



UNIVERSITY COLLEGE LONDON

MENG MECHANICAL ENGINEERING

MECH0074 ENGINEERING IN EXTREME ENVIRONMENTS

EXTREME PRESSURE COURSEWORK

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Part I

Review of expansion loops in the energy sector

Part II

Technical assessment of water hammer for expansion loops

1 Frequency of the water hammer pressure fluctuation in a pipe

1.1 MATLAB script

1.1.1 Pressure fluctuation and Joukowsky prediction

The calculation for maximum magnitude of the water hammer pulse is (1). This is also known as the Joukowsky equation.

$$\Delta p = \rho a_0 \Delta v \text{ (Pa)} \quad (1)$$

where Δp is the change in pressure, ρ is the fluid density, a_0 is the sonic velocity in the pipe and Δv is the change in fluid velocity. Running the MATLAB script with the values shown in Table 1, Figure 1 was generated.

The pressure values at the first peak of the rise were chosen to find the maximum pressure differential. A settled pressure differential value was also found (shown in Figure 2). Table 2 shows the maximum pressure differentials for each index point (s - value: point along length of pipe from valve) and the pressure differential of the settled value.

The Joukowsky prediction makes assumptions in order to calculate a value for our pressure differential. This may explain the discrepancy we see for our differential values. For example, the Joukowsky prediction does not account for frictional losses within the system. We also assume there is no column separation, cavitation or any other fluid effects.

Parameter	Value
a_0	1100 m s^{-1}
ρ	455 kg m^{-3}
Δv	1.5 m s^{-1}
$\Delta p_{joukowsky}$	750 750 Pa

Table 1: Values for Joukowsky Prediction and calculated pressure differential.

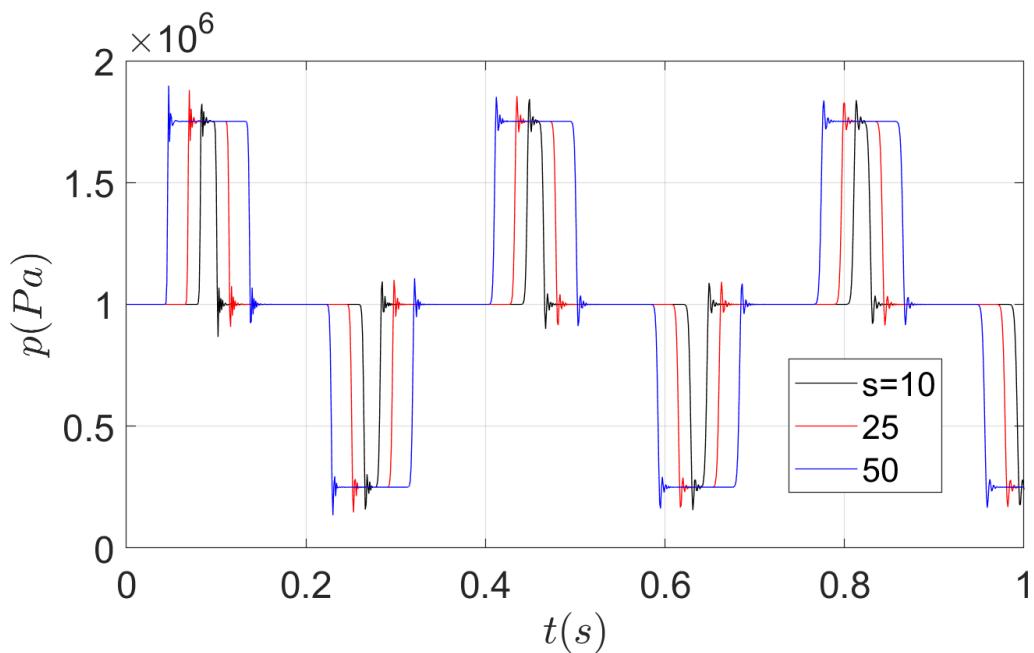
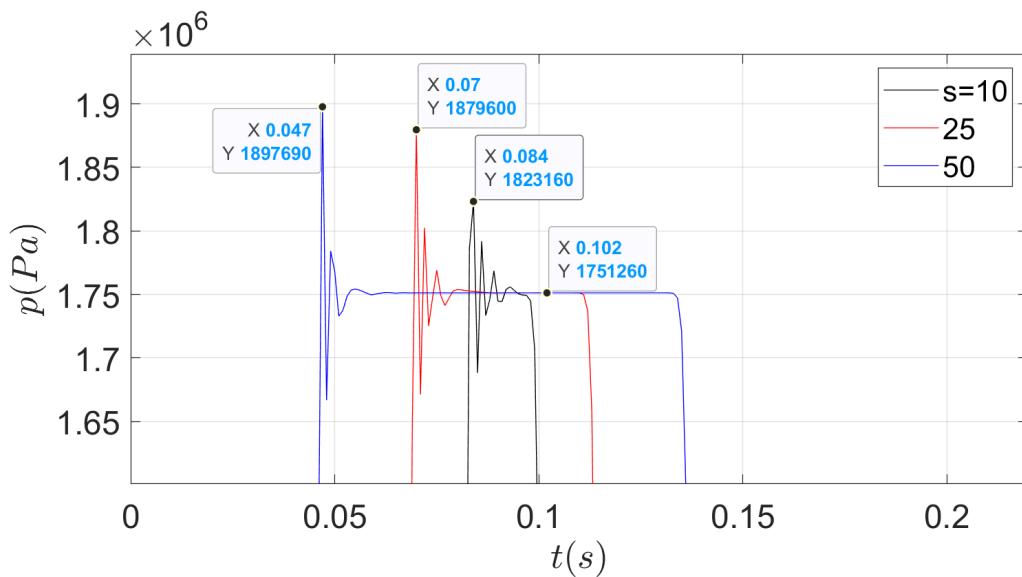
Figure 1: MATLAB script results showing pressure against time for $s = 10, 25, 50$ m.

Figure 2: Index points for finding maximum pressure differential and settled pressure differential.

s- value	Maximum pressure differential/Pa	Percentage difference compared to analytical
10	823160	8.80%
25	879600	14.65%
50	897690	16.37%
Settled value	751260	0.08%

Table 2: Pressure differentials for indexed s - values and settled values.

1.1.2 Frequencies associated with pressure fluctuation and analytical prediction

An analytical result for the frequencies of the pressure fluctuation can be found using (2)¹.

$$T = \frac{1}{f} = \frac{2L}{c} = \frac{2 \cdot s}{1100} \quad (2)$$

Indexing the values found in our MATLAB result, we can find the frequencies of the pressure fluctuation in Table 3. The index points were selected as the first peak of the rise and the first peak of the return to 1 MPa. The time between these points represents the duration of the pressure rise from the wave propagating past a particular point in the pipe. The results show a relatively low percentage error. Our discrepancy may stem from a lack of temporal resolution, as we are only using a time step of 1×10^{-3} s. A higher time-step may allow us to accurately find the value of the first peaks.

<i>s</i> - value	Frequency/Hz		Percentage difference
	MATLAB	Analytical	
10	10.99	11	0.09%
25	21.74	22	1.18%
50	55.56	55	1.02%

Table 3: Frequency of pressure fluctuation.

1.2 Force analysis

1.2.1 Force acting on bend due to reflection of pressure wave

Using MATLAB, the forces acting on each bend of the loop were calculated. Figure 3 shows the modulus of the magnitude of the force acting on each bend for the given time period. Figure 4 shows a plot of the forces from the first reflection of the pressure wave. We see that the waves are reflected multiple times through the expansion loop, decreasing in magnitude on each reflection. For the purposes of finding the force exerted on the bend due to the reflection of the pressure wave, the peak forces from the first reflection were chosen. Analysing the MATLAB script given, the force components and directions were determined. Figure 5 shows the direction of the force acting on each bend. Table 4 shows the magnitude and direction of the force acting at each bend, and the overall force exerted on the expansion loop. We find that the expansion loop experience a net force pushing it to the left with a slight downwards push.

¹Referenced from ‘Piping Handbook Seventh Edition’ by Mohinder L. Nayyar, page 1266, publisher: McGraw-Hill

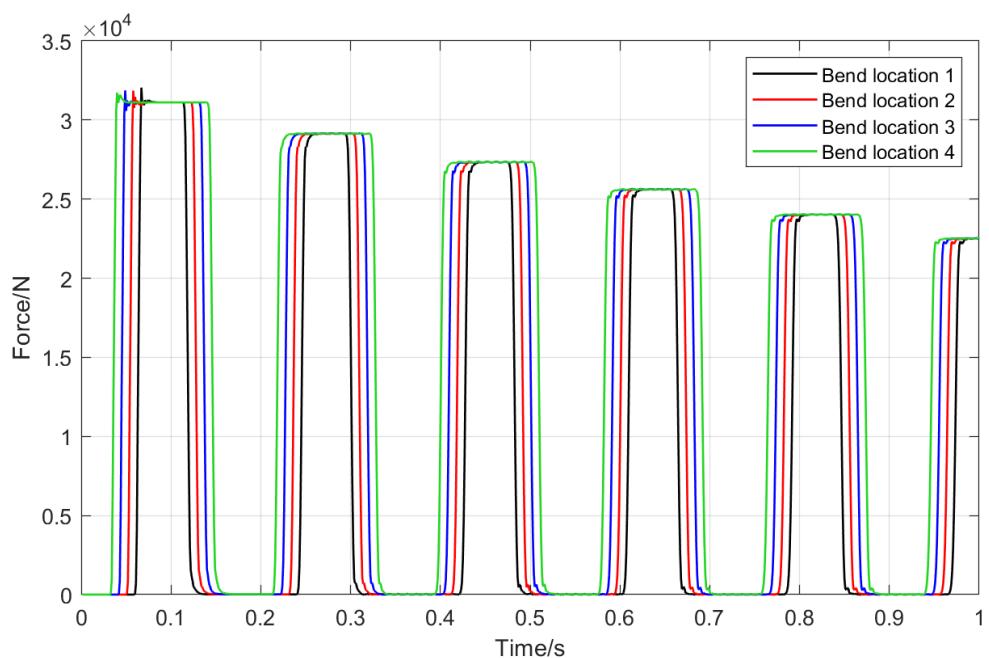


Figure 3: Magnitude of force on each bend from reflections of pressure wave over the time period.

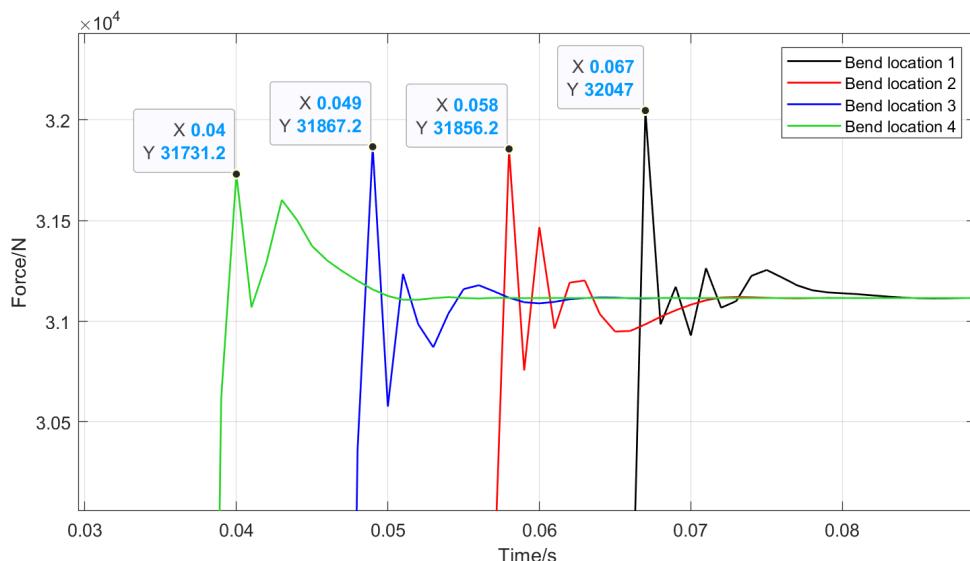


Figure 4: Forces on each bend from first reflection of pressure wave.

Figure 5: Schematic showing forces acting on expansion loop.

Table 4: Table to show magnitude and direction of forces acting on the expansion loop.

1.2.2 Direction of total force on expansion loop

2 Modal analysis of expansion loop

2.1 Modal analysis of a cantilevered pipe

In order to conduct our ANSYS analysis, we need to find the effective pipe density, we can use (3).

$$\rho_{eff} = \rho_s + \rho_{LNG} \left(\frac{d_i^2}{d_e^2 - d_i^2} \right) \quad (3)$$

Table 5 shows the values used in the ANSYS simulations.

Parameter	Value
d_i	793.94 mm
d_e	813 mm
ρ_s	7850 kg m ⁻³
ρ_{LNG}	455 kg m ⁻³
ρ_{eff}	17 214.06 kg m ⁻³

Table 5: Values for cantilevered beam analysis in ANSYS.

2.1.1 List of unique frequencies & mesh sensitivity test

Table 6 shows values of the modal frequencies for a cantilevered pipe with lengths $L = 10, 20, 30$ m. For the theoretical values, we can calculate the bending moment of a cantilevered beam using (4).

$$f_n = \frac{\alpha_n^2}{2\pi} \left(\frac{EI}{AL^4} \right)^{\frac{1}{2}} \quad (4)$$

$$I = \frac{A}{16} (d_e^2 + d_i^2) \quad (5)$$

where coefficient α_n values were found in literature. The theoretical values were calculated using MATLAB (Appendix A.2)

Mode	Frequency/Hz							
	ANSYS		Theory		ANSYS		Theory	
	$L = 4$ m		$L = 10$ m		$L = 20$ m			
1	31.65	33.86	5.35	5.42	1.35	1.35		
2	149.93	212.23	31.49	33.96	8.37	8.49		
3	332.21	594.32	80.99	95.09	22.69	23.77		
4	521.06	1164.66	143.54	186.35	42.93	46.59		
5	712.09	1925.06	214.08	308.01	68.06	77.00		
6	857.22	2875.85	289.12	460.14	97.07	115.03		

Table 6: Table to show values of modal frequencies for a cantilever beam of length $L = 10, 20, 30$ m, compared to theoretical values.

A mesh sensitivity test was conducted using the $L = 4\text{ m}$ case. The element size was varied from 0.5 m to 0.01 m and the modal frequencies were found. By taking the frequency of a particular mode and element size and comparing it to the frequency found in the simulation with element size 0.01 m (for the same mode), a percentage difference was calculated. Figure 6 shows how frequency values converge. We see that the percentage difference in frequency for element size 0.125 m and smaller is negligible. Hence, for the simulations conducted, a mesh size of 0.125 m was used.

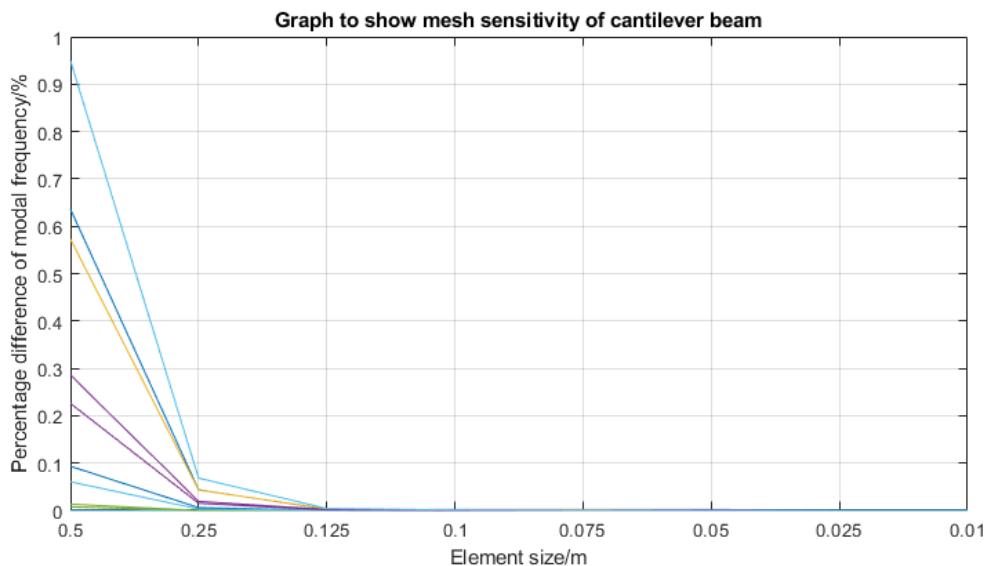


Figure 6: Mesh sensitivity test results. Conducted on a beam with length $L = 4\text{ m}$.

2.1.2 Differences and similarities between theoretical and calculated modes

Using MATLAB, the percentage difference between the ANSYS values and the theoretical values was found. This is shown in Table 7.

Mode	Percentage difference (%)		
	$L = 4\text{ m}$	$L = 10\text{ m}$	$L = 20\text{ m}$
1	7.00	1.27	0.34
2	41.56	7.84	1.43
3	78.90	17.41	4.77
4	123.52	29.82	8.52
5	170.33	43.88	13.14
6	235.49	59.15	18.51

Table 7: Table to show percentage difference between ANSYS modal frequencies and theoretical values.

We find that for lower L -values the modal frequencies are higher in our ANSYS simulation. This trend matches with the theoretical values, where modal frequency decreases as a function of L . For the first mode, we find that ANSYS and theory values are closely matched with single digit or decimal percentage difference. For higher modes, the percentage difference increases substantially. As L increases, our percentage difference between ANSYS and theoretical values decreases. As the mode number increases, the percentage difference increases.

As we are using the Euler-Bernoulli theorem to calculate our theory values, the assumptions made can potentially explain the discrepancy between our modal frequency values. One such assumption is that plane sections remain plane. We know that for higher modes, the beam undergoes a larger displacement, as more sections of the beam are displaced from the normal axis. This large displacement results in plane sections no longer remaining plane, making our assumption invalid for higher modes.

2.2 Modal analysis of an expansion loop

The geometry was constructed within ANSYS as shown in Figure 7. A circular tube profile was used and its geometry is shown in Figure 8. A number of constraints were applied to the geometry and are shown in Figure 9. At the ends of the pipe, we use fixed supports as we are investigating the modal frequencies of this specific section. For the lateral constraints, the justification for such a support is that the piping section should be allowed to slide back and forth within the support. This is because the pipe will thermally expand and contract under normal operation. Hence, this sliding motion will reduce the stress on the piping section, reducing cyclic fatigue. This lateral constraint also allows the pipe some degree of movement in the case of a water hammer, where if it was fixed laterally, may excessively wear (or even) damage the joint. A basic mesh sensitivity test was completed and the results are shown in Figure 10. A mesh size of 0.01 m was chosen as the percentage error is mostly converged and negligible.

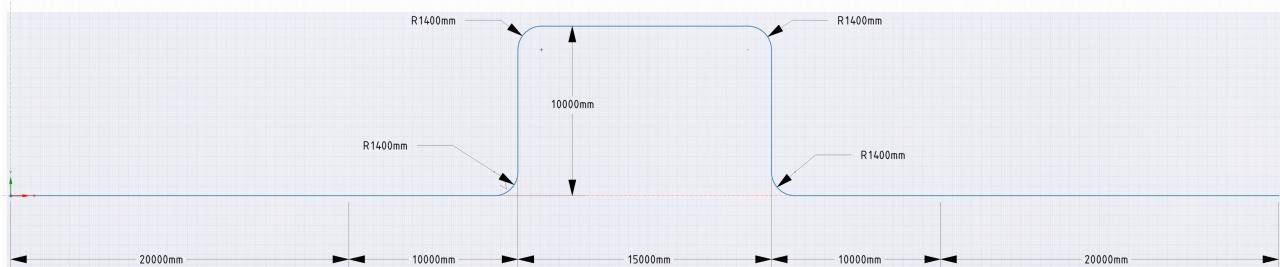


Figure 7: Geometry used for expansion loop modal analysis.

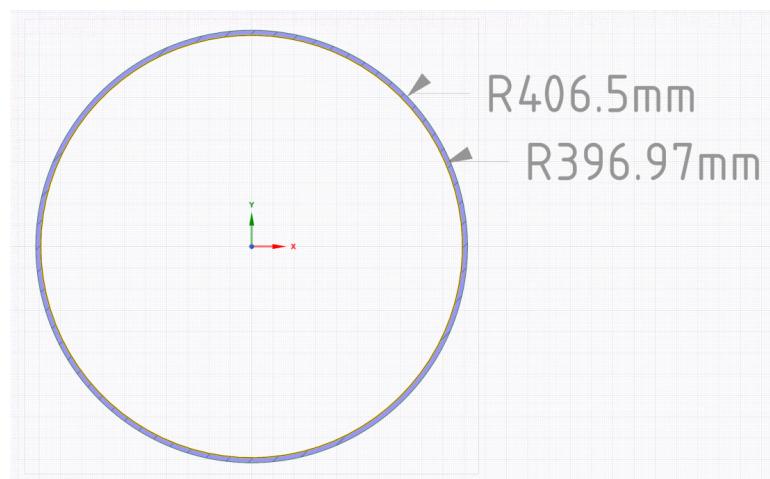


Figure 8: Geometry used for circular tube profile.

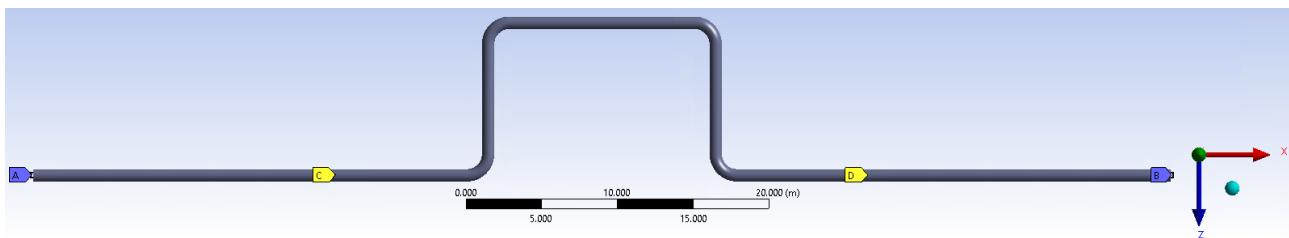


Figure 9: Constraints used for geometry. A & B: fixed constraints. C & D: y- and z- dimensions fixed, x-dimension free.

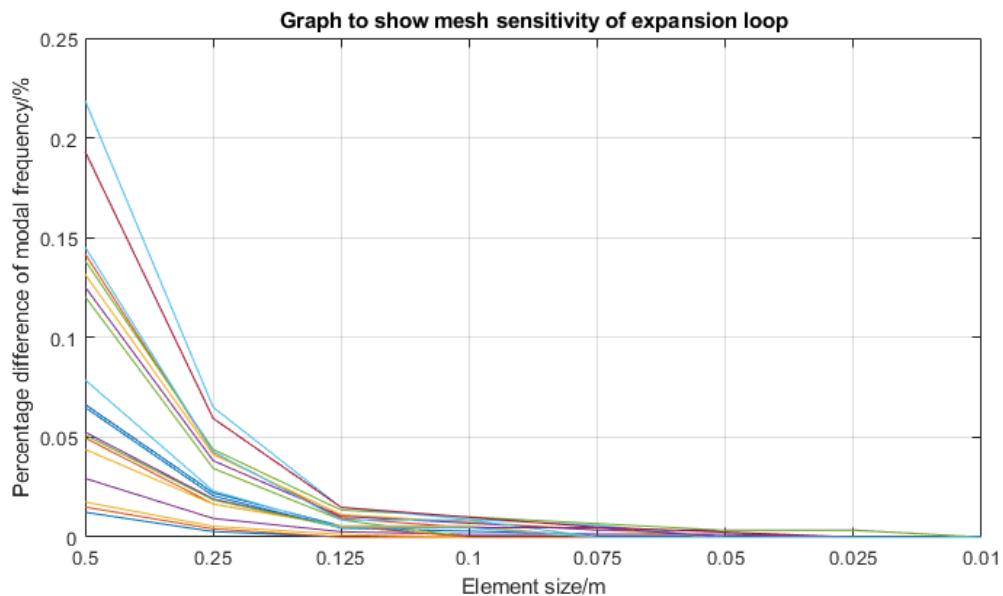


Figure 10: Mesh sensitivity test results for expansion loop.

2.2.1 Results

The results from the simulation are shown in the following figures.

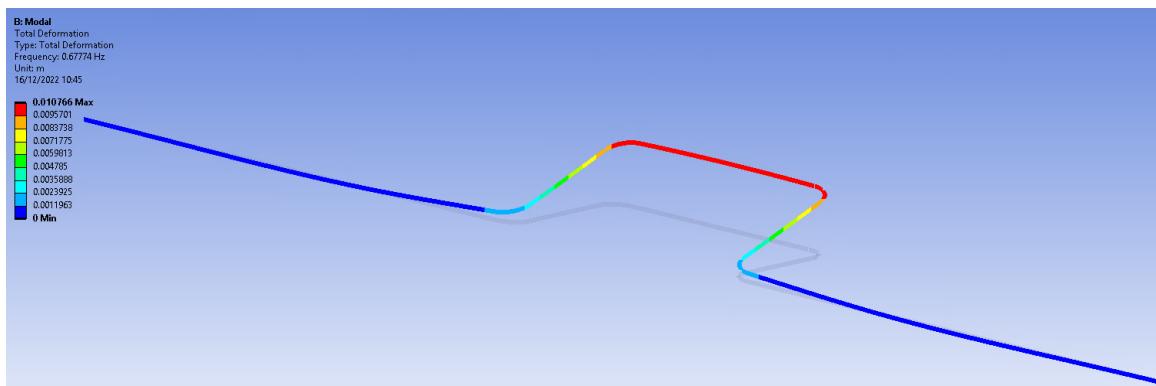


Figure 11: Modal frequency: 0.68 Hz. Out-of-plane deformation.



Figure 12: Modal frequency: 2.11 Hz. In-plane deformation.

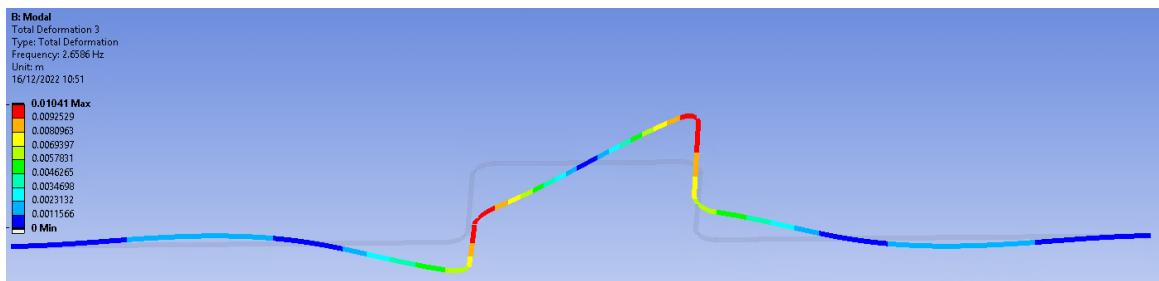


Figure 13: Modal frequency: 2.66 Hz. Out-of-plane deformation.

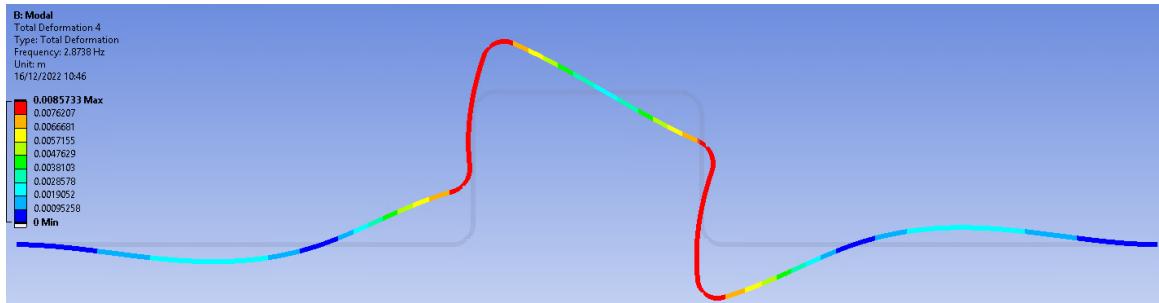


Figure 14: Modal frequency: 2.87 Hz. In-plane deformation.

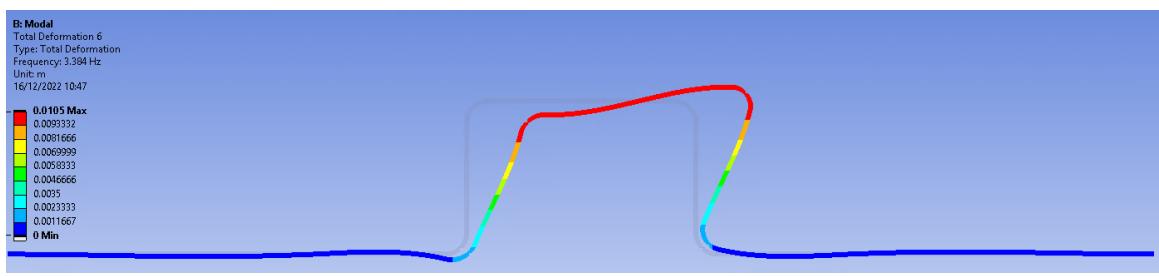


Figure 15: Modal frequency: 3.38 Hz. In-plane deformation.

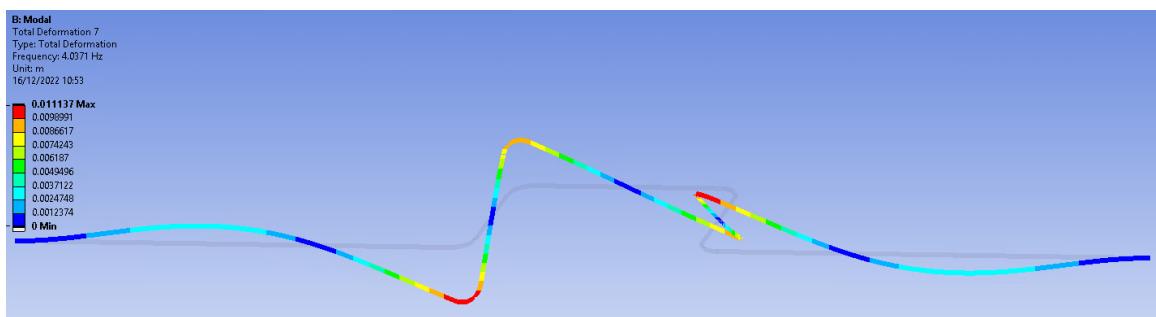


Figure 16: Modal frequency: 4.04 Hz. Out-of-plane deformation.

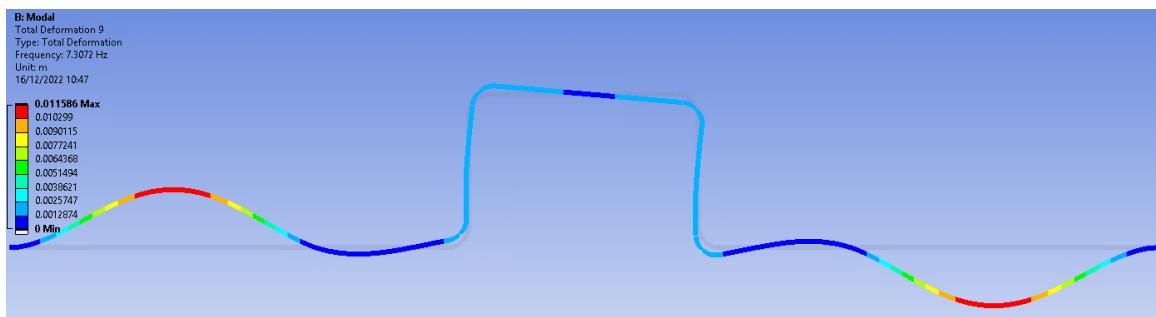


Figure 17: Modal frequency: 7.31 Hz. In-plane deformation.

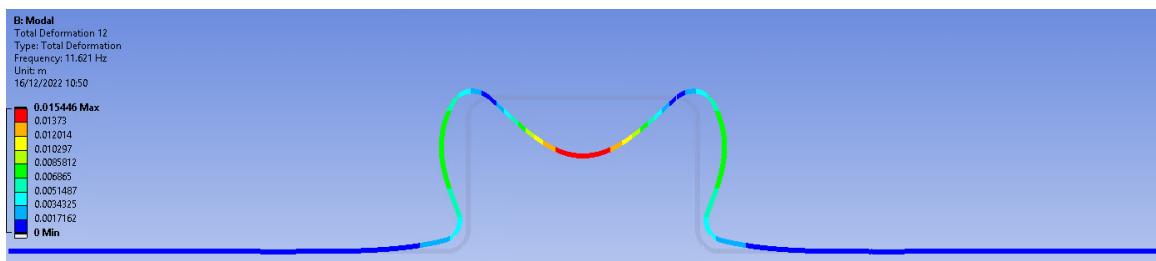


Figure 18: Modal frequency: 11.62 Hz. In-plane deformation.

Figure 11 shows the loop section bending backwards and forwards out-of-plane.

3 Discussion and context

A MATLAB code

A.1 Mesh sensitivity plots

```
%HD

clc
clear
close all

%import data
data = readmatrix('data.xlsx','Sheet','L4','Range','A4:P23');
elSize = [0.5;0.25;0.125;0.1;0.075;0.05;0.025;0.01];

%array initialise
A = zeros(numel(data(:,1)),numel(data(1,:))/2);

%calculate percentage difference of result compared to 0.01m el size
for i = 1:8
    for j = 1:numel(data(:,1))
        A(j,i) = 100*(data(j,i*2) - data(j,end))/data(j,end);
    end
end

%plot
for i = 1:numel(A(:,1))
    plot(A(i,:))
    hold on
end
xticklabels(elSize);
grid on
axis auto
xlabel('Element size/m');
ylabel('Percentage difference of modal frequency/%');
title('Graph to show mesh sensitivity of cantilever beam')
hold off
```

A.2 Theoretical modal frequency values for cantilevered beam

```
%HD

clc
clear
close all

alpha = [1.875;4.694;7.855;10.996;14.137;17.279]; %modal bending coeffs
E = 200e9; %Youngs modulus GPa
L = [4;10;20]; %length of beams
de = 0.813; %external diameter m
di = 0.79394; %internal diameter m
rhoS = 7850; %density steel
rhoLNG = 455; %density gas
```

```
rhoEff = rhoS + rhoLNG*((di^2)/(de^2-di^2)); %density effective
A = (pi*(de/2)^2) - (pi*(di/2)^2); %cross-sectional area m^2
I = (A/16)*(de^2 + di^2); %second moment of inertia m^4

fn = zeros(3,6);
for i = 1:3
    for j= 1:6
        fn(i,j) =
            ((alpha(j)^2)/(2*pi)) * ((E*I)/(rhoEff*A*((L(i))^4)))^(0.5);
    end
end
```
