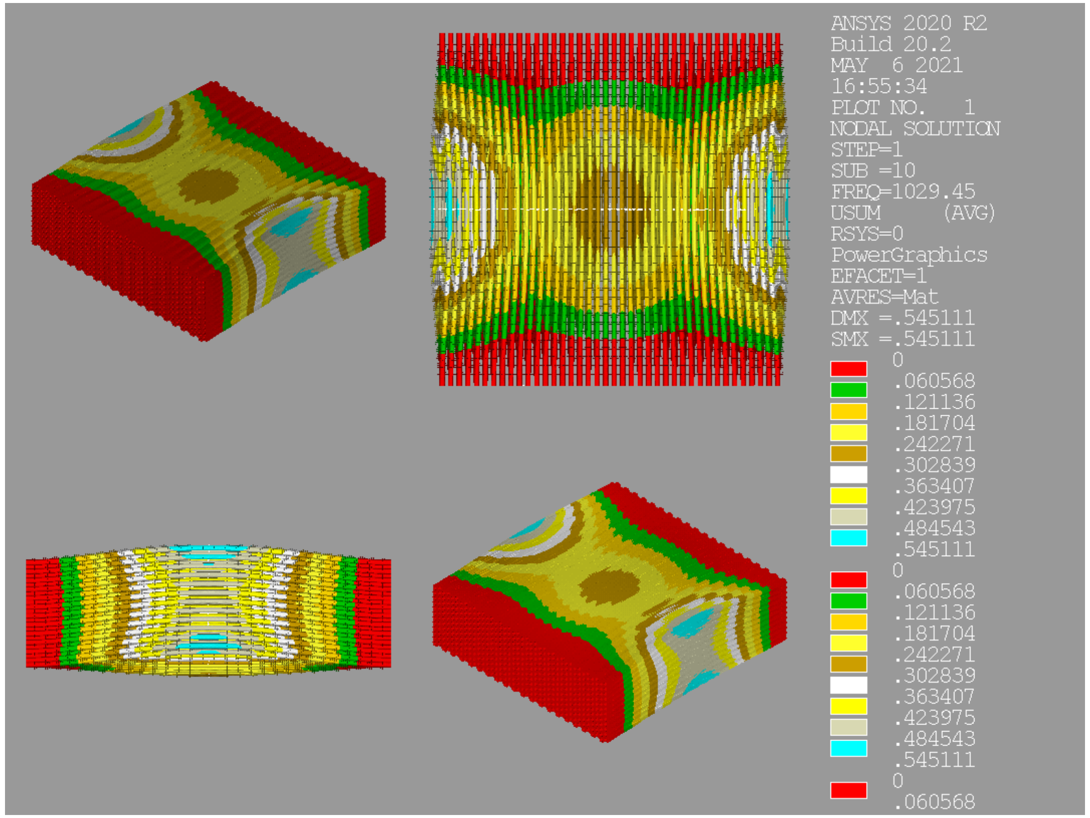
- Comparison between natural frequencies resulting from analytical model, FE simulations and experiments
- Two different heat exchanger units:
  - tested in free-free conditions
  - tested in “fixed" boundary conditions

Two types of lower core modes

Bending Type



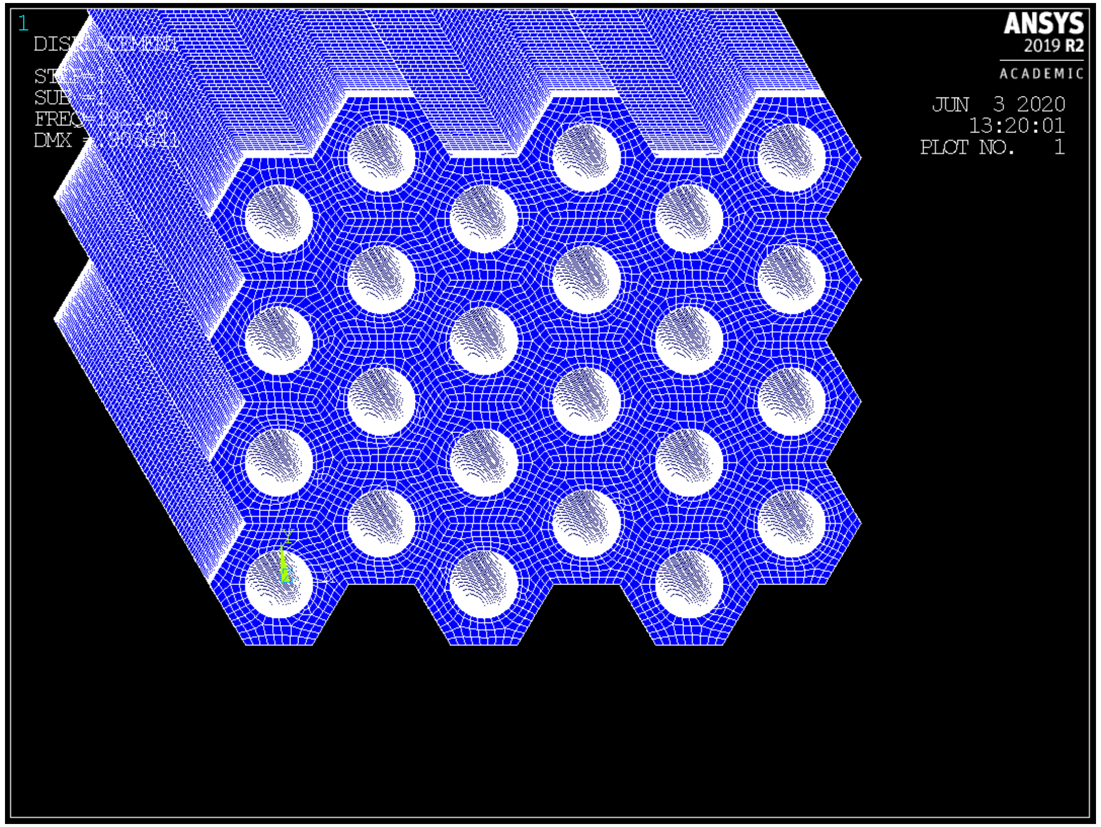
Twisting Type



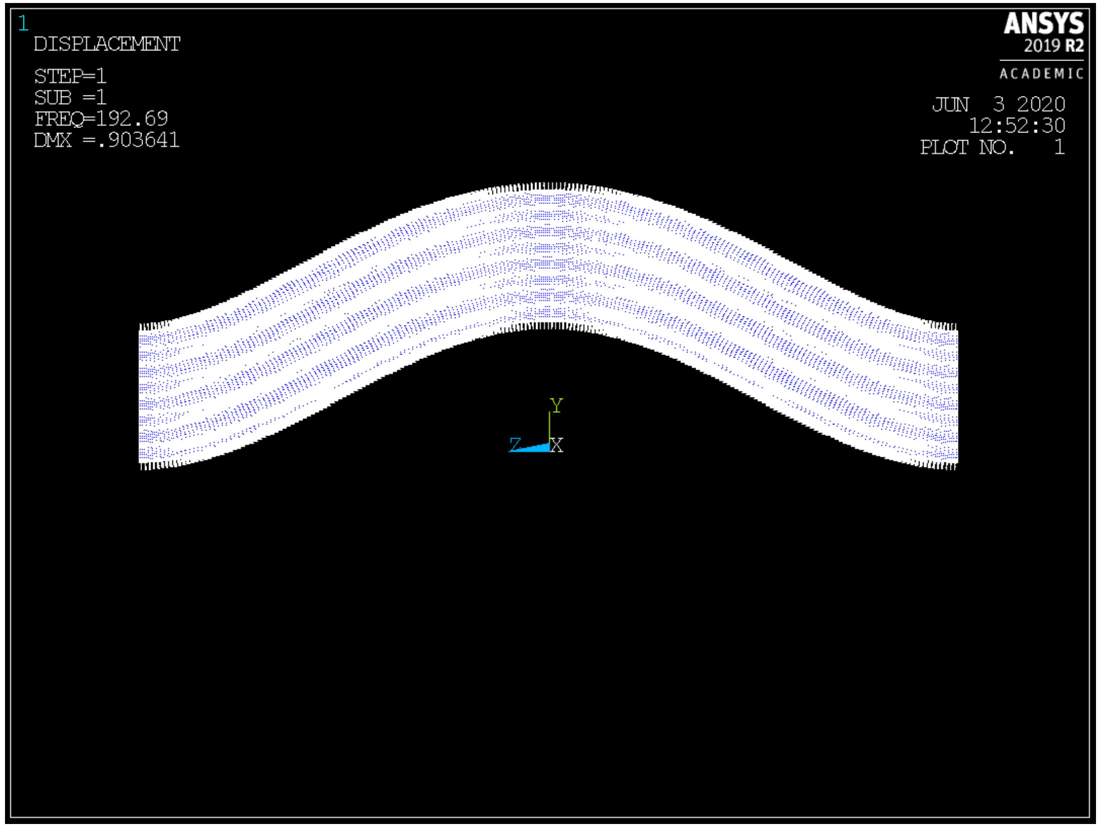
Fixed-fixed boundary condition

Bending type mode

Front



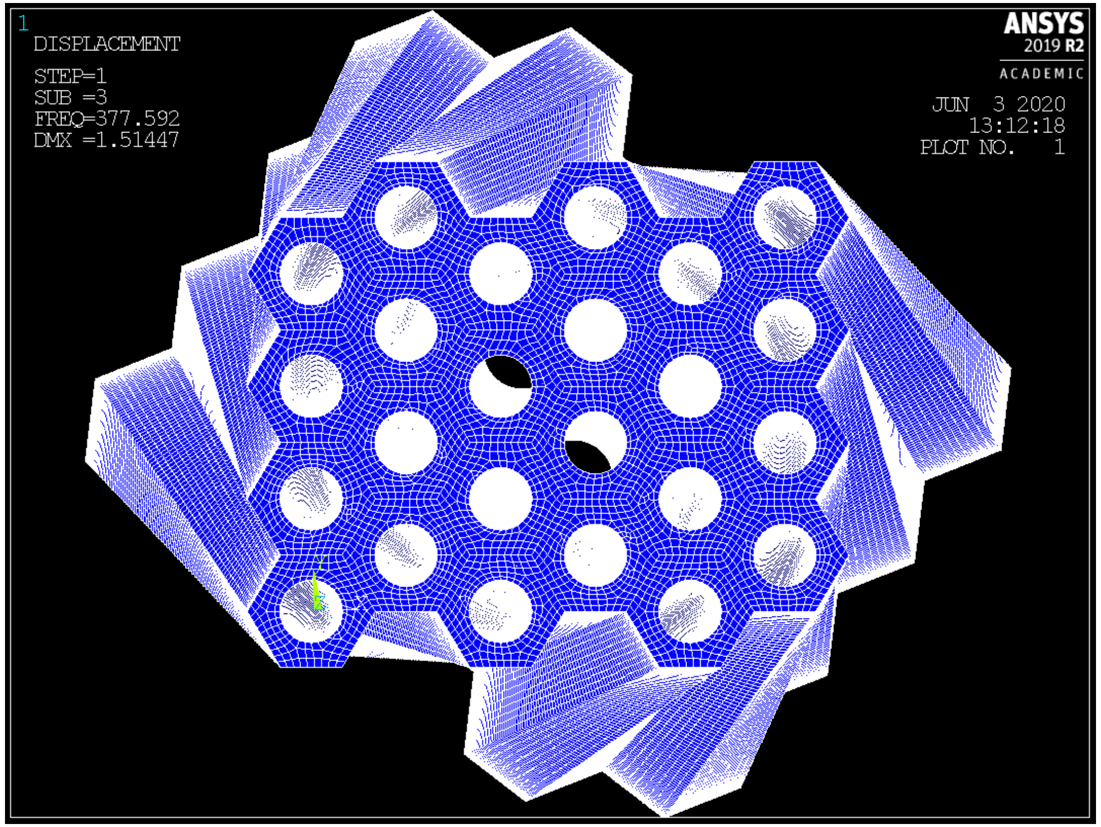
Side



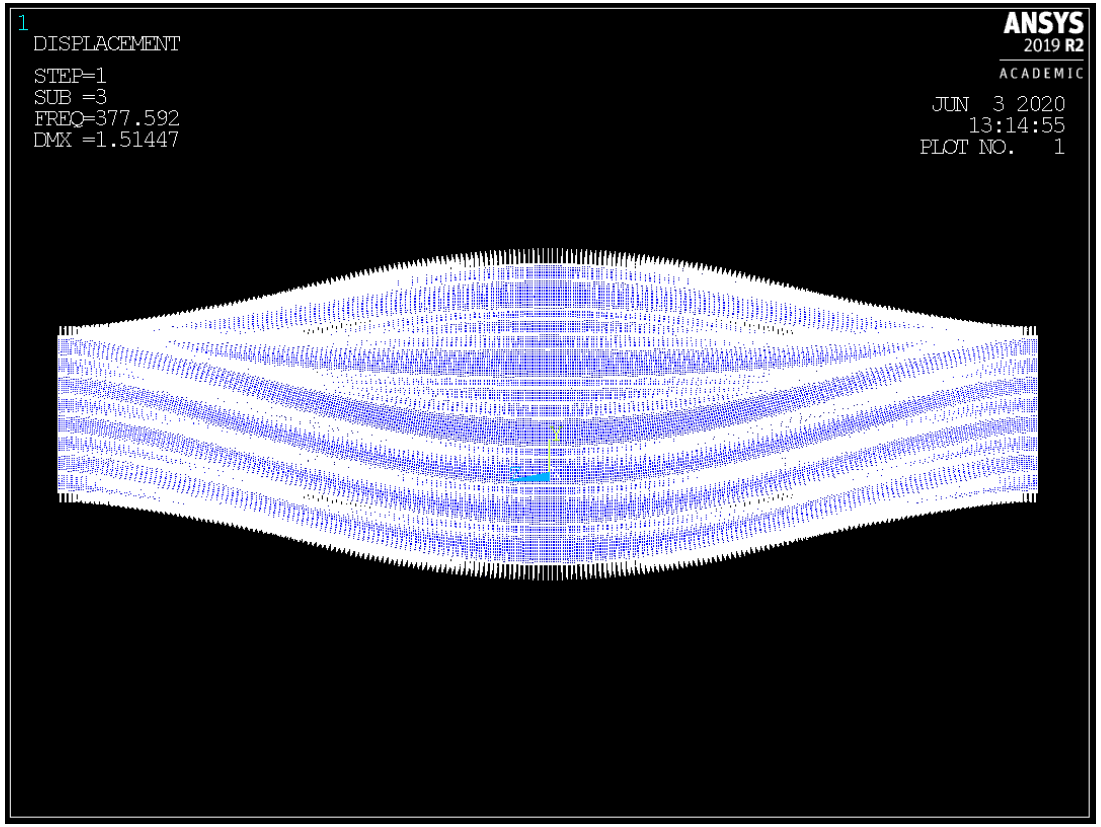
Fixed-fixed boundary condition

Twisting type mode

Front



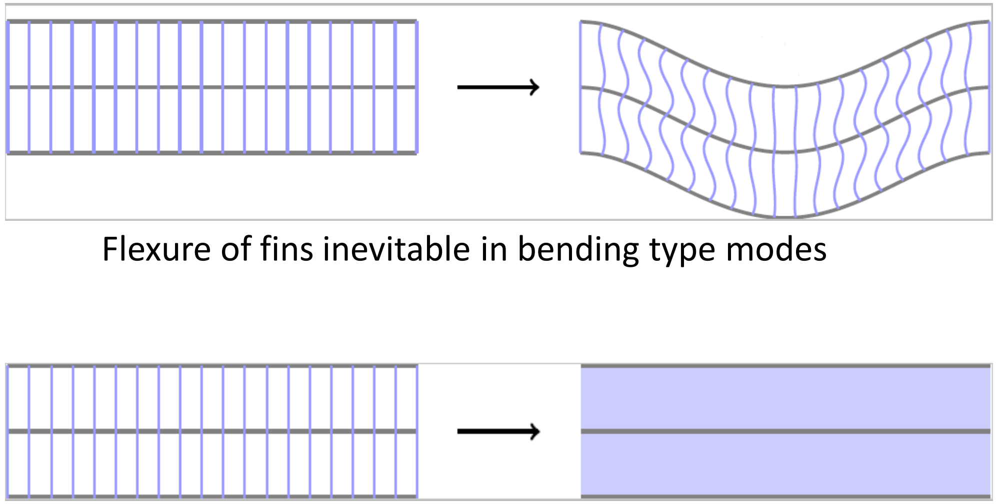
Side



Fixed-fixed boundary condition

Based on Euler Bernoulli approximation with added mass load due to fins.

Additionally, the effect of fin flexure can be taken into account.



Flexure of fins inevitable in bending type modes

Model fins as soft shear core

$$\omega^2 = \frac{\overbrace{E_T I_T C_2}^{\text{Tube Flex}} + \gamma \overbrace{\frac{12 E_F I_F C_3}{d_x L_{UC}}}^{\text{Fin Flex}}}{\underbrace{\mu_T C_1 + \mu_F \frac{d_x}{L_{UC}} C_1}_{\text{Tube and Fin Mass}}}$$

$\gamma = 0$ gives the Euler-Bernoulli estimate

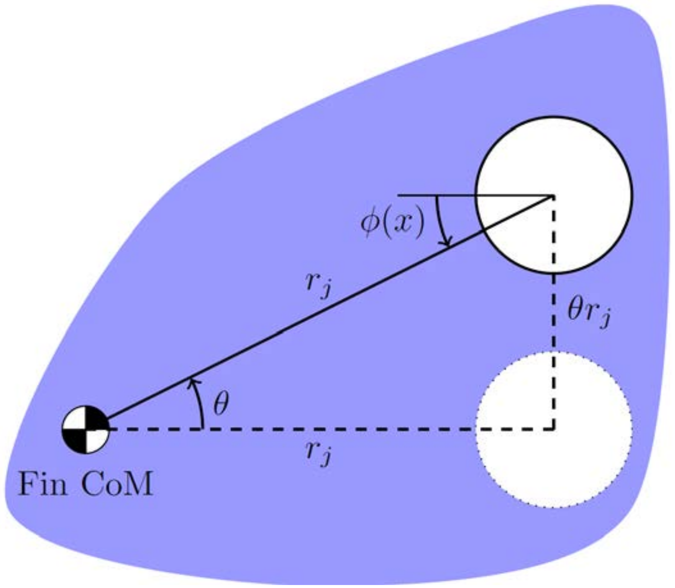
$\gamma = 1 + \epsilon$ for rectangular tube arrangement

$\gamma > 1$ for other tube arrangements

InDEStruct Model – Twisting type modes

Based on simple physics of rigid body rotation which links rotation and twist angles

Fin flexure is minimal and hence ignored



$$\omega^2 = \frac{\overbrace{E_T I_T R_{sq} C_2}^{\text{Tube Flex}} + \overbrace{N_T G_T J_T C_3}^{\text{Tube Twist}}}{\underbrace{\mu_T R_{sq} C_1}_{\text{Tube Disp}} + \underbrace{\rho_F J_{Tot} C_1}_{\text{Fin Rot}}}$$

Heat exchanger configurations

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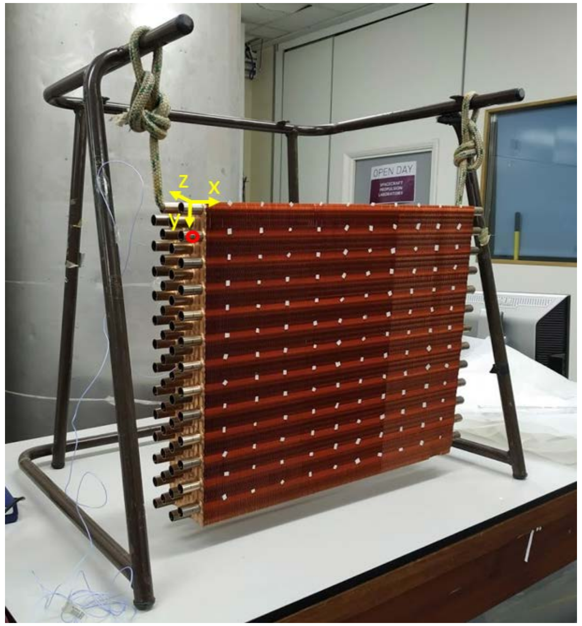
Free-Free BC

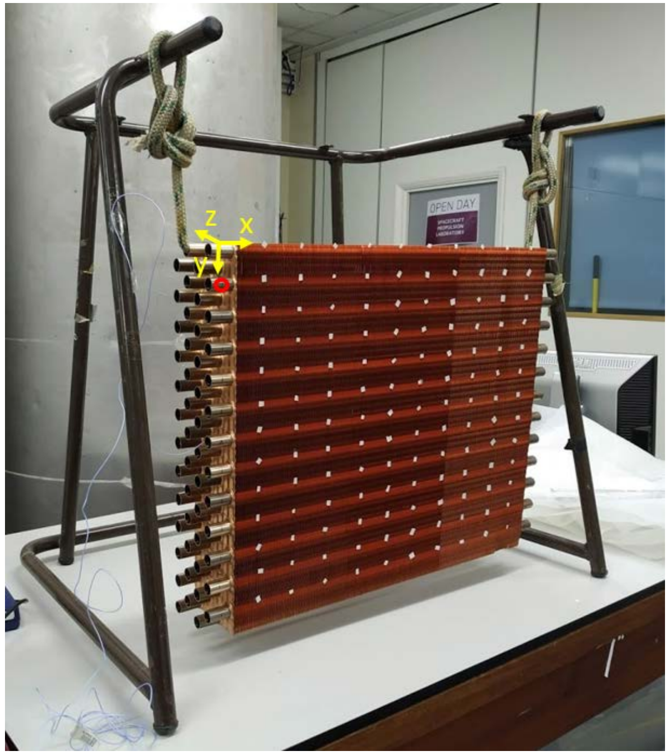
- 262 Cu fins
- 0.515m finned length
- 4 X 13 CuNi10 tubes



Fixed-Fixed BC

- 197 Cu fins
- 0.493m finned length
- 40 X 12 CuNi10 tubes

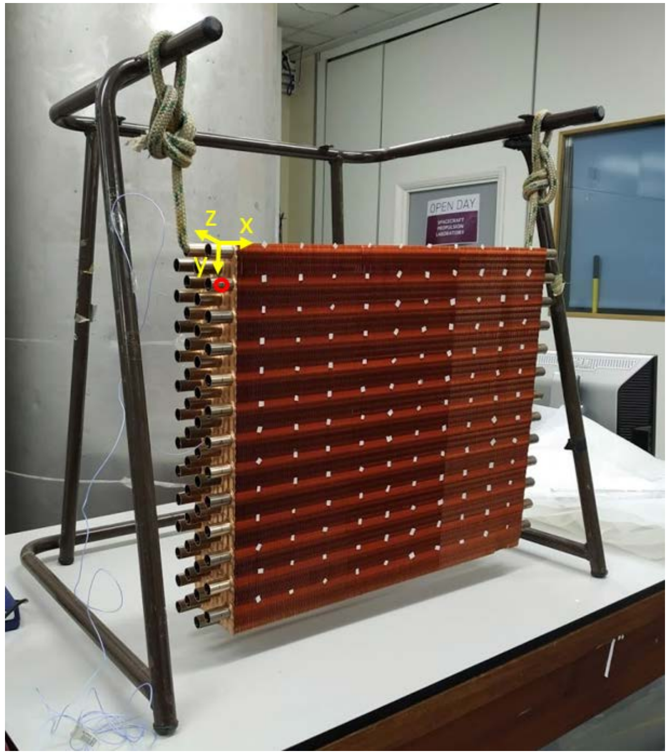
Experiments on heat exchanger with free BC



Heat exchanger under test
simulating free boundary conditions

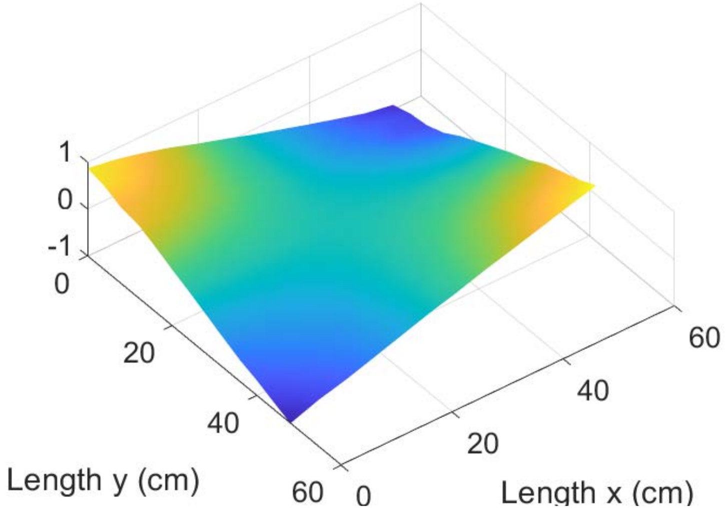
Type of test	EMA - Impact roving hammer test
Number of measured points	156
Measured direction	z
Modal parameters extraction method	Least squares global polynomial method
Max frequency	2 kHz

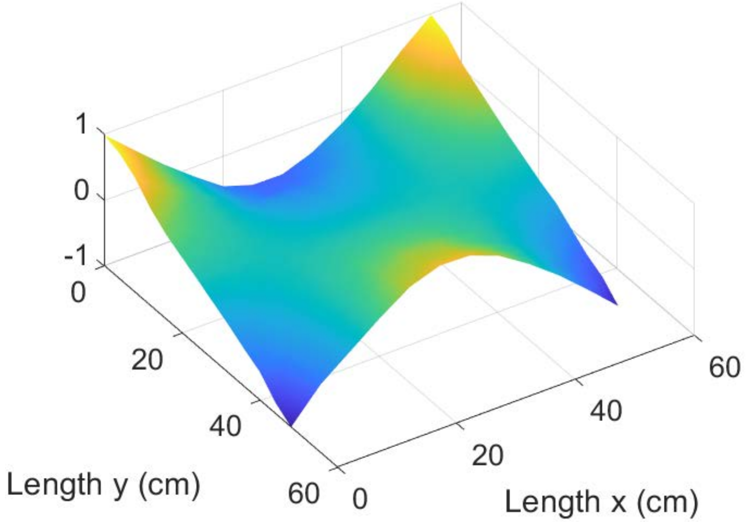
Experiments on heat exchanger with free BC



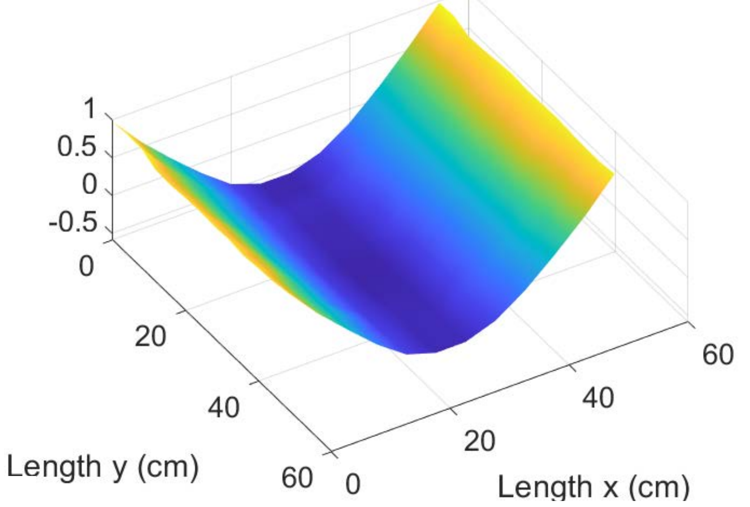
Heat exchanger under test
simulating free boundary conditions

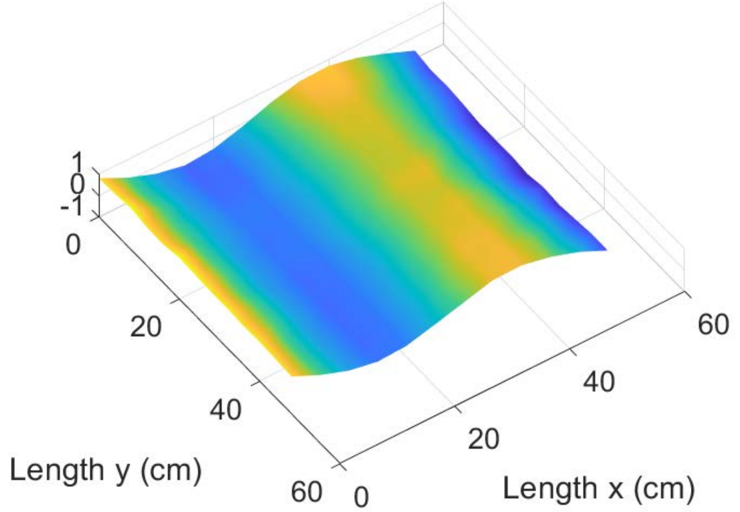
Twisting Type

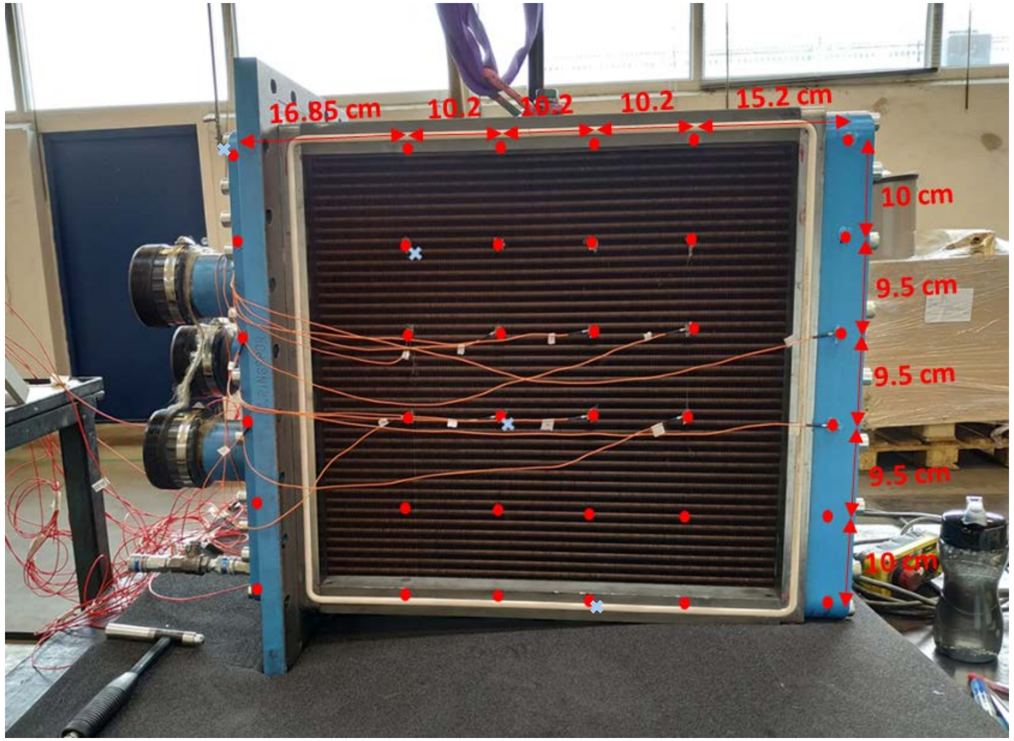




Bending Type







Heat exchanger under test for the case of fixed boundary conditions

Excitation only at heat exchanger core

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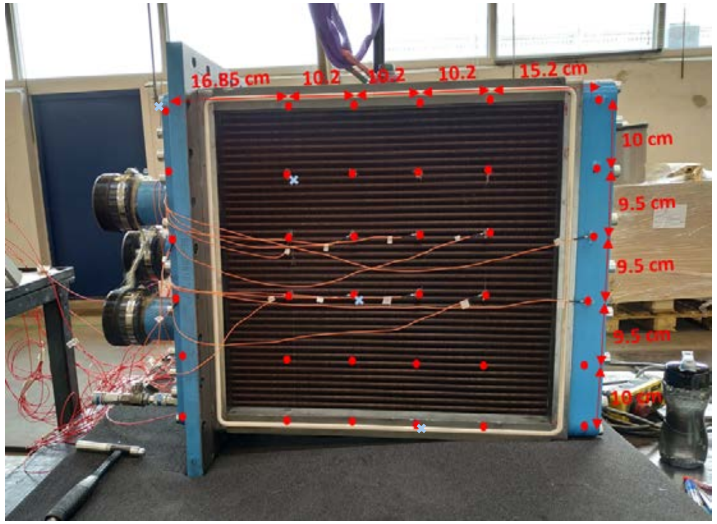
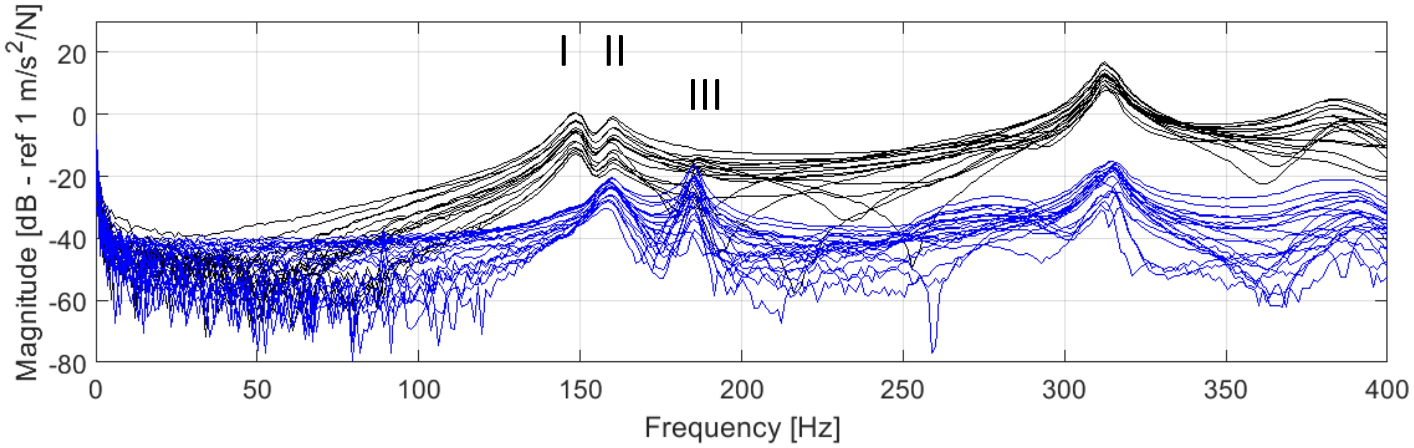


motion of covers

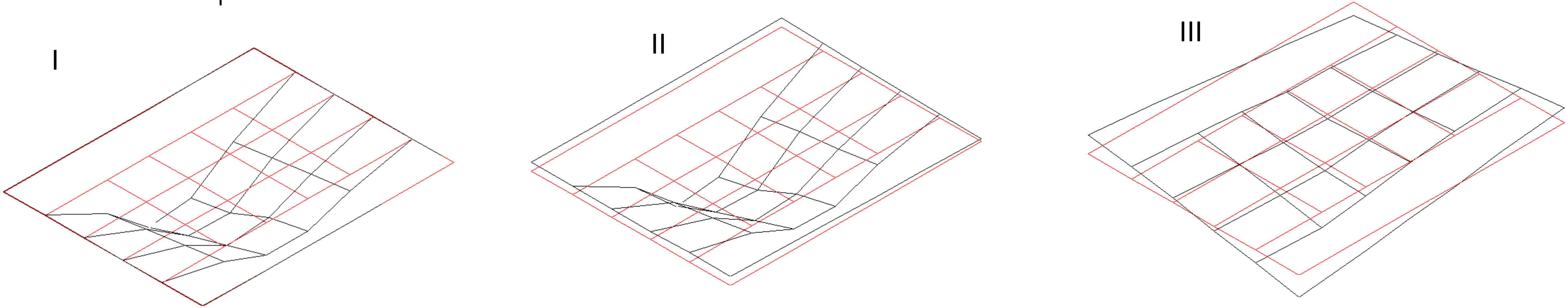


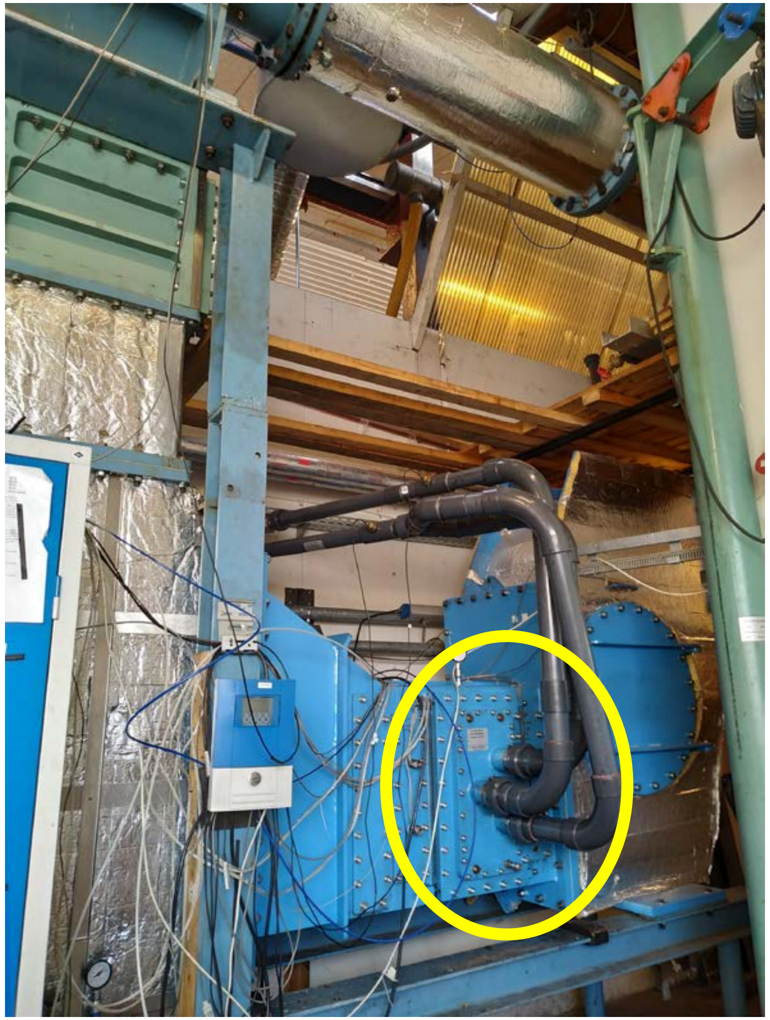
mode shapes

I

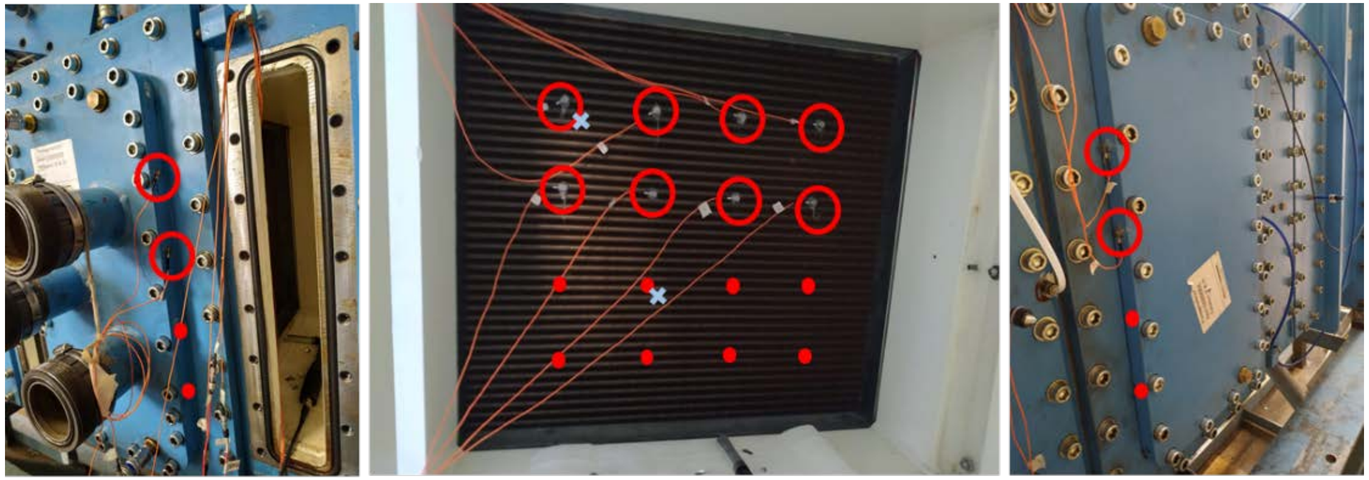
II

III

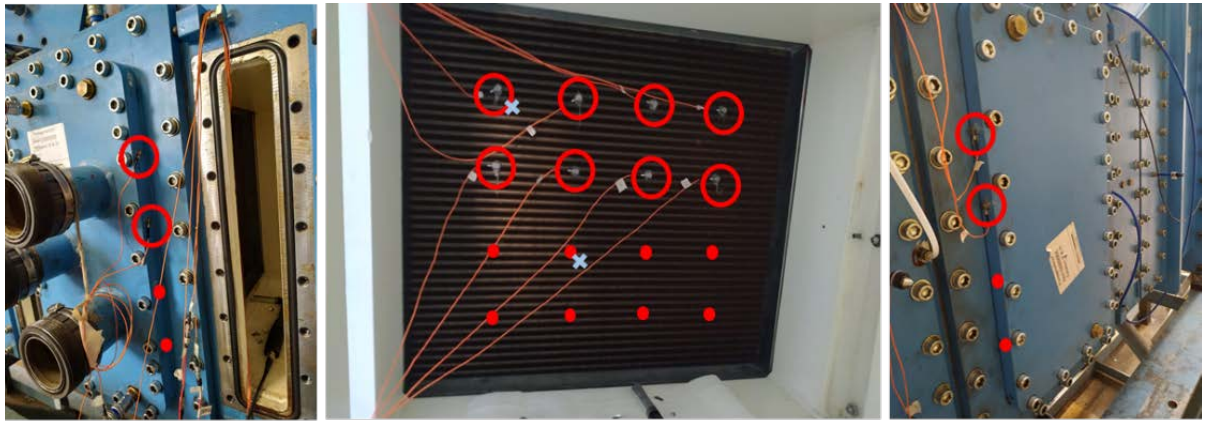




Test rig at Vestas aircoil

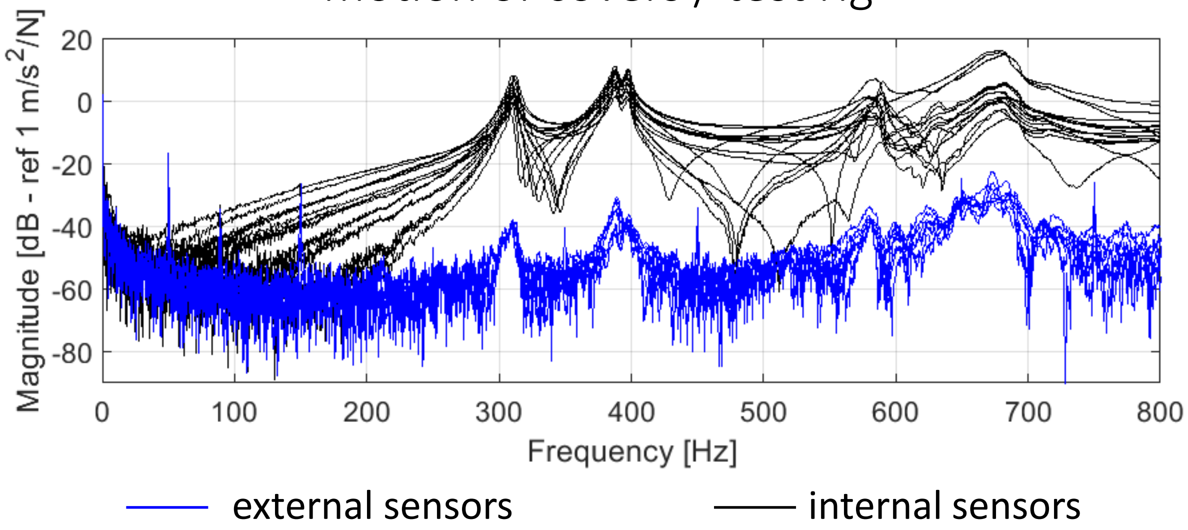


Heat exchanger under test inside the test rig

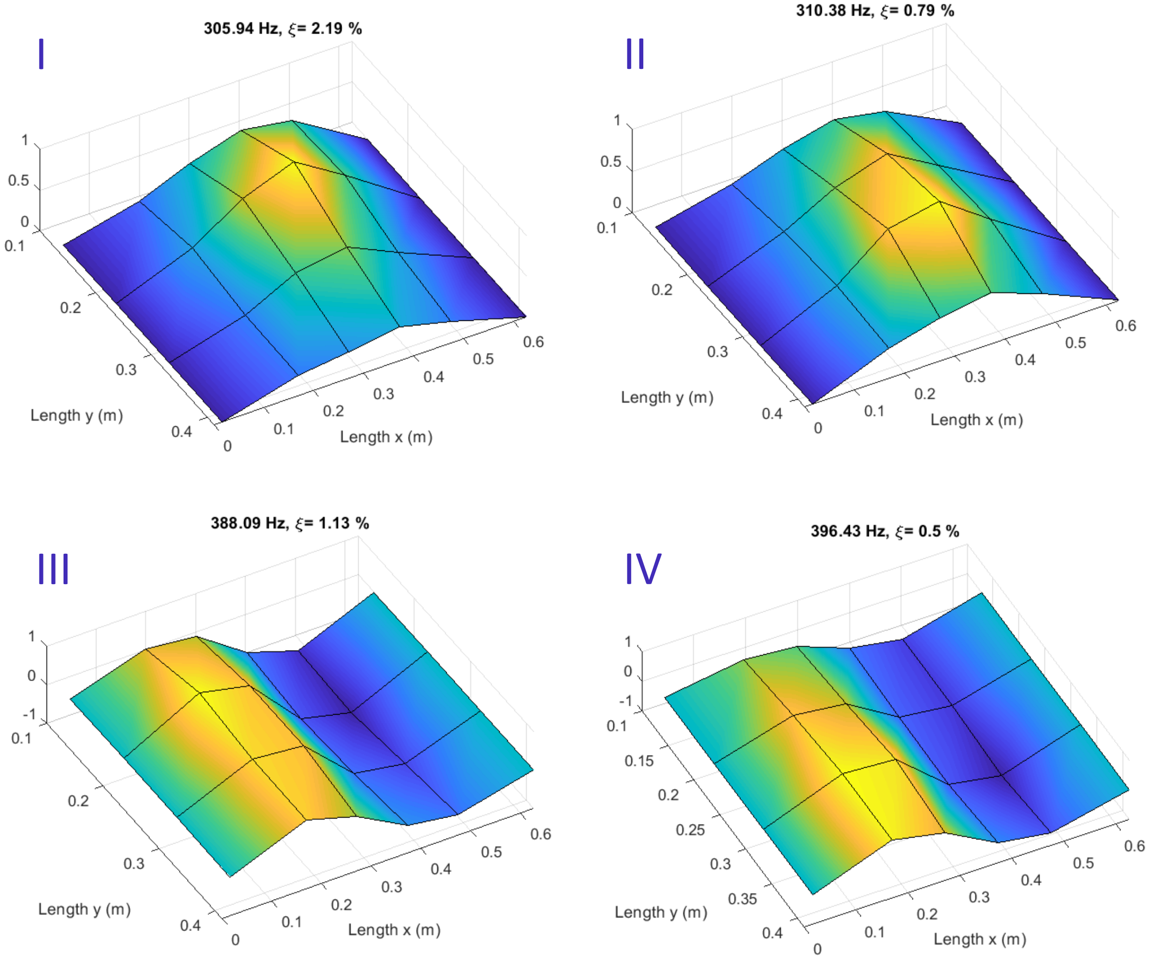


Heat exchanger under test inside the test rig

motion of covers / test rig

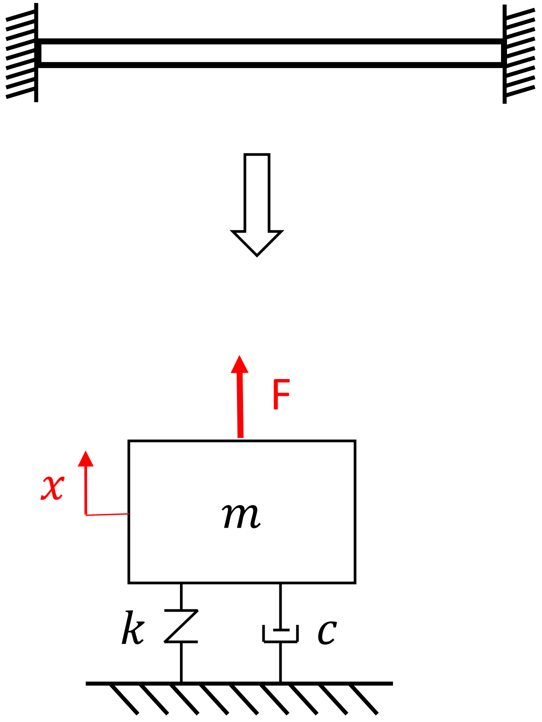


“asymmetric” mode shapes



Ideally – if we had fixed BCs

Beam model

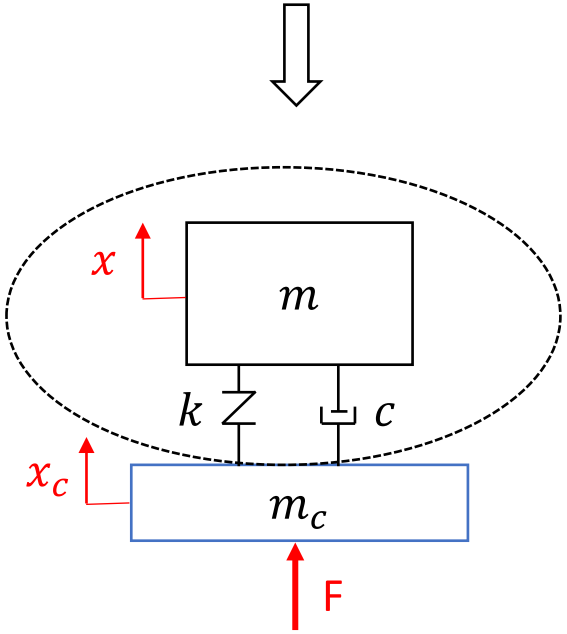


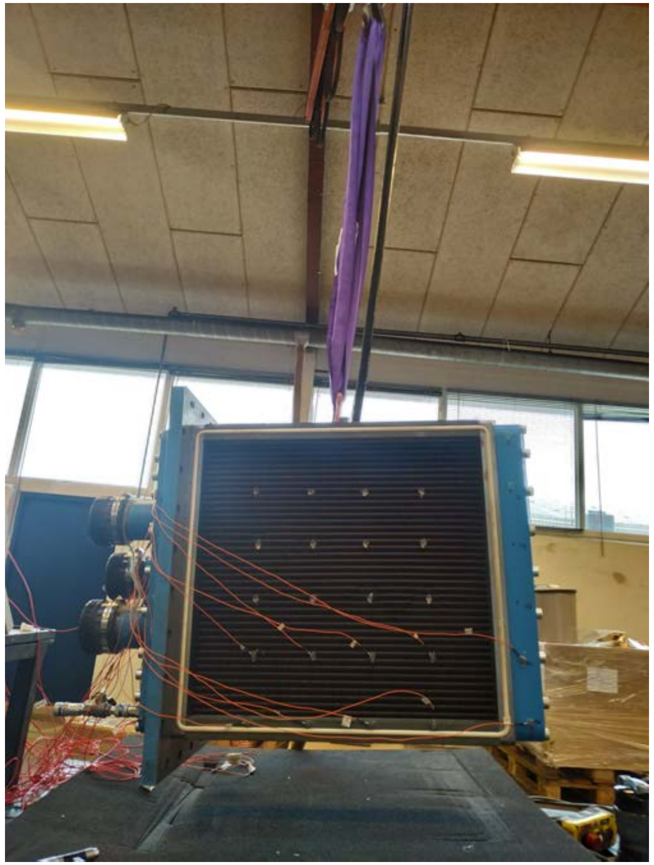
Lumped parameter model

Peaks of receptance, mobility, accelerance.

Approximate cover to lumped mass

Beam model



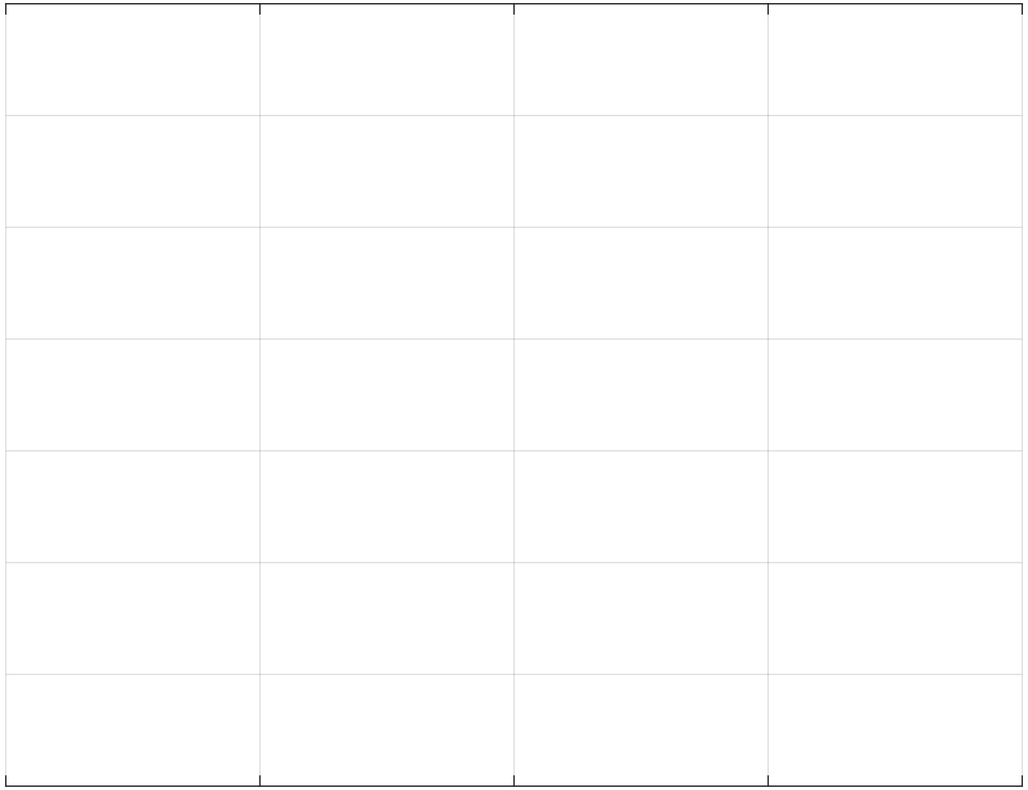


1- Dynamic stiffness or impedance at the cover

2- Transmissibility between cover and core sensors

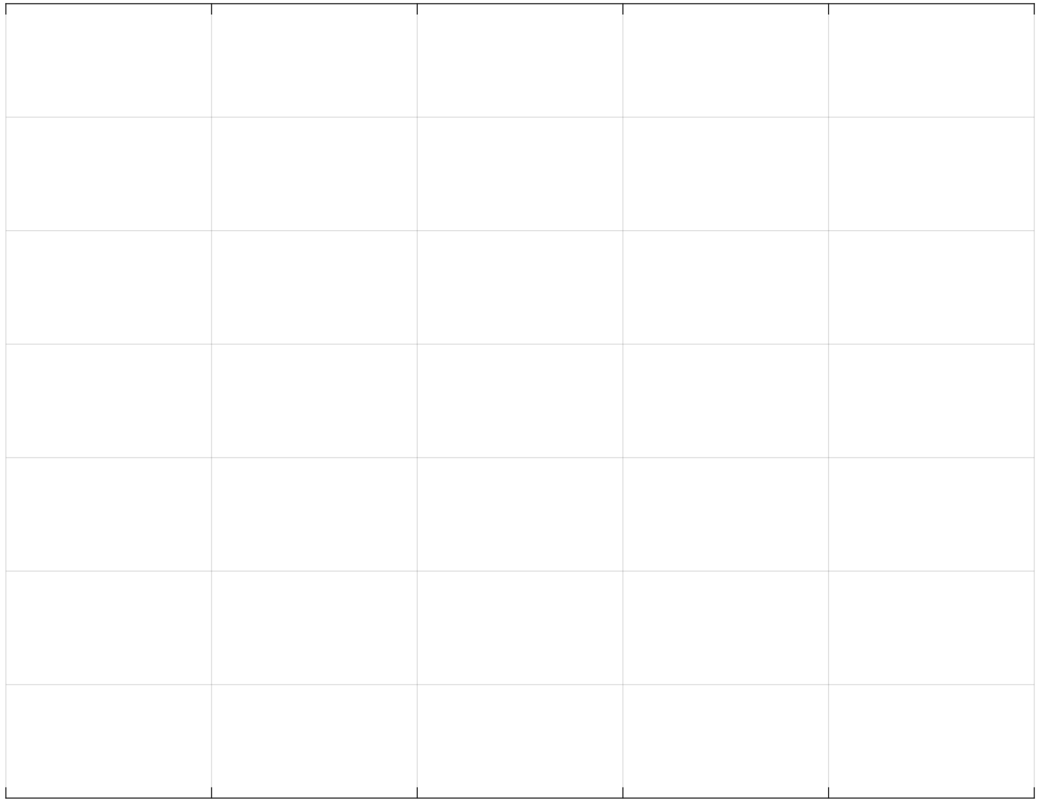
Experiments on heat exchanger with fixed BC

Transmissibility



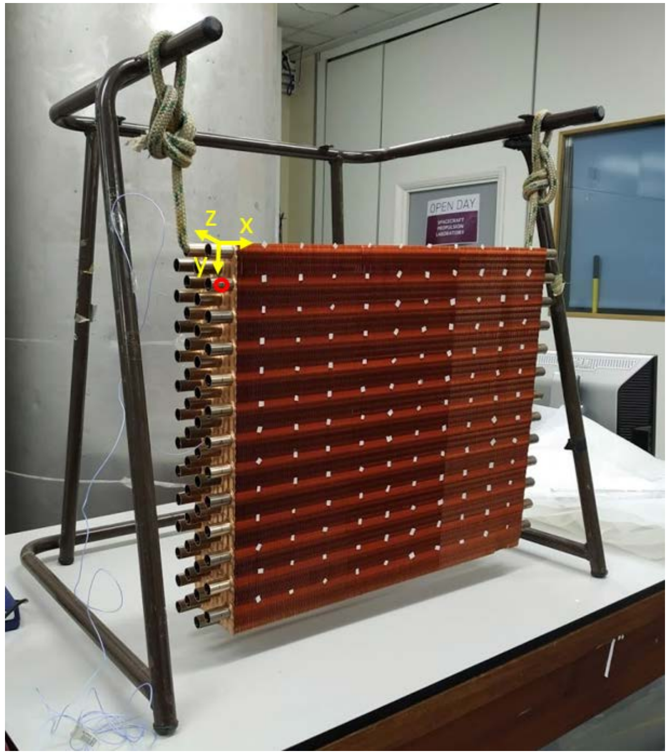
Frequencies: 146.5, 149, 167 Hz

Impedance



Frequencies: 146.5, 149, 168.5 Hz

Validation of analytical results: Free–free case



Heat exchanger under test
simulating free boundary conditions

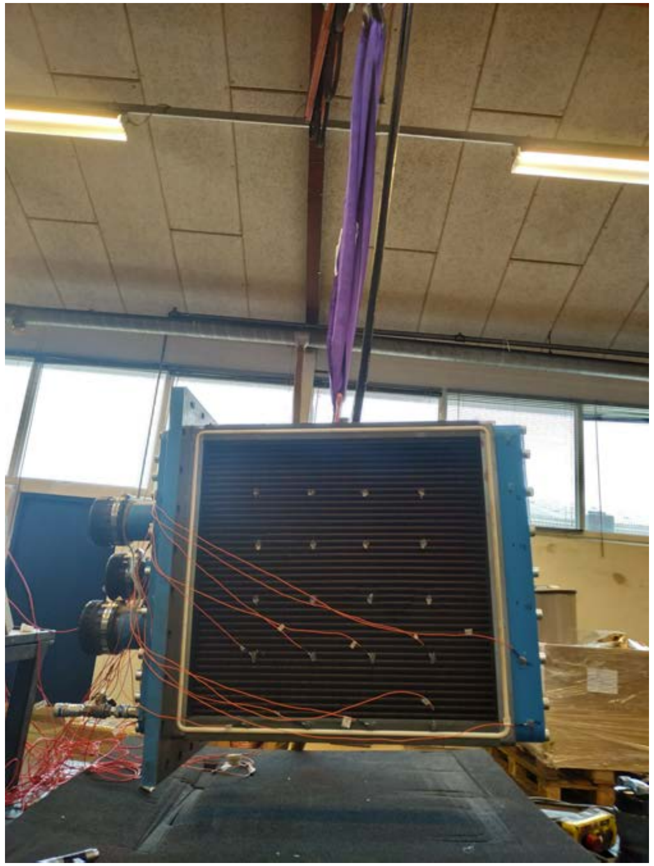
Bending Type Modes

FE Sim		Model ($\gamma = 0$)		Experiments		Model ($\gamma = 1$)		Model ($\gamma = 7.4$)	
228	229	173	-24%	224	-2%	182	-20%	229	0%
567	570	477	-16%	498	-12%	484	-15%	526	-7%

Twist Type Modes

FE Sim	Model		Experiments	
93	91	-2%	81	-13%
292	253	-13%	265	-9%

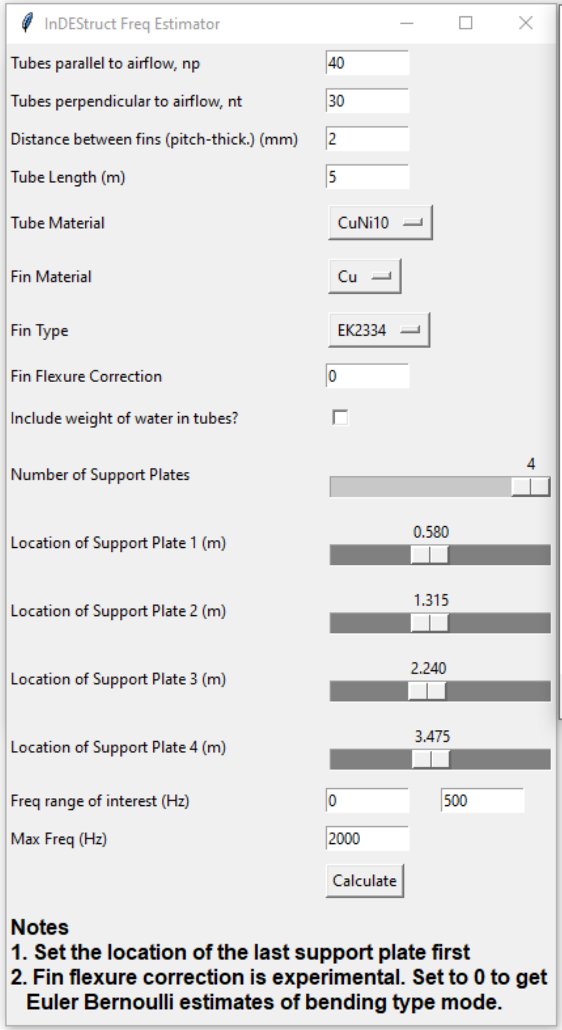
Validation of analytical results: in–situ case

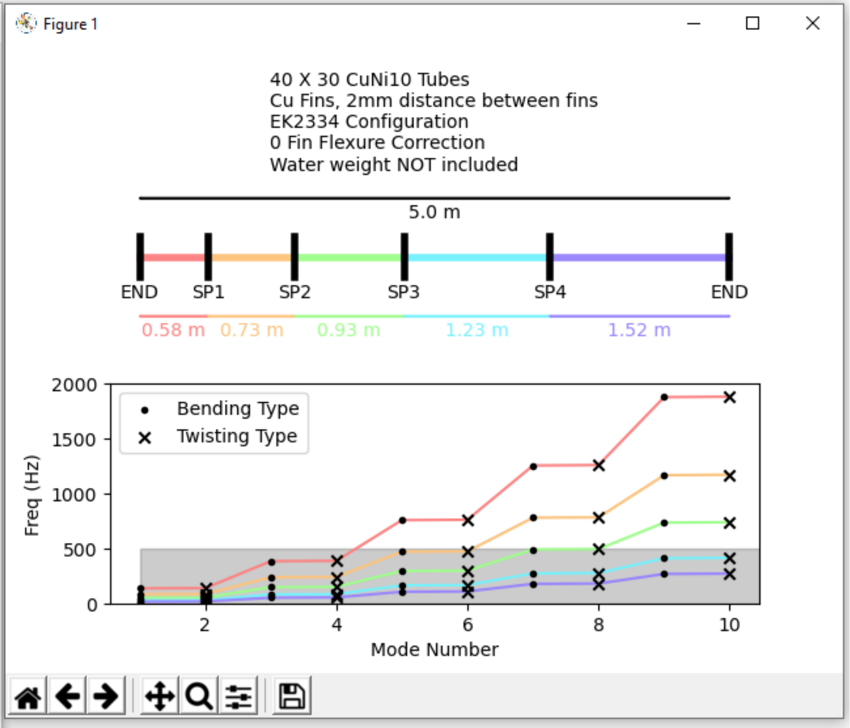


Heat exchanger under test simulating fixed boundary conditions

Mode Type	FE Sim (Hz)	Model (Hz)	Experiments (Hz)
Bend 1	146, 150	148	~~306~~ 146.5, 149
Twist 1	156	135	X
Bend 2	369, 375	374	~~388~~
Twist 2	384	354	X
Bend 3	691, 698	702	X
Twist 3	708	680	X
Bend 4	1111, 1119	1136	~~901~~

Deployment of analytical model: a GUI interface





Summary and conclusions

- Fin flex energy correction gets model and FE results to agree
- Percentage error between FE and experiments of 2 - 13% for free-free case (FE results are always higher)
- Percentage error between updated analytical model and experiments of 2 - 12% for free-free case
- Boundary conditions are a challenge. Business call needs to be taken how important it is to understand the detailed dynamics of Heat Exchangers

Acknowledgements

- Prof Anders Brandt and SDU for training and equipment provision
- Frank Nielsen and Søren Madsen for their help with setting up the tests