

CHARACTERISATION OF A RESONANT BENDING FATIGUE TEST SETUP FOR PIPES

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Abstract This paper discusses the resonant bending fatigue test setup designed at laboratory Soete for full-scale fatigue tests on pipes. Following an enumeration of other types of fatigue test setups an attempt is made to characterise the resonant bending machine. The characterisation is obtained by conducting different tests on a steel pipe of grade API X65. Concordance between measured and calculated stresses and influence of excentre position on stress amplitude is discussed. High frequencies and small power input make this test setup very effective. The analytical model correctly predicts the measured stresses and a stress versus excentre curve is obtained. However not yet fully defined, it gives a first indication for the excentre position when preparing for a fatigue test.

Keywords Resonant bending fatigue, stress amplitude, X65 pipe, full-scale

1 INTRODUCTION

Pipes used in offshore applications often fail prematurely under cyclic loads of which the maximum stress value is lower than the upper tensile strength of the material. This phenomenon is called fatigue. Especially at welded or threaded pipe joints, which act as stress raisers, a fatigue crack can initiate. It goes without saying that accurate prediction of the fatigue life of these structures is important. In this paper the characterisation of the resonant bending fatigue test setup build at laboratory Soete is discussed.

As a first part of this paper different setups to perform full-scale fatigue tests are discussed. Axial cyclic loading, 4 point-, rotating and resonant bending. The two last setups give rise to the same stress state in the pipe joint or weld and will therefore be jointly discussed.

The characterisation of the laboratory Soete setup will commence with strain gauge measurements. These will confirm the match between the analytical model discussed in [1] and the reality. The influence of different excentre positions on the stress amplitude will be handled as well. All test are done on a seamless steel tube of 4,8 m length, 324 mm diameter, 12,7 mm thickness and grade API X65.

2 FULL-SCALE FATIGUE TEST SETUPS

2.1 Axial fatigue

The principle of axial fatigue is illustrated in Figure 1. A cyclic axial force is applied which gives rise to cyclic axial stresses according to equation (1). A setup for large diameter pipes can be found at Stress Engineering Services [2]. This machine applies its load according to $R = \sigma_{\min} / \sigma_{\max} = 0$ and its main characteristics are summarized in Table 1. Note that very high forces need to be applied to exert relatively low stresses. This method however causes a uniform stress distribution throughout the thickness of the pipe (1) which can be necessary to accurately predict fatigue life in certain applications (e.g. structural columns).



Figure 1: Axial fatigue

$$\sigma \approx \frac{F}{\pi D_o t} \quad (1)$$

2.2 Four point bending fatigue

The schematics of a 4 point bending test setup are given in Figure 2. This test is widely used to determine fatigue life of risers and columns. The cyclic load is now exerted perpendicular to the pipe.

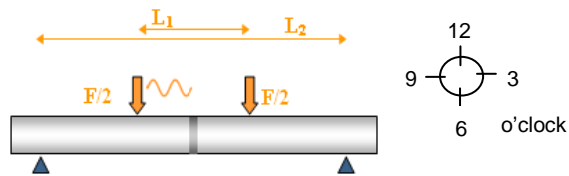


Figure 2: 4 point bending fatigue

Details of two setups have been found in literature. One setup was used to apply loads to relatively small diameter pipes with a load ratio $R = -1$ [3] and the second one for larger diameter pipes but with a load ratio $R \approx 0$ [4]. The characteristics of both setups are again given in Table 1.

In between the load points the bending moment remains constant. The pipe undergoes a cyclic load with a stress amplitude at the outer fibres at 6 and 12 o'clock calculated according to the formulas (2,3,4) in case of $R = -1$. Hence there are some important differences with the axial fatigue setup. The stress is not divided uniformly throughout the thickness and not all points on the circumference see the same load.

$$M = F (L_2 - L_1) / 2 \quad (2)$$

$$\sigma = \frac{MD_o}{2I} \quad (3)$$

$$I = \frac{\pi}{64} (D_o^4 - D_i^4) \quad (4)$$

2.3 Rotating and resonant bending fatigue

2.3.1 Rotating bending fatigue

A constant load is applied perpendicular to the pipe which is rotated to introduce time dependency. The same formulas (2,3,4) can be applied for this setup. Again a constant bending moment is obtained in between the load points. The only difference is that every point at the outer fibre will now be subjected to the same cyclic stresses, shifted in time. Given that the joint is placed at the centre of the setup, it will be subjected to exact the same stress state as in a resonant bending fatigue setup (Figure 3).

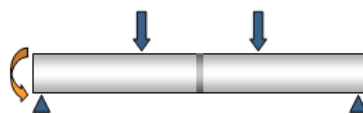


Figure 3: Rotating bending fatigue

Two setups were found in literature. In the first, build and designed by Petrobras and Coppe, an additional axial load can be superimposed to the bending moment. It is suitable for relatively small diameters [5]. The second is used to test larger diameter pipes and was constructed by the University of Oklahoma [6, 7]. An additional axial load can be superimposed here as well. Notice that applying an additional axial force does not have any effect on the oscillating character of the stresses. It results in a non zero mean stress and allows testing with different load ratios. The characteristics are given in Table 1 where reported working values are listed for the second setup whereas actual maximum capacity values are listed in all other cases.

2.3.2 Resonant bending fatigue

As an example of a resonant bending fatigue setup the laboratory Soete machine, which is the subject of this paper, is discussed (Figure 4). Similar setups are used at Stress Engineering Services[8, 9], TWI

[10, 11] and EWI [12]. The pipe is excited according to its first eigenmode by means of a rotating excentric mass. In the absence of axial preload the stress ratio is thus $R = -1$. Again, every point at the outer fibre sees the same cyclic stresses shifted in time. However, an area where the bending moment is constant (as is the case in the rotating bending setup), does not exist anymore. A more detailed description of this setup can be found in [1].

As can be seen in Figure 5, the bending moment is now a fluent line which reaches its maximum at the centre of the tube while the supports are positioned at the nodes of the pipe's first bending mode. The previous remark that the joint at the centre of the tube will be subjected to the same cyclic stresses as in the rotating bending test setup is thus confirmed. Given that the same stress amplitude is constructed. A certain stress amplitude is constructed by altering the position of the excentric mass. The characteristics of our resonant bending fatigue machine are also given in Table 1. The pipe is pressurized with water to lower the eigen frequency and to detect fracture as pressure drops. Due to this pressure, a small axial preload is applied.

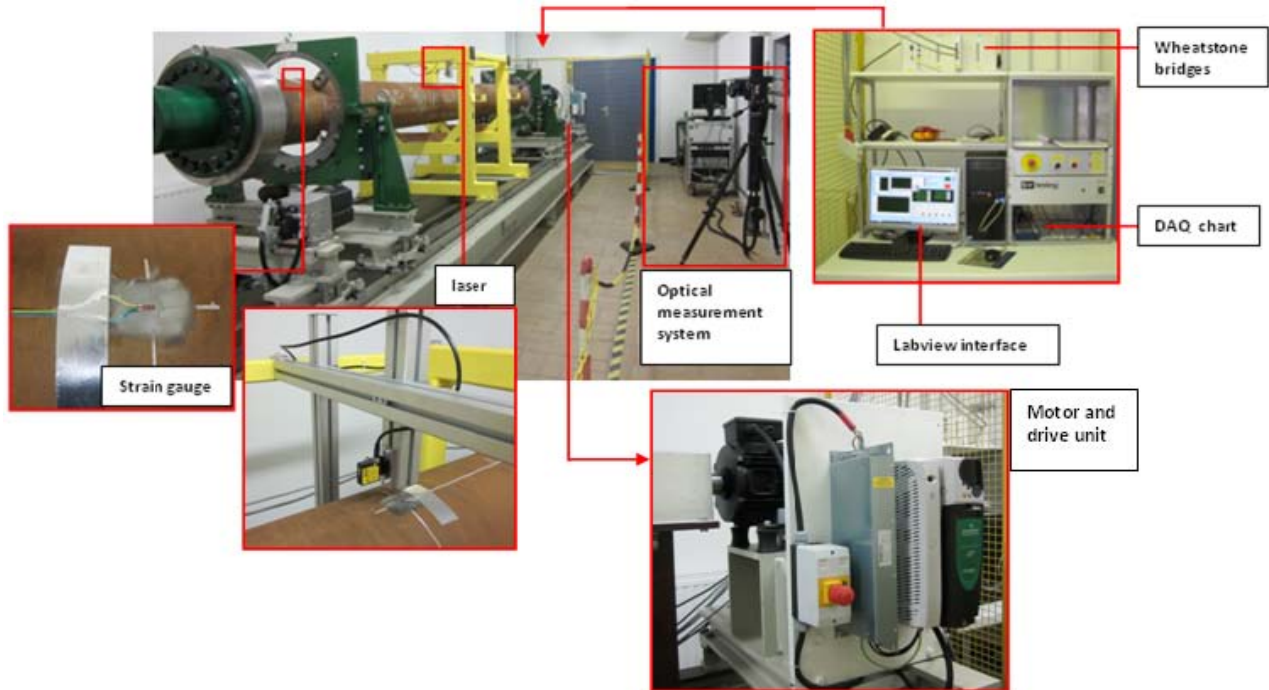


Figure 4: Labo Soete resonant bending fatigue test setup

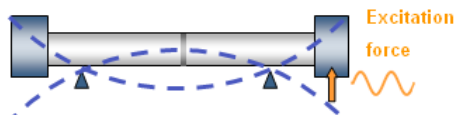


Figure 5: Resonant bending fatigue

Setup	D_o (mm)	L (m)	t (mm)	F (kN)	M (kNm)	σ_{max} (MPa)	f (Hz)
Axial fatigue [2]	351-813	4,5-9,1	11,1-25,4	185	/	15,1	1,0
4 point bending							
$R = -1$ [3]	114,3	1,2	8,6	/	66,3	944,8	0,1
$R \approx 0$ [4]	273,1	4,5	14,3	/	179,0	125,0	0,1
Rotating bending							
[5]	324	6,0	9,5	2000	600,0	1043,6	5,0-15,0
[6,7]	610	9,14	25,4	5780	451,0-995,0	69,0-152,0	2,0
Resonant bending [1]	152,4-508,0	3,7-5,7	40-5,7	/	/	175,0	20-40

Table 1: Characteristics of test setups reported in literature

The advantages of the resonant bending machine when compared to rotating bending setups are the higher possible testing frequencies and the smaller power usage. The resonant bending setup uses between 0,33 kW and 5,9 kW and is driven by a motor with a nominal power of 7,5 kW whilst the rotating bending setup needs a motor of 22,4 kW nominal power for a test with a lower maximum stress and at a reduced testing speed [6, 7].

3 FURTHER CHARACTERISATION OF THE LABORATORY SOETE RESONANT BENDING FATIGUE TEST SETUP

3.1 Stress amplitude

3.1.1 Theoretical versus measured stresses

A semi-analytical theoretical model has been developed to predict the pipe's deflection and stress amplitude at the pipe's mid section, in function of different characteristics of the setup [1]. Predicted stress amplitudes are compared with experimentally determined values derived from strain gauge measurements at the mid-section of the pipe. This is illustrated for a 75° excentre position in Figure 6.

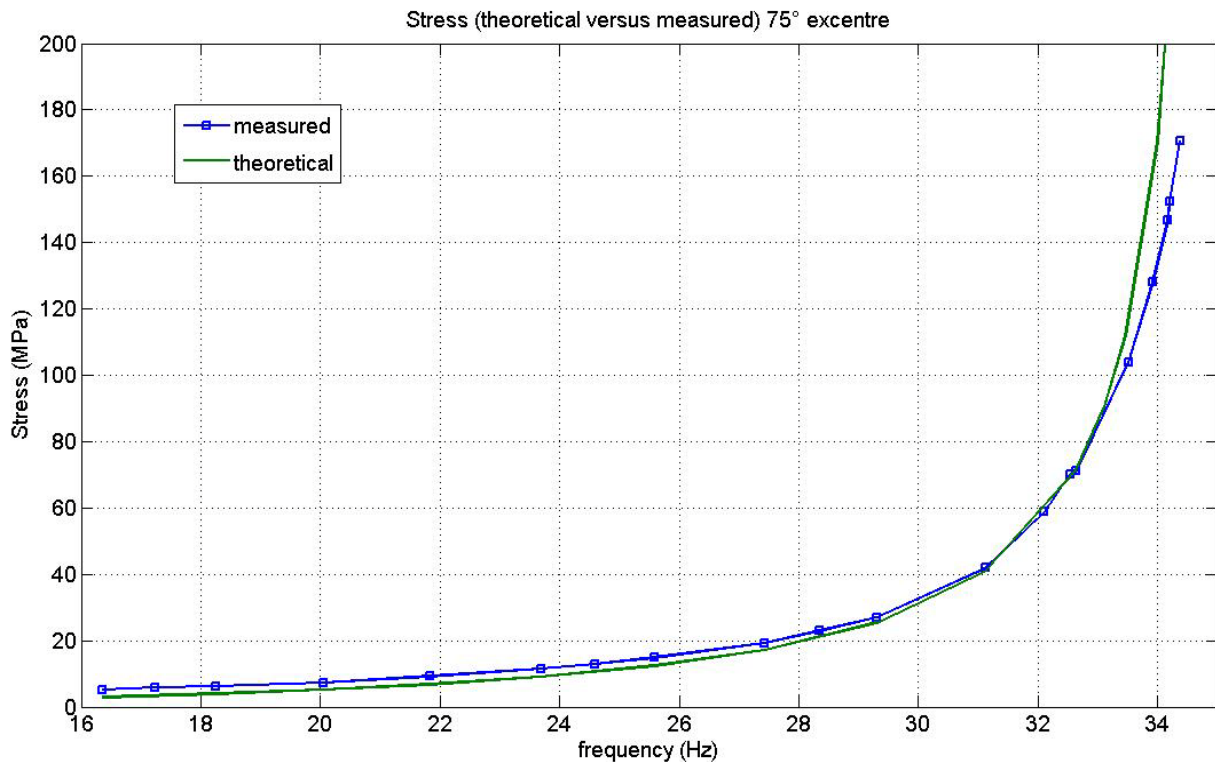


Figure 6: Theoretical versus measured stress amplitude

A good correspondence between the measured and theoretical stress values is observed, except for the higher frequencies. This is explained by the damping effect which is not accounted for by the theoretical model.

3.1.2 Influence of varying excentre position

The influence of altering the excentre position is measured with five strain gauges mounted at symmetrical positions with respect to the mid-section (Figure 7).

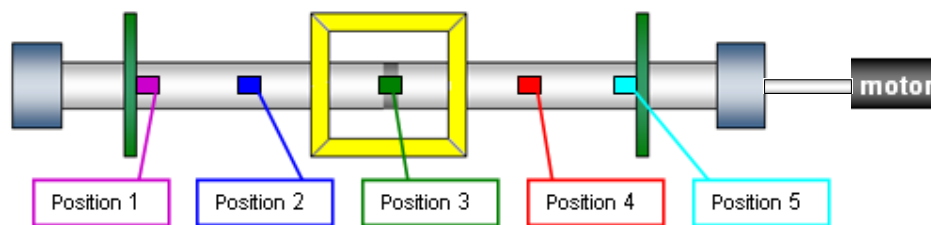
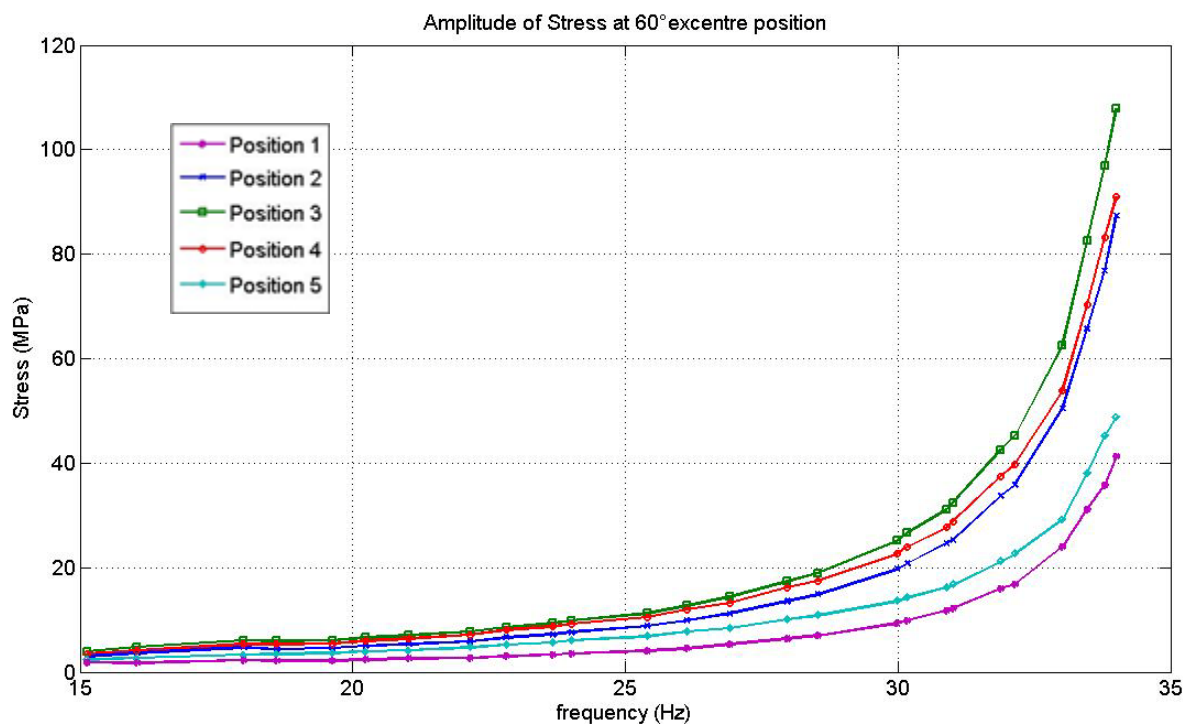


Figure 7: Position strain gauges (top view setup)

In Figure 8 the results of the strain gauge measurements corresponding to one particular excentre position (60°) are illustrated. The strains are recalculated to overall stress by multiplying with the E modulus of the pipe metal.

Figure 8: Stress amplitude at 60° excentre position, calculated from strain gauge measurements

The asymmetry in the setup due to the presence of the motor at the right end is clearly visible. This results in higher stresses at positions 5 and 4 when compared to positions 1 and 2, completely according to expectations. In a future development a comparison between the experimentally asymmetry (Figure 8) and the theoretical asymmetry that is taken into account in the semi-analytical model will be made. Furthermore, equivalent graphs can be built for each excentre position thus evaluating the influence of an excentre position alteration.

The machine excentre design is illustrated in Figure 9. The force exerted on the pipe varies from zero at 0° excentre position to $2F$ at 180° degrees excentre position according to the formula given in (5) where α equals the excentre position. This force is proportional to the stress amplitude in the pipe. Thus a fitting between these curves and the stress data measured at the centre of the pipe for different excentre positions can be used to determine the constants $C(F)$ in (6).

Measurements for 20° up to 125° are illustrated for various frequencies (Figure 10). By taking the logarithmic value of the different constants used to fit the cosinus function given in (6) to the results, an estimation can be made of the constant to be used for plotting the stress amplitude versus excentre position curve at the upper frequency of 34,4 Hz which corresponds to 98% of the pipe's resonance frequency.

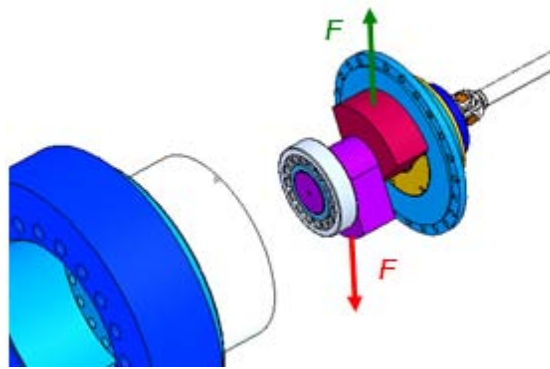


Figure 9: Forces at excentre position 0°

$$Force = 2F \cos\left(\frac{180^\circ - \alpha}{2}\right) \tag{5}$$

$$\sigma = C(F) \cos\left(\frac{180^\circ - \alpha}{2}\right) \tag{6}$$

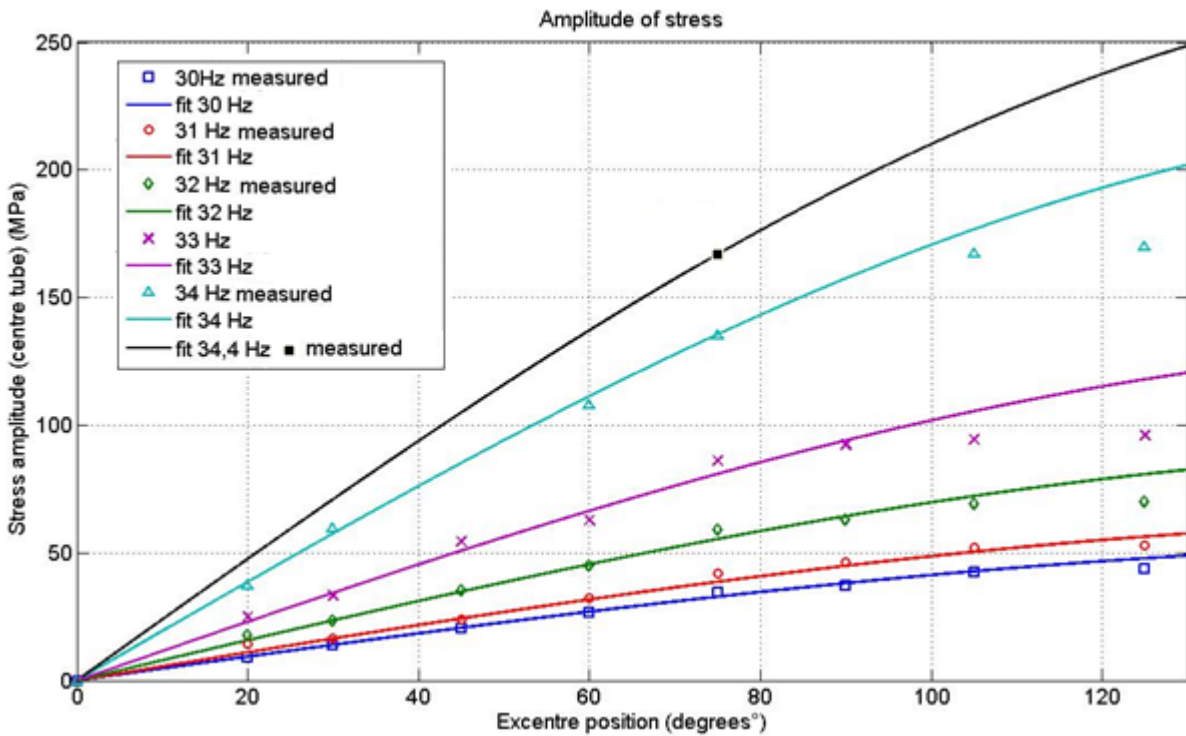


Figure 10: Stress amplitude at the centre of the tube for different excentre positions

When taking back Figure 6 it is seen that for an excentre position of 75° this upper black curve indeed gives a correct approximation of the stress. However further evaluation of this diagram is necessary. For instance the non correspondence between the measured and theoretical values at high excentre positions needs to be examined.

4 CONCLUSIONS

- A comparison of different alternatives to perform full scale pipe fatigue tests has been performed. Different test methods result in different stresses at the centre of the pipe. Compared to rotating bending, resonant bending fatigue tests show the advantages of higher testing frequency and smaller power usage.

- As concluded by J. Van Wittenberghe in [1], the analytical model proves to be accurate. This was evaluated by means of strain gauge measurements. Further evaluation is necessary to evaluate if the analytical model also correctly accounts for the asymmetry in the setup.
- An attempt was made to display the influence of excentre position on stress amplitude at the mid-section of the tube. If the non correspondence between measured and theoretical values for excentre positions larger than 90° will be explained in the future, this graph will prove useful when setting up a fatigue test. It makes it possible to predict in advance which excentre position needs to be installed to exert a certain fatigue stress.

5 NOMENCLATURE

σ	stress	MPa
R	load ratio	
F	force	N
D_o	outer diameter	m
D_i	inner diameter	m
L	length	m
t	wall thickness	m
f	frequency	Hz
M	moment	Nm
I	area moment of inertia	m^4
α	excentre position	degrees

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