Vibration based damage assessment of type 1 compressed natural gas cylinder

Zhang, Yao

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**VIBRATION BASED DAMAGE ASSESSMENT OF
TYPE 1 COMPRESSED NATURAL GAS CYLINDER**

ZHANG YAO

SCHOOL OF CIVIL AND ENVIRONMENTAL ENGINEERING

2015

VIBRATION BASED DAMAGE ASSESSMENT OF TYPE 1 COMPRESSED NATURAL GAS CYLINDER

Zhang Yao

School of Civil and Environmental Engineering

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ABSTRACT

Compressed Natural Gas (CNG) is widely used around the world due to its environmentally friendly nature and significantly less cost as compared to other conventional fuels such as petrol, diesel and Liquefied Petroleum Gas (LPG). A typical 30 L or 40 L CNG cylinder fitted in vehicle usually experiences a larger number of pressurization cycles during its service life than other cylinders used in the industry. When these CNG cylinders are approaching their service lifespan, defects such as corrosion and surface cracks are discovered. If the defects increase to certain sizes, the allowable internal pressure of the damaged CNG cylinder will be lower than the service internal pressure, making them extremely unsafe. In fact, there have been several serious CNG cylinder explosion accidents in the past decades. Therefore, it is crucial to carry out periodic inspections and testing to prevent such catastrophic failure. A few methods have been used for many years but they are still somehow unreliable, time consuming and expensive.

The ultimate aim of this research is to propose and develop a quick, convenient and reliable technique to determine the category, location and severity of damages occurring in Type 1 all metal CNG cylinders. The typical damage is corrosion occurring on the internal surface, whereby reducing wall thickness; it is distinguished into global corrosion covering a large area and local corrosion having strip like shape. Although crack seldom occurs in practice, once it is present, it is dangerous as it may cause catastrophic rupture. Therefore, inspecting and assessing the damage of crack at the early stage has to be carried out as well.

Firstly, the relationship between natural frequencies and wall thickness reduction is derived and analyzed from which the corrosion can be distinguished whether it is global corrosion or local corrosion. For global corrosion, wall thickness reduction covers a large area and the severity can be evaluated accordingly. Because the weight of accelerometer sometimes cannot be ignored, it is considered as an auxiliary mass; the influence of auxiliary mass on the frequencies of CNG cylinders is studied and a method to eliminate such influence is proposed and validated by both numerical examples and experimental tests.

Secondly, a new concept called Frequency Shift Curve (FSC) is proposed, which represents the change of frequency with respect to the position of the auxiliary mass attached on the cylinder. The FSC is related to the square of mode shapes, and therefore, the circumferential FSCs have periodic peaks and troughs; in particular, the fundamental circumferential FSC which can be measured more easily and accurately is sensitive to local corrosion, and the trough with minimum value indicates the circumferential location of local corrosion. Then, the curvature of fundamental longitudinal FSC is utilized for locating the longitudinal location.

Finally, the idea of FSC is extended to Frequency Shift Surface (FSS) and a novel damage index based on FSS and the similarity of longitudinal FSCs is proposed. For both a single and multiple local corruptions, not only the locations can be determined, the sizes and severities can also be assessed by this damage index. In addition, it does not require prior information from undamaged CNG cylinders which is difficult to obtain in practice. The damage due to crack in a 7 L CNG cylinder is also investigated in details, and it shows good performance in both numerical examples and experimental results.

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LIST OF SYMBOLS

Symbols

For Chapter 2

K_c	Spring stiffness
K_d	Defect stiffness
x_i	Coordinate of point i
Δl	Distance between two adjacent testing points
φ	Mode shape
κ	Mode shape curvature
ω_i	The i th natural frequency
$\boldsymbol{\varphi}_i$	The i th mode shape
$\mathbf{F}$	Flexibility matrix
Δ	Difference matrix
EI	Bending stiffness
U_{ij}	The j th strain energy at the i th element
EI^d	Bending stiffness of the damaged structure
φ_j^d	The j th mode shape of damaged structure
β_j	Damage index of strain energy based method
R	Radius of cylindrical shell
h	Thickness of cylindrical shell
l	Length of cylindrical shell
ν	Poisson ratio
f	One dimensional signal

$*$	Complex conjugate of wavelet function
Wf	Wavelet coefficients
f_k	The k th frequency of undamaged structure
β_j	Stiffness reduction factor
$[M]$	Global mass matrix
$SE_{j,k}$	Maximum modal strain energy of the j th element
$\frac{\Delta f_k}{f_k}$	Fractional change in frequency
d	Damage distribution function
E_d	Elastic modulus at damage area

For Chapter 3

R	Radius of circular ring
h	Thickness of circular ring
ρ	Material density
E	Elastic modulus
u_R	Displacement of radius direction
u_θ	Displacement of circumferential direction
ω_{n1}	Frequencies related to radial modes
ω_{n2}	Frequencies related to circumferential modes
h_0	Initial wall thickness

For Chapter 4

M	Mass of mass lump
k	Spring stiffness

$\bar{m}$	Mass of beam in unit length
q	Vertical displacements of the mass
u	Vertical displacements of the beam
δ	Dirac delta function
f	Contact force
φ_n	The n th mode shape of the beam
q_n	The n th modal coordinate
ω_{bn}	The n th natural frequency of the beam
ω_n	The n th natural frequency of the beam-mass interactive system
D	Bending stiffness of plate
$\tilde{m}$	Mass of plate in unit area
φ_{nm}	The (n, m) th mode shape of the plate
q_{nm}	The (n, m) th modal coordinate
ω_{nm}	The (n, m) th natural frequency of the plate
Ω_{nm}	The (n, m) th natural frequency of the plate-mass interactive system
El_r	Bending stiffness of circular ring
ρ_r	Density of circular ring
A	Cross-sectional area of circular ring
a	Radius of circular ring
M_r	Mass of auxiliary mass
k_r	Stiffness of auxiliary mass
$U_{\pm n}$	The $\pm n$ th mode shape of circular ring
$q_{\pm n}$	The $\pm n$ th modal coordinate
$\omega_{\pm rn}$	The $\pm n$ th frequency of circular ring

$\omega_{\pm n}$ The $\pm n$ th frequency of ring-mass interactive system

$\phi_{\pm nq}$ The $\pm n$ th mode shape of cylindrical shell

For Chapter 5

$\omega_{\pm drn}$ The $\pm n$ th frequency of damaged ring

$\omega_{\pm dn}$ The $\pm n$ th frequency of damaged ring-mass interactive system

$U_{\pm n}$ The $\pm n$ th mode shape of damaged ring

$\Delta\omega_{\pm drn}$ The perturbations of $\pm n$ th natural frequencies of intact ring

$\Delta U_{\pm n}$ The perturbations of $\pm n$ th mode shapes of intact ring

t_i The lowest trough value

t_j The other three trough values

Δ Damage index based on circumferential FSC

For Chapter 6

$\omega_{\pm c,nq}$ The $\pm(n,q)$ th frequency of cylindrical shell

$\omega_{\pm nq}$ The $\pm(n,q)$ th frequency of cylindrical shell-mass interactive system

ω The +2nd frequency of the cylinder

D The damage index based on FSS

Abbreviations

ANSI American National Standards Institute

CNG Compressed Natural Gas

COMEAC CO-ordinate Modal Energy Assurance Criterion

EMI Electromechanical Impedance

FEM Finite Element Method

FFT Fast Fourier Transform

FRF Frequency Response Function

FSC Frequency Shift Curve

FSS Frequency Shift Surface

GFM Global Fitting Method

GSM Gapped Smoothing Method

ISO International Organization for Standardization

LPG Liquefied Petroleum Gas

MAC Modal Assurance Criterion

NDT Non-Destructive Testing

NGV Natural Gas Vehicle

PZT Lead Zirconate Titanate

CHAPTER 1. INTRODUCTION

1.1 Research background

For the past few years, Compressed Natural Gas (CNG) has been increasingly used around the world because it is environmentally friendly and costs less as compared to other conventional fuels such as petrol, diesel and Liquefied Petroleum Gas (LPG). According to Natural Gas Vehicle Global survey, the number of natural gas vehicles had grown to 16.7 million by the end of 2012 (Natural Gas Vehicle Global, 2013). This number is expected to reach 19.9 million vehicles with a compound annual growth rate of 7.9%. In Singapore, the government encourages the public to use CNG as an alternative to conventional fuel. Fig. 1.1(a) shows common vehicles which are being converted to CNG vehicles, while Fig. 1.1(b) shows a typical Type 1 CNG cylinder mounted in the boot of a private vehicle.

(a)

(b)

Fig. 1.1 (a) Converted CNG vehicles and (b) A typical Type 1 all metal CNG cylinder

Natural gas has to be compressed at a very high pressure (200 to 300 bar) and stored in cylinders. Usually, a typical 30 L or 40 L CNG cylinder fitted in a vehicle suffers a large number of pressurization cycles (up to 1000 times) during its service life (up to 20 years) than other cylinders used in the industry. When these cylinders are approaching their service lifespan, sometimes defects such as corrosion and surface crack can be found. If these defects grow to certain sizes, the allowable internal pressure of the damaged CNG cylinder will be lower than the service internal pressure, making the CNG cylinders extremely unsafe. Moreover, as a cylinder is subjected to high pressure, the situation is highly risky when it leaks or ruptures. Several CNG cylinder explosion accidents did occur and were reported in recent years. For example, a serious accident happened in Brazil in 2001 shown in Fig. 1.2(a); while Fig. 1.2(b) shows an accident of CNG bus explosion that happened in Seoul, South Korea in 2010; and Fig. 1.2(c) shows the damaged vehicle at the gas filling station at Rawalpindi, Pakistan in 2014, where one person was killed and two others were injured.

Therefore, it is vital to prevent such accidents from occurring through regular periodic inspection of in-service CNG cylinders. Although a few techniques have been developed and recommended in some standards (ANSI/CSA NGV2-2002, ISO 11439, ISO/FDIS 19078), they still have some drawbacks such as high cost and long inspection time. As a

result, a low cost and effective approach for regular inspection of in-service CNG cylinders is required in the industry.

(a) A serious accident happened in Brazil, 2001 (Suparek et al., 2008)

(b) Korean CNG bus explosion in Seoul, South Korea, 2010

(c) CNG cylinder exploded in Rawalpindi, Pakistan, 2014

Fig. 1.2 Some serious CNG cylinder explosion accidents

1.1.1 Category of CNG cylinders

CNG cylinders are designed for storage of high pressure natural gas so that they have to be safe, light and cheap.

According to ISO 11439 (2000), “International standard: Gas cylinders – High pressure cylinders for the on-board storage of natural gas as a fuel for automotive vehicles”, there are four basic types of CNG cylinders:

Type 1 – All metal cylinders (steel or aluminium)

This type of CNG cylinders are completely made of metal (steel or aluminium) so they are the cheapest; on the other hand, most of this type of cylinders are seamless, which makes them more reliable. Therefore, Type 1 CNG cylinders are the most widely used in many countries around the world at the present time. Most Type 1 CNG cylinders are made of steel and the amount of Type 1 CNG cylinders made of aluminium is much less, hence, in this research only the cylinders made of steel is considered and the Type 1 CNG cylinders in this thesis indicate the seamless steel CNG cylinders.

Type 2 – Cylinders with metal liner and hoop wrapped composites (carbon or glass fibres)

The number of Type 2 CNG cylinders used is less than that of Type 1 CNG cylinders, and they are only slightly more expensive and heavier. The main body of the tank consisting the metal cylindrical shell needs to be wrapped by composite fibres to enhance the capacity of retaining pressure, which is based on the fact “that a sphere will contain twice as high a pressure as a long cylinder of the same diameter and wall thickness” (Chamberlain, 2004).

Type 3 – Cylinders with metal liner and fully wrapped composites (carbon or glass fibres)

This type of CNG cylinders was developed later than the first two types. Different from Type 2 CNG cylinders which are wrapped only at the part of cylindrical shell, Type 3 CNG cylinders are fully wrapped by composites, including the two hemispherical ends; therefore they are lighter but more expensive.

Type 4 – Cylinders with plastic liner and fully wrapped composites

This type of CNG cylinder is all composite CNG cylinder and they were introduced in 1993.

Because they have the lightest weight, they are getting more popular although the cost is still high. At present the amount of this type of CNG cylinders is less than the other three used in general.

1.1.2 Manufacture of Type 1 CNG cylinder

Generally, there are three common manufacturing processes for seamless Type 1 CNG cylinders (Chamberlain, 2004): Billet Piercing Process, Deep Drawing Process and Hot Spinning of Seamless Tubes Process; and they have similar steps which are illustrated below.

1.1.2.1 Billet Piercing Process

The procedures of Billet Piercing Process are shown in Fig. 1.3. The steel billet is pre-heated up to 1200°C and formed into cylinder through piercing and hot extrusion. Subsequently, its wall thickness is reduced through drawing process and its neck is formed by neck machining after shot blasting, shell inspection and hot spinning. Then, it is quenched and tempered to satisfy the requirement of hardness, strength and toughness, after which its neck is threaded to install the safety valve. Finally, it is inspected through ultrasonic testing and hydrostatic testing.

The main advantages of Billet Piercing Process are higher safety and productivity, while the drawbacks of this method include higher weight-capacity ratio, more surface manufacturing defects.

Fig. 1.3 Procedures of Billet Piercing Process

1.1.2.2 Deep Drawing Process

The procedures of Deep Drawing Process are shown in Fig. 1.4. Different from Billet Piercing Process, the raw materials are steel plates of uniform thickness which are easily available in the industry. After surface treating and lubricating, they are punched into cup-like shapes and deeply drawn into cylinder shapes. Similar to Billet Piercing Process, the step of hot spinning is applied to form the neck for installation of valve and then the CNG cylinders need to be inspected through ultrasonic testing and hydrostatic testing.

The Deep Drawing Process can produce CNG cylinders with uniform wall thickness, smooth appearance and lower weight-capacity ratio; however, it has “high risk of producing wrinkles and ruffles on the shell side of the cylinder, lower productivity and higher cost” (Chamberlain, 2004).

Fig. 1.4 Procedures of Deep Drawing Process

1.1.2.3 Hot Spinning of Seamless Tubes Process

The procedures of Hot Spinning of Seamless Tubes Process are shown in Fig 1.5. For simplicity, the seamless tubes are directly used so that the first step is to cut the tubes. After the formation of necks and ends, the final steps are almost the same as those in the previous two processes.

Fig. 1.5 Procedures of Hot Spinning of Seamless Tubes Process

The Hot Spinning of Seamless Tubes Process manufactures seamless Type 1 CNG cylinders with low weight capacity ratio, uniform thickness and it is highly productive and efficient; however, it costs more and has limitation because of the usage of seamless tubes, moreover, it has high “risk of lack of proper fusion at the middle of bottom” (Chamberlain, 2004).

1.1.3 Damage types of Type 1 CNG cylinder

With reference to ISO 19078 (2006), the typical damages of Type 1 CNG cylinders are listed as follows: marking and labelling, cuts or gouges, abrasion, charring or soot, gas leakage, chemical attack, bulge, dents, cracks and corrosion. For each type of damage, there are different levels of severity based on which the acceptance and rejection criteria are determined.

The marking and labelling damage means the loss of essential information of the CNG cylinders like manufacture date and expiry date, etc. If so, the cylinders need to be eliminated.

The damages of cuts and gouges are usually found on the external surface, and therefore, they can be identified through regular visual inspection. Usually the rejection criteria of this type of damage are provided by the manufacturers, if not, the defects with depth higher than 0.25 mm need further inspection or the cylinder should be condemned.

Abrasion is similar to cuts and gouges and it usually exists on the external surface as well, therefore, it can be easily detected through visual inspection. The surfaces abraded by heavy loading appear like a group of cuts while those under light loading are smoother. Generally, the CNG cylinders with abrasion deeper than 1.25 mm need to be rejected while those with abrasion less than 0.5 mm can be accepted.

Occasionally, CNG cylinders come under fire attack and then damages of charring and soot occur. The cylinder needs an immediate inspection and it will be condemned if the discoloration is permanent.

Gas leakage is very dangerous so that the cylinder needs to be condemned immediately once it has gas leakage.

Chemical damage can dissolve the metal and the wall thickness of CNG cylinders may decrease so that the safety of the cylinders cannot be guaranteed. Usually, the CNG cylinders suffering chemical attack can be accepted only if the chemical can be cleaned and removed completely.

Bulge is visible and it is very severe damage, therefore, the CNG cylinder which has bulge needs to be rejected.

Dents are depressions which usually occur after impacts. Small shallow dents are more dangerous because they can bring stress concentration and initiate cracks. The CNG cylinders need to be rejected when either the depth of dent is more than 1.6 mm or the diameter is less than 50mm.

Crack damage is not included in ISO 19078, however, when a crack is discovered, it should be investigated carefully. Corrosion, cycle re-pressurization and dents can initiate cracks and then, the CNG cylinders may suffer fatigue failure. However, cracks are seldom discovered in Type 1 all metal CNG cylinders because the cylinders are seamless and cracks usually initiate from seams at girth welds.

(a)

(b)

Fig. 1.6 Internal corrosion of a 5-year old 7 L CNG cylinder (a) upper half surface (b) lower half surface

Most damages occur on the external surface, and therefore they can be identified by regular visual inspection. However, corrosion is always found on the internal surface (Fig. 1.6); therefore, it is difficult to detect the corrosion by regular visual inspection. Moreover, it will reduce the wall thickness due to chemical reaction, decreasing the carrying capacity of the CNG cylinders subjected to high pressure. As a result, it is necessary to investigate and develop a corrosion detection technique in the industry.

Generally, there are three types of corrosion: pitting corrosion, line corrosion and general corrosion.

Pitting corrosion creates small isolated holes as shown in Fig. 1.7. The tolerance of pitting corrosion is high because it reduces the wall thickness locally. Generally, the isolated pits with depth less than a quarter of wall thickness can be acceptable; otherwise the manufactures' rejection criteria should be applied.

Fig. 1.7 Typical pitting corrosion (Pattaya, 2007)

Line corrosion shown in Fig. 1.8 consists of several pits located in a narrow band; usually the distance between pits is more than the typical width of pits. It can be found that the line corrosion is a series of isolated pitting corrosion; therefore, it is more harmful than isolated pitting corrosion. Generally, CNG cylinders containing line corrosion with length more than 100 mm or depth more than 10% of wall thickness should be condemned.

Fig. 1.8 Typical line corrosion (Pattaya, 2007)

General corrosion contains a lot of pits covering large area of internal surface which can be seen in Fig. 1.9. It reduces wall thickness over a large area, and therefore, it is more serious than the other two types of corrosion. If the corrosion is no deeper than 10% of wall thickness or smaller than 25% area of internal surface, it can be acceptable otherwise it requires further inspection. If the corrosion area is more than 25% of internal surface, it should be rejected no matter how deep the corrosion is.

Fig. 1.9 Typical general corrosion (Pattaya, 2007)

1.1.4 Inspection and requalification of CNG cylinder

CNG cylinders need to be inspected periodically so that serious accidents and severe damage or injury can be prevented effectively. ISO 19078 (2006) and ANSI/NGV2 standard (2000) advise that in-service CNG cylinders should be inspected regularly every 3 years or less. However, CNG cylinders need to be inspected more frequently if they are improperly designed, more frequently used, exposed to extreme weather and filled by unsuitable natural gas. In fact, when CNG cylinders experience accidents, they should be inspected immediately.

Generally, hydrostatic testing, ultrasonic testing and acoustic emission are recommended by the manufactures for regular inspection of in-service CNG cylinders; while hydrostatic testing and ultrasonic testing are required in manufacturing by ISO 11439 (2000). All these techniques will be discussed in details in the next chapter.

1.2 Objective and scope of the research work

This research aims to investigate the influence of corrosion and crack on the vibration characteristics of CNG cylinders (cylindrical shells) and subsequently develop a quick and effective technique to assess the internal corrosion of CNG cylinders.

The objectives of this research are as follows:

To investigate the influence of reduction of wall thickness on the natural frequencies of CNG cylinders. In fact, corrosion will reduce the wall thickness of CNG cylinders, causing the decrease of the natural frequencies of CNG cylinders. Different types of corrosion, for example, line corrosion and general corrosion have different influence on the natural frequencies of CNG cylinders.

To study the influence of auxiliary mass on the natural frequencies of CNG cylinders. The weight of the accelerometer which is usually used in the vibration testing cannot be ignored, and it will affect the natural frequencies of CNG cylinders. Therefore, the influence of the accelerometer considered as auxiliary mass will be studied further.

To propose and develop a technique to detect single local corrosion in the circumferential and longitudinal directions respectively and furthermore to detect multiple local corrosions. The natural frequencies of CNG cylinders change with respect to the location of auxiliary mass. In particular, when the auxiliary mass is approaching the local corrosion where the local stiffness decreases, the natural frequency will drop significantly because the auxiliary mass increases the local inertia.

To detect the damage due to crack using the proposed method. Erosion and external impact can generate crack in a seamless cylinder, hence, the damage due to crack is sometimes found in Type 1 CNG cylinders. Therefore, it is investigated in this research. It is found that the proposed approach is sensitive and effective to detect crack. This is verified by both numerical examples and experimental tests.

1.3 Layout of the research work

This chapter includes the background and objectives of this research. Chapter 2 reviews the existing non-destructive testing techniques of cylindrical shells and previous analytical and experimental studies. Chapter 3 presents the relationship between reduction of wall thickness or different corrosion and natural frequencies of CNG cylinders. Chapter 4 illustrates the influence of accelerometer considered as auxiliary mass on the natural frequencies of CNG cylinders, based on which an approach of detecting local damage in the circumferential and longitudinal directions is discussed in Chapter 5 and Chapter 6, respectively. In Chapter 6, the scenario of multiple damages is considered as well and in Chapter 7 the scenario of crack damage is investigated. Finally, conclusions and future works are discussed in Chapter 8.

CHAPTER 2. LITERATURE REVIEW

2.1 Introduction

Non-Destructive Testing (NDT) has received a lot of attention in the past decades, and several approaches have been proposed in recent years to identify early damages in civil engineering structures.

Generally, there are four levels of damage identification; an NDT method obtaining higher level of damage identification is better. The four levels are as follows:

Level 1: determination of presence of damage;

Level 2: determination of location of damage;

Level 3: determination of severity of damage;

Level 4: prediction and evaluation of remaining life of damaged structures.

Although there have been advances made in several NDT techniques in recent years, many of them still have limitations and are not suitable to be used for damage identification of CNG cylinders. In this chapter, the frequently used NDT methods and techniques will be reviewed first, and those which could be used for CNG cylinders will be specially studied in details.

2.2 Frequently used NDT methods

In practice, the most frequently used NDT methods are visual inspection, ultrasonic testing, X-ray testing, magnetic particle testing, eddy current method, thermal testing and vibration based method.

2.2.1 Visual inspection

Visual inspection is the earliest NDT method and it is especially used for visible damages like surface cracks and abrasion. It has the advantages of low cost and easy operation;

however, it can only be operated by trained and qualified inspector. Furthermore, only visible surface damages can be identified effectively, it may fail when the damage is hidden from the surface of the structure.

2.2.2 Ultrasonic testing

Ultrasonic testing is frequently used to inspect defects by high frequency ultrasonic wave which is generated by transducer and propagates through the materials. Fig. 2.1 shows a typical ultrasonic testing system. When the ultrasonic wave meets the lower bound, it will be reflected back and the reflected signal will be received by the transducer. The initial wave and the reflected wave can be displayed on the screen and the time from the generation of the initial wave to the arrival of reflected wave can be read. If there is a flaw across the wave path, the initial wave will also be reflected back and the reflected wave caused by the flaw will be displayed on the screen. This is the basic principle of dimensional measurements and damage identification using ultrasonic testing.

Fig. 2.1 Typical ultrasonic system

At present, a large number of instruments based on ultrasonic wave have been designed and manufactured, and advanced machine vision techniques have been developed to improve their capabilities. Some typical techniques include Guided Waves, Time of Flight Diffraction Technique, Phased Array and Visualization of Ultrasonic Waves, etc. In particular, visualization of ultrasonic waves is a novel technique to visualize ultrasonic waves propagated in a general solid medium. In fact, “ultrasonic waves generated by a

fixed transducer are received by a laser probe at grid points in a give scan area and thereafter a contour map for the amplitude of each waveform can be presented a snapshot of travelling waves” (Yashiro et al., 2008)

The main advantages of ultrasonic testing include a wide range of usage like metal and composites, high sensitivity to surface and subsurface flaws, high accuracy in measuring flaw position and estimating flaw size, harmless to human body. However, like all other non-destructive testing methods, ultrasonic testing has its limitations as well: high demand of surface for easily transmitting ultrasound, requirement of training and qualification, difficulty to explain signal, requirement of coupling medium, failure to detect defects parallel to the ultrasonic wave. In practice, ultrasonic testing is a powerful tool and it can be used for damage identification of CNG cylinders. The procedures will be reviewed in Section 2.3.3.

2.2.3 X-ray testing

X-ray testing is the most frequently used radiographic testing. Although there is other radiographic testing method like Gamma-ray testing method, radiographic testing usually indicates the X-ray testing. X-ray can pass through object and then a shadow image can be captured on film located on the opposite side of the X-ray source. The internal flaws can be seen directly on this image.

X-ray was first discovered by Wilhelm Conrad Roentgen in 1895 and then it became popular in the realm of medicine. It has been approved by the American Society of Mechanical Engineers until 1931 to check the integrity of welded pressure vessels. Nowadays, the quality of the image becomes higher and higher but the used procedures and processes are still similar to those in the early days. In addition to damage inspection, X-ray is usually used in the security systems at airports and subway stations. Films are likely to be eliminated and inspectors can capture digitized images and inspect defects through screens. Advanced image processing techniques will help to identify flaws and computers are likely to replace human beings in X-ray testing in the future in order to avoid the adverse effect to human bodies caused by radiation.

Usually the image of X-ray is intuitive and easy to be understood; however, it is harmful to human body, and it is only effective for metallic material, which limits its applications.

2.2.4 Magnetic particle testing

Magnetic particle testing uses magnetic particles and artificially created magnetic fields to detect surface defects, especially surface and subsurface cracks. When some defects exist on the surface of testing components, the magnetic field lines will be curved and spread out if they are passing through the defects. Once iron particles are sprinkled on the testing components, they will be attracted to the defects where the magnetic flux leaks. Therefore, the defects can be visually inspected through the iron particle cluster.

Magnetic particle testing has advantages of high sensitivity, easy operation, low cost and direct visual inspection; however, it is suitable for ferromagnetic materials and effective for surface and subsurface flaws only.

2.2.5 Eddy current testing

Eddy current testing which is based on the principle of electromagnetism is popular and it has a lot of applications. In fact, the eddy currents are induced on the surface of a conductor such as the copper wire when the alternating current is approaching the surface. The induced eddy currents will be disrupted if defects exist on the surface and this change identifies the defect and its location.

Eddy current testing is a typical non-contact testing and it does not require coupling medium; moreover, it has high sensitivity and visible results. However, it is only effective for surface and subsurface defects of conductive materials, which limits its applications.

2.2.6 Thermal testing

Thermal testing based on heat conduction and thermal radiation is relatively new compared to other NDT approaches. Infra-red camera can record surface temperature distribution and heat flows disrupted by defects; therefore, defects can be visually identified from thermal image. At present, advanced image processing techniques and pattern recognition methods can help to detect damages automatically.

Thermal testing is also a typical non-contact testing. It is highly sensitive, easy to conduct and suitable for either small or large specimen. But the equipment is usually expensive, and it is more effective for surface or subsurface defects.

2.2.7 Vibration based NDT methods

The above NDT methods are only effective when the approximate damage area is already known; otherwise, the entire specimen needs to be scanned which is inconvenient and time-consuming. However, vibration based NDT method is a promising alternative because vibration signals can be measured easily and they contain both global and local information (Salawu, 1997; Doebling et al., 1998; Farrar, 2001; Yan et al., 2007; Wei and Qiao, 2011). In fact, damages or defects may change the mass, stiffness and damping of structures so that the vibrational characteristics like natural frequencies and mode shapes change at the same time. Therefore, the damages can be inversely identified through analyzing these vibrational characteristics.

Tapping method (Cawley and Adams, 1988; Cawley and Adams, 1989) is the earliest vibration based NDT method, thereafter a large number of advanced vibration based NDT methods have been proposed and developed. The analytical model is illustrated in Fig. 2.2. The intact area with contact stiffness is modelled as a spring having a stiffness of K_c ; on the left side, there is damage inside the structure, and then it may be considered the stiffness is weaker than that of the intact area. As a result, the effective stiffness over a defect is the contact stiffness K_c in series with the defect stiffness K_d , which is assumed infinite over the good area.

Fig. 2.2 The analytical model of a tapping method

The typical relationships between contact time and contact force in time and frequency domain are illustrated in Figs. 2.3 (a) and (b) respectively, from which conclusion can be obtained easily: in time domain, the duration of contact time is longer and the amplitude of contact force is lower if tapping is over damaged area; in frequency domain, the amplitude of contact force over undamaged area falls off slowly with increasing frequency, while the energy of impact on the damaged area concentrates on a narrow band of frequency.

(a)

(b)

Fig. 2.3 The relationship between amplitude of contact force and (a) contact time, and (b) frequency

In the earlier days, natural frequency was usually used to detect damages because it can be measured more easily and accurately. Moreover, the natural frequency is found to decrease with the presence of damage which reduces the stiffness of structure. Adams et al. (1978) and Cawley and Adams (1988) firstly pointed out that damage location can be represented as a function of the ratio of the change of any two modal frequencies, and therefore, damage location can be determined through measuring multiple modal frequencies. However, Chen et al. (1995) later questioned the effectiveness of this approach and verified that the change of natural frequencies was polluted by noise and measurement errors. Messina et al. (1996) proposed Multiple Damage Locations Assurance Criterion which had been verified by several simple laboratory experiments;

but Palacz and Krawczuk (2002) illustrated that it was only effective when the measurement noise was less than 0.1%.

As the understanding on this research topic increases, researchers have discovered that global properties like natural frequencies are not sensitive to local damages (Doebbling et al., 1998; Farrar, 2001; Wei and Qiao, 2011). Therefore, it is ineffective to use these vibrational characteristics directly; however, they still use these vibrational characteristics for damage identification because of their ease of measurement. In order to improve the effect of using vibrational characteristics, several advanced dynamics properties based on mode shapes have been proposed, namely, mode shape curvature, flexibility matrix, strain energy and flexibility curvature.

2.2.7.1 Mode shape curvature based method

Pandey et al. (1991) first proposed the idea of mode shape curvature, which is the second derivative of mode shape with respect to location. The mode shape curvature is usually presented by a central difference as

$$\kappa(x_i) = \frac{\varphi(x_{i-1}) + \varphi(x_{i+1}) - 2\varphi(x_i)}{\Delta l^2} \quad (2.1)$$

where $\varphi(x_i)$ is the value of mode shape at x_i , Δl is the distance between two adjacent testing points.

Conventionally, the mode shape curvatures of structures with and without damage can be compared directly; however, the prior information of mode shape curvature of undamaged structure is difficult to obtain, especially for old structures constructed many years ago. In order to overcome the drawback of the requirement of prior information, Ratcliffe (1997) and Yoon et al. (2009) found that the mode shape curvature of undamaged structure is usually smooth but that of damaged structure has anomaly at the location of damage. Then, they developed the anomaly detection method based on curve fitting, namely, Gapped Smoothing Method (GSM) and Global Fitting Method (GFM). GSM does not require comparison but it just looks for anomalies through fitting with a smooth polynomial function. While for GFM, the fitted curve was smoother, and it was simulated as prior knowledge of mode shape curvature of undamaged structures; then it

was used to compare with mode shape curvature of damaged structures and consequently the anomaly which indicated the damage can be identified.

Mode shape curvature is more sensitive to local damage than natural frequency and mode shape; it has been proven by a number of researchers (Salawu and Williams, 1994; Xiang and Zhang, 2009). Some of these researchers found that mode shape curvature could be substituted by strain measured by strain gauges; however, it is ineffective for very large structures. Currently, mode shape curvature is usually obtained by a central difference of measured mode shape, which depends highly on the accuracy of measurement. Moreover, the calculated error of a central difference also influences the effect of this approach. In addition, the measurement requires many sensors and the selection of sensors location should be considered carefully. Hence, mode shape curvature based approaches provide promising results; however, they still have several limited applications in practice.

2.2.7.2 Flexibility matrix based method

Pandey et al. (1994) proposed flexibility matrix based method after they developed the mode shape curvature based method. The flexibility matrix is defined as

$$\mathbf{F} = \sum_{i=1}^n \mathbf{F}_i = \sum_{i=1}^n \frac{1}{\omega_i^2} \boldsymbol{\varphi}_i \boldsymbol{\varphi}_i^T \quad (2.2)$$

where ω_i is the i th natural frequency, $\boldsymbol{\varphi}_i$ is the corresponding mode shape. Therefore, the difference matrix can be presented as

$$\Delta = |\mathbf{F}^d - \mathbf{F}| \quad (2.3)$$

where d indicates the damaged state. Finally, the damage index is defined as the maximum absolute values in each column of Δ :

$$\delta_j = \max_{k=1,2,\dots} |\delta_{kj}| \quad (2.4)$$

From Eqs. (2.2) to (2.4), it can be seen that both natural frequencies and mode shapes are included, and this is the advantage of this approach. Natural frequencies will increase rapidly with the order of mode, therefore, the flexibility matrix converges quickly by using the first few modes and it can be accurate enough even though only the first few modes are used. Similar to mode shape curvature based approaches, the measurement of mode shape is crucial, and hence, the selection of sensors location has to be optimized case by case which is expensive and time consuming. Furthermore, this approach also requires prior information of the undamaged structures which is usually difficult to obtain in practice.

2.2.7.3 Strain energy based method

Stubbs et al. (1992) proposed the idea of strain energy and for one element it was defined as

$$U_{ij} = \frac{1}{2} \int_{x_i}^{x_{i+1}} EI \left(\frac{\partial^2 \varphi_j}{\partial x^2} \right)^2 dx \quad (2.5)$$

where i and j indicate the i th test point and j th mode, EI is the bending stiffness between the i th and $i+1$ th test point.

Similarly, the strain energy of one element in damaged structure can be obtained:

$$U_{ij}^d = \frac{1}{2} \int_{x_i}^{x_{i+1}} EI^d \left(\frac{\partial^2 \varphi_j^d}{\partial x^2} \right)^2 dx \quad (2.6)$$

where EI^d is the stiffness between the i th and $i+1$ th test point of the damaged structure, φ_j^d is j th mode shape of damaged structure. Finally, the damage index can be presented as

$$\beta_j = \frac{\sum_i EI_i}{\sum_i EI_i^d} = \frac{1}{n} \sum_{i=1}^n \frac{\int_{x_i}^{x_{i+1}} \left(\frac{\partial^2 \varphi_j^d}{\partial x^2} \right)^2 dx \int_0^l \left(\frac{\partial^2 \varphi_j}{\partial x^2} \right)^2 dx}{\int_{x_i}^{x_{i+1}} \left(\frac{\partial^2 \varphi_j}{\partial x^2} \right)^2 dx \int_0^l \left(\frac{\partial^2 \varphi_j^d}{\partial x^2} \right)^2 dx} \quad (2.7)$$

It can be observed that the basic idea of strain energy is similar to mode shape curvature because it uses the second derivative of mode shape as well. Therefore, the damage index is also sensitive to local damage (Shi and Law, 1998; Shi et al. 2000), but it encounters the same dilemma that the mode shape is difficult to be measured accurately. Fortunately, the integration of the second derivative of mode shapes does help to reduce measurement and calculation errors.

2.2.7.4 Flexibility curvature based method

Inspired by the mode shape curvature and flexibility matrix, Zhang and Aktan (1995) proposed the flexibility curvature based method. The flexibility matrix is defined similar to Eq. (2.2), but the difference matrix becomes

$$\Delta = \sum_{i=1}^n |F_i^{d''} - F_i''| \quad (2.8)$$

where i relates to the i th mode, d indicates the damaged status and F is the flexibility matrix.

This approach is the combination of mode shape curvature and flexibility matrix; therefore, it has the advantages of these two methods such as high sensitivity to local damage and requirement of first few modes. However, it has to meet the difficulties of obtaining highly accurate mode shapes.

2.2.7.5 Other vibration based methods and indices

In addition to the above methods, several advanced vibration based methods have also been reported recently, for example, frequency-response function and operational mode shape based method (Lee and Shin, 2002; Fanning and Carden, 2003; Fanning and

Carden, 2004; Limongelli, 2010; Limongelli, 2011; Limongelli, 2014), wavelet based methods (Hou et al., 2000; Liew and Wang, 1998; Lu and Hsu, 2002; Zhu and Law, 2006; Zhu and Law, 2007), and finite element model updating method (Fritzen and Jennewein, 1998). Several researchers also proposed damage detection method based on the vibration signal from a vehicle passing through the structures (Yang et al., 2004; Lin and Yang, 2005; Yang and Lin, 2005; Yang and Chang, 2009; Zhang et al., 2012; Zhang et al., 2013a). Moreover, Zhong and Oyadiji (2008a) proposed a natural frequency based damage detection method by using an auxiliary mass after Low (2003) analyzed the change of frequencies due to multiple lump masses. Later, they (Zhong and Oyadiji, 2008b; Zhong et al., 2009) used the wavelet transformation and the derivative of natural frequencies of the structure by probing the auxiliary mass over the entire structure. Zhang and Xiang (2011), Zhang et al., (2013b) subsequently analyzed the relationship between natural frequencies and the location of auxiliary mass, and extended this approach to plate structures.

The above proposed methods required the information from undamaged structures as a reference, and this is usually difficult to obtain in practice. To overcome this problem, several researchers had developed some damage indices which avoid the requirement of information from undamaged structures. Ractcliffe (1997) first proposed gap smoothing method for beam structures which uses operating curvature shapes to substitute mode shape curvatures but it frequently showed misleading results at the boundary of the structures. Wu and Low (2004) extended this index to 2-D structures and Yoon et al. (2005; 2009; 2010) improved this index to global fitting method which uses trigonometric functions to fit the operating deflection shape. Zhang et al. (2013b) also developed global filtering method in which the local damage was considered as an anomaly.

In addition, a few researchers (Chandrashekhar and Ganguli, 2009a; Chandrashekhar and Ganguli, 2009b; Chandrashekhar and Ganguli, 2009c; Beena and Ganguli, 2011) studied the machine learning and pattern recognition to improve the performance of the above methods and make them capable of identifying the severity of the damage.

2.3 NDT methods for CNG cylinder

As described above, a number of NDT methods have been proposed and developed for the past decades; however, they can rarely be used directly for damage detection and

evaluation of CNG cylinders because they are time-consuming and expensive. In the following sections, the NDT methods which are effective for CNG cylinders will be reviewed in details.

2.3.1 Visual inspection for CNG cylinder

Visual inspection is one of the most frequently used testing methods for CNG cylinders. Periodic inspection, the interval of which is 36 months, is important to keep the integrity of CNG cylinders in service. According to ISO 19078 (2006), various defects such as damaged label and mark, cuts, scratches, gouges, abrasion, charring and soot, bulge, surface corrosion, dents and loose brackets need to be identified through regular visual inspection. In order to ensure the accuracy of inspection results, only trained and qualified inspectors can perform this regular inspection. Recently, a number of advanced equipments like high resolution camera and advanced techniques such as the latest image processing methods have been developed to improve the effectiveness of visual inspection. With the help of these advanced equipments and techniques, micro defects that can hardly be identified by naked eyes, can now be detected. The regular visual inspection is easy to implement; however, it is usually time-consuming and the inspection interval is long.

2.3.2 Hydrostatic testing for CNG cylinder

Hydrostatic testing is also used to test leakage of CNG cylinders, especially at the pre-service stage. It is recommended in some standards (ISO 11439, 2000; ISO 19078, 2006), however, it is not completely full proof. In fact, the water injected in hydrostatic testing may cause corrosion, decreasing the wall thickness of the CNG cylinder. Moreover, the gauges and valves need to be removed during the testing. As a result, this process increases the risk of improperly re-installing, the thread strength may be reduced and the high pressure gas may leak. More importantly, it has been proven that this method is not so sensitive to minor damages, and therefore, it has been removed from a few standards (ANSI/CSA NGV 2, 2000).

2.3.3 Ultrasonic testing for CNG cylinder

Ultrasonic testing (Dustin et al., 2004; Giurgiutiu et al., 2004; ISO 11439, 2003; ISO 19078, 2003; Lin and Giurgiutiu, 2011; Rose, 2002; Thien et al., 2005) is the most

frequently used NDT method for regular inspection of CNG cylinders because it is much more accurate than other NDT methods. It is usually used for flaw detection and thickness measurement. For flaw detection, pulse-echo system is often used while for thickness measurement either resonance method or pulse-echo system is utilized. Before the test is carried out, the external surface is carefully cleaned, and then couplant is applied to ensure the transmission of ultrasonic energy.

For the flaw detection, the CNG cylinder should be able to rotate and translate at a constant velocity so that the probe can detect the CNG cylinder helically (ISO 11439, 2003). Furthermore, the half width of the helix should be less than the diameter of the probe to make sure the entire CNG cylinder can be covered. Sometimes the guided wave is more convenient because the damage can be displayed without moving the probes.

The periodical check is crucial for keeping the effectiveness of the equipment. A reference standard is always prepared from a cylindrical shell of similar dimension which is made of the materials with the same acoustic characteristics. Longitudinal and transverse reference notches are introduced on both the external and internal surfaces and they have to be separated so that there would be no interference. The reference notches for longitudinal defects and circumferential defects can be seen in Fig. 2.4. The length of the notches (E) should be less than 50mm; the depth of the notches (T) should be 5% of the thickness with a minimum of 0.2mm and a maximum of 1mm; the width (W) of the notches should be less than twice the depth.

The wall thickness measurement is different from the flaw detection as it is usually conducted at the end of the production stage. It also uses reference standard specimen to ensure the measurement accuracy and consistent.

Fig. 2.4 The reference notches for (a) longitudinal and (b) circumferential defects (ISO 11439, 2003)

Although ultrasonic testing can detect minute defects which can be as small as the wavelength of the ultrasonic waves, however, it is very time-consuming and the equipment is too expensive. Moreover, the ultrasonic signal is often too complicated to interpret so that only trained, qualified and experienced technicians can carry out this testing professionally.

2.3.4 Acoustic emission testing for CNG cylinder

Acoustic emission (Bohse et al., 2006; Jee et al., 2012; Tsimogiannis et al., 2002) indicates the phenomenon that transient elastic waves are produced by “a sudden

redistribution of stress in a material”. Considering a crack initiated at the internal wall of a CNG cylinder, as the internal pressure increases, it grows and propagates and the energy is released in the form of stress waves to the surface which can be recorded by pre-installed sensors with an integral pre-amplifier. The recorded energy amplitude of this mechanical movement is then compared with a pre-determined threshold. Once it exceeds, a digital pulse is produced simultaneously and these pulses are counted. Finally, the total number of such pulses is the indicator to determine whether there is any defect. Moreover, multi-channels measurement circuits are developed to locate the acoustic source. In practice, the linear location, zonal location and point location techniques are the three most frequently used acoustic source location methods.

Acoustic emission does not require power supply because it only “listens” to the acoustic signal produced by the tested specimen itself. It also highly focuses on the dynamic changes which highlights the active features like crack growth and propagation. However, it usually provides the qualitative analysis only and requires noiseless test condition which is inconvenient for industrial and commercial use.

2.3.5 Electromechanical Impedance based NDT for CNG cylinder

Electromechanical Impedance (EMI) based NDT has received a lot of attention recently because it is sensitive to local damages and it can be used for real time monitoring. In fact, the structural mechanical impedance changes once local defect occurs. Small size Piezoceramic Transducers (PZT) are usually pre-bonded on structures, and therefore, the EM coupling between the PZT transducers and the structure will also change accordingly. Then, the PZT EM admittance spectrum at different time can be used to compare with each other, and the change in the spectrum indicates the change of EM and structural integrity. Theoretically and experimentally, it is found that leftward shift of the predominant peaks occurs if local defect exists.

Up to now, the EMI based NDT technique has been applied to various structures like beams, plates and shells. It is especially suitable for circular cylindrical shell which is the main part of a CNG cylinder. Rossi et al. (1993) first proposed the EMI-type model, and it could predict a good velocity response of a circular ring. Zhou et al. (1994) derived an analytical solution of the EMI model of thin cylindrical shell bonded with PZT transducer. Lalande et al. (1995) calculated the structural mechanical impedance of a cylindrical shell by finite element method (FEM). Yang et al. (2008) recently proposed

the EMI model of PZT transducer of cylindrical shells, obtained the EMI of PZT analytically, and furthermore, showed the capacity of detecting local defect on cylindrical shells through comparing the spectrum between damaged and undamaged structures.

Although the damage can be identified as a special type of reverse problem, it is still difficult to locate the damage and evaluate the severity of the damage. In addition, it usually requires the information from the undamaged structure as a reference which limits its applications in practice.

2.3.6 Vibration based NDT for CNG cylinder

In the past, the majority of the vibration based NDT methods were proposed for beam and plate-like structures, and some typical ones have been reviewed in Section 2.2.6. Only very few of them can be extended to cylindrical shells and they will be reviewed in details in this subsection. Because cylindrical shell is the main part of the CNG cylinder and local damages are always found at the cylindrical part, these methods are believed to be capable of detecting local defects.

2.3.6.1 Modal strain energy based damage index algorithm

Srinivasan and Kot (1998) first extended the modal strain energy based damage index algorithm to cylindrical shells and the strain energy of the cylindrical shell is defined as

$$\begin{aligned}
 U = \int_0^l \int_0^{2\pi} \frac{Eh}{2(1-\nu^2)} & \left\{ u_x^2 + \frac{1}{R^2} (v_\theta + w)^2 + \frac{2\nu}{R} u_x (v_\theta + w) + \frac{1}{2} (1-\nu) \left(v_x + \frac{1}{R} u_\theta \right) \right. \\
 & + \frac{h^2}{12} \left[w_{xx}^2 + \frac{1}{R^4} (w_{\theta\theta} - v_\theta)^2 + \frac{2\nu}{R^2} w_{xx} (w_{\theta\theta} - v_\theta) \right. \\
 & \left. \left. + \frac{2(1-\nu)}{R^2} (w_{x\theta} - v_x)^2 \right] \right\} R d\theta dx
 \end{aligned} \tag{2.9}$$

where R , h , l are the radius, thickness and length of the cylindrical shell respectively; E and ν are the elastic modulus and the Poisson ratio; u , v , w are the displacements in the x , θ , r directions.

If the displacements are substituted by mode shapes in Eq. (2.9), the strain energy becomes modal strain energy. Usually the cylindrical shell is uniformly divided into

several small parts, and the modal strain energy in each small part can be calculated accordingly. Then a complicated index based on the modal strain energy before and after damage can be used for determining the local defect. Hence, the quality of the measured mode shape is crucial for identifying the damage successfully. However, this approach usually requires the prior knowledge of the undamaged cylindrical shell; it is difficult in practice because many of the cylindrical shells have been used for years.

Marwala and Hunt (2000) validated the modal strain energy based methods by numerical examples and experimental results. They also found that using modal strain energy and modal properties directly showed better performance than using the correlation criteria like Modal Assurance Criterion (MAC) and COordinate Modal Energy Assurance Criterion (COMEAC).

Hu et al. (2011) also validated this approach by numerical examples and experimentally testing “an aluminium circular hollow cylinder with 200 mm diameter, 505 mm height and 5 mm thickness”, which is marked by 20×17 grid points. An artificial surface crack with a length of 30 mm, width of 1 mm and depth of 2.5 mm was introduced. They also used differential quadrature method to better calculate modal strain energy. By using differential quadrature method, the partial derivative of a function could be evaluated by multiplication of a weighting function.

It was concluded that the modal strain energy based NDT method could successfully detect a surface crack in the numerical examples when the element mesh was fine enough. With the help of scanning damage index method (Hu et al., 2011), a surface crack was identified experimentally. However, this method depends highly on the mode shape which is difficult to be measured accurately; furthermore, the mode shape before and after damage shows small difference because the mode shape is not sensitive to local damage. Therefore, these factors limit the application of this method in practice.

2.3.6.2 Wavelet based modal analysis

Signal processing techniques are very important in conventional vibration testing and modal analysis, and hence, they have received a lot of attention in recent years. Some time frequency analysis techniques have been proposed and developed such as Short Time Fourier Transform, Wavelet Transform (Raghuyeer and Bopardikar, 1998) and Hilbert-Huang Transform (Huang et al., 1998). They have been widely used in damage

detection approaches for various structures; especially, some minor defects which can hardly be detected directly from the structure dynamic response, can now be identified through the wavelet coefficients.

The continuous wavelet transform of a one dimensional signal $f(x)$ is defined as

$$Wf(u, s) = \frac{1}{\sqrt{s}} \int_{-\infty}^{+\infty} f(x) \Psi^* \left(\frac{x-u}{s} \right) dx \quad (2.10)$$

where $\Psi^*(x)$ is the complex conjugate of the wavelet function, s is the scale factor, u is the translational value, and $Wf(u, s)$ is the corresponding wavelet coefficient.

Generally, a wavelet has n vanishing moments which determine how a function decays toward and are crucial in detecting the singularities of signals, if the following condition has been satisfied:

$$\int_{-\infty}^{+\infty} x^k \Psi(x) dx = 0, \quad k = 0, 1, 2, \dots, n-1 \quad (2.11)$$

Rucka and Wilde (2006) used the experimental modes and wavelet analysis to detect defects on a cylindrical shell. The location of the defect could be identified by a peak in the spatial variation of the wavelet transform of the mode shape. Furthermore, the artificial neural network (multilayer feed-forward network) is capable of predicting the location of defect even if the environment is noisy.

2.3.6.3 Vibration Power Flow Analysis

The vibration of a structure can be considered as a special case of structural wave propagation. The presence of a crack will change the motion of a wave and consequently influence the power flow characteristics, where power is the rate at which work is done (Goyder and White, 1980). Therefore, the vibrational power flow analysis could be used for damage detection.

Li et al. (2004a) first studied the power flow of the cracked periodic beam based on which the cracks were identified. Later they (Li et al., 2004b; Zhu, 2007) used the same

method to detect peripheral surface crack on a circular plate. Furthermore, they investigated the power flow characteristics of an infinite thin cylindrical shell to detect circumferential surface crack in which the crack was modelled as a distributed line spring.

It is concluded from their studies that the presence of a crack will change the characteristics of power flow and the crack location and depth will also influence on the input and transmitted power flow. As a result, the crack location and depth can be identified by using different driving frequency. However, this method has only been verified by numerical examples and needs to be verified by experiments in the future.

2.3.6.4 Locating damage based on combination of frequency sensitivities and mode shapes

The frequency sensitivity (Messina, 1998) is defined by Eq. (2.12) based on the assumption that the damage would cause a reduction of stiffness without change of mass:

$$\frac{\partial f_k}{\partial \beta_j} = \frac{1}{8\pi^2 f_k} \frac{\{\phi\}_k^T [k_j] \{\phi\}_k}{\{\phi\}_k^T [M] \{\phi\}_k} \quad (2.12)$$

where f_k is the k th frequency of undamaged structure; $[M]$ is the global mass matrix; $\{\phi\}_k$ is the k th mode shape; β_j is the stiffness reduction factor.

By normalizing $\{\phi\}_k$ to the global mass matrix so that $\{\phi\}_k^T [M] \{\phi\}_k = 1$, Eq. (2.12) can be re-written as

$$S_{k,j} = \frac{1}{f_k} \frac{\partial f_k}{\partial \beta_j} = \frac{SE_{j,k}}{\omega_k^2} \quad (2.13)$$

where $\omega_k = 2\pi f_k$, $SE_{j,k}$ is the maximum modal strain energy of the j th element.

Then, the fractional change in frequency can be expressed as

$$\frac{\Delta f_k}{f_k} = S_{k,j} \times \Delta \beta_j \quad (2.14)$$

The damage could be located by comparing $\{\Delta f / f\}_k$ to the predicted change in frequency $\{\Delta f / f\}_j$ for each damage mode $j=1, 2, \dots, p$. The damage index, Damage Location Assurance Criterion (DLAC), was proposed by Messina et al. (1998) and expressed and simplified as

$$DLAC(j) = \frac{|\{\Delta f / f\}_m^T \{S\}_j|^2}{(\{\Delta f / f\}_m^T \{\Delta f / f\}_m)(\{S\}_j^T \{S\}_j)} \quad (2.15)$$

The DLAC value is larger than 0 and smaller than 1. It shows no correlation when $DLAC = 0$ and exact damage match when $DLAC = 1$. Therefore, the axial location can be evaluated by DLAC. For the circumferential location, the information of mode shapes has to be used. It can be found that the damage occurs at a position where anti-nodes of the individual modes coincide. That is, the damage occurs at the trough of an even mode while at the peak of an odd mode. Then, the circumferential location of the damage can be determined by comparing any two modes which have an even and odd number of sinusoidal wave patterns.

The mode shape curvature has been demonstrated to be quite sensitive to local damages for beam and plate like structures. Sarker et al. (2011) used Ritz method to analyze the dynamic response of a thin cylindrical shell and illustrated the mode shape curvature can also be used for damage detection on cylinder structures. It is found that the crack changes the natural frequency very slightly, but the mode shape curvature can indicate the presence of crack by the peak on the curve. However, it is difficult to obtain the mode shape accurately to make mode shape curvature reliable in practice. This is the main drawback of the mode shape curvature based method.

2.3.6.5 Identification of damages based on Frequency Response Function (FRF)

Frequency Response Function contains most information of structural dynamic characteristics, and it is widely used in various vibration based non-destructive methods. From the FRFs, the natural frequencies, mode shapes and damping can also be extracted. Many researchers have used the FRF to identify damages on beam and plate-like structures.

Lee et al. (2004) extended the method to the cylindrical shells in which the local damage was simulated as elastic modulus reduction at a small area:

$$E_d(x, \theta) = E[1 - d(x, \theta)] \quad (2.16)$$

where E is the elastic modulus of the undamaged cylindrical shell and $d(x, \theta)$ is the damage distribution function. Through a series of complex calculation of the reverse problem, the damage distribution function can be identified, and therefore, multiple local damages can also be determined simultaneously.

Lee and Kim (2006) had extended this method to identification of multiple directional cracks in a thin cylindrical shell. It is found that the material behaviour of the small material volume containing a line crack can be depicted by effective orthotropic elastic stiffness. Hence, the change of local stiffness may indicate the cracks and their directivities.

This method had been verified by numerical examples (Lee and Kim, 2006) and it identified locations, severities and directions of multiple cracks successfully. When 7% noise was added to the FRF data, the method could still show reliable performance. However, it needs to be verified by experiments, and it usually requires highly accurate FRF which limits its wide application.

2.3.6.6 Split-mode phenomena based fault detection of cylindrical shells

Perfectly axisymmetric structures like circular rings and cylinders always have degenerate mode pairs of vibration. Two modes in one pair have identical natural frequencies and mode shapes except that the mode shapes have a phase shift in circumferential direction. But axisymmetric structures with imperfection will lose the degeneracy characteristics and show the split-mode phenomena. The previously identical natural frequencies split into two frequencies which have slight difference; the previously degenerate pair of modes would be determined by the location of the imperfection. Royston et al. (2000) investigated the split-mode phenomena and proposed the fault detection method of cylindrical shells.

Because of the imperfection, the axisymmetry of the cylindrical shell has been damaged. Then a NDT method can be proposed by applying an impact excitation at some different circumferential locations, measuring the vibration at that point by laser Doppler vibrometer and analyzing the vibration in frequency domain. If the structure is perfectly axisymmetric, the vibration amplitudes at certain natural frequency will be identical; if there is imperfection, the vibration amplitudes at certain natural frequency will fluctuate periodically with respect to the circumferential location. This method has been verified by numerical examples and experiments and it shows good performance. However, it is not so easy to implement, limiting its application in practice.

2.4 Summary

This chapter reviews the NDT methods and especially those used for CNG cylinders. Among the various approaches, the vibration based NDT methods for cylindrical shells are reviewed in details. Although there are quite a number of approaches proposed and verified by numerous numerical examples, very few of them have been verified by experiments, making them unreliable. Moreover, the advantages and limitations of each method are analyzed, and from which it can be concluded that further investigation is required to develop a more reliable and easy to use NDT method specifically suited for CNG cylinders.

CHAPTER 3. THE RELATIONSHIP BETWEEN WALL THICKNESS AND NATURAL FREQUENCIES

3.1 Introduction

A number of NDT methods have been proposed for CNG cylinders (or cylindrical shells), and some of them are recommended in various standards. However, as reviewed in Section 2.3, visual testing requires skillful and experienced technicians; ultrasonic testing is accurate but inconvenient, expensive and time consuming and it requires the probe to scan the entire CNG cylinder inch by inch, making it quite difficult to test a CNG cylinder in-situ; acoustic emission testing needs a quiet testing environment and the cylinder has to be pressurized, limiting its in-situ application and making it time consuming as well; smart material like PZT would help to make in-situ and convenient tests but further investigations have to be carried out to make it a reliable NDT system. Therefore, it is important to develop a quick and convenient NDT method in practice as increasingly more CNG cylinders are being used throughout the world.

Vibration based NDT methods provide the quickest, easiest and most convenient operation, although sometimes they are not as accurate as ultrasonic testing. This also makes it possible to develop a quick and in-situ testing method based on vibration. Among the basic vibration characteristics, natural frequencies having a global characteristic are rarely used for damage detection because they do not contain the information of the location. It has been demonstrated that the natural frequency of some structures changes slightly due to the presence of defects, and it can be measured much more easily and accurately than mode shapes and damping. For CNG cylinders or cylindrical shells with similar dimensions, it is fortunate to find that their natural frequencies are relatively high (10^2 Hz – 10^3 Hz) as compared to common beam and plate-type structures (1 Hz – 10 Hz). Therefore, the absolute change of the natural frequencies can be detectable although the relative change is relatively quite small (1%). Moreover, the damage occurring most frequently in CNG cylinders is corrosion which reduces the wall thickness, and hence, decreases the natural frequencies. These natural frequencies can be measured easily and conveniently by a single accelerometer; they are also quite accurate and the resolution that can be achieved is 0.1~0.25 Hz. Considering

these two important aspects, the frequencies can be used as an indicator to detect damages in a CNG cylinder.

In this chapter, the CNG cylinder is simplified as a cylindrical shell. Using the basis of a circular ring, the relationship between wall thickness and natural frequencies is derived. Subsequently, different numerical examples have been adopted to verify this relationship.

3.2 Simplified model and theory

Expressing explicitly the natural frequencies of a CNG cylinder is quite difficult, and therefore, the CNG cylinder has to be simplified. Considering that the main middle part of a CNG cylinder is a cylindrical shell and most susceptible to corrosion, it is modeled as a cylindrical shell, as illustrated in Fig. 3.1. Furthermore, it is simplified as a circular ring because the fundamental vibration mode of a cylindrical shell is the ring like mode.

Fig. 3.1 The simplified cylindrical shell and circular ring

The circular ring has a radius of R and its wall thickness is h . The other material properties are as follows: the elastic modulus E , the material density ρ . U_R and U_θ are displacements in the R direction and θ direction respectively.

The governing equations (Royston et al., 2000; Werner, 2005) are written as:

$$L_1 u_\theta(\theta, t) - L_2 u_R(\theta, t) = 0 \quad (3.1)$$

$$L_2 u_\theta(\theta, t) - L_3 u_R(\theta, t) = 0 \quad (3.2)$$

where

$$L_1 = \frac{Eh}{R^2} \left[(1+p) \frac{d^2}{d\theta^2} - \frac{1}{\omega_0^2} \frac{d^2}{dt^2} \right] \quad (3.3)$$

$$L_2 = \frac{Eh}{R^2} \left[p \frac{d^3}{d\theta^3} - \frac{d}{d\theta} \right] \quad (3.4)$$

$$L_3 = \frac{Eh}{R^2} \left[1 + p \frac{d^4}{d\theta^4} + \frac{1}{\omega_0^2} \frac{d^2}{dt^2} \right] \quad (3.5)$$

$$\omega_0^2 = \frac{E}{\rho R^2} \quad (3.6)$$

$$p = \frac{h^2}{12R^2} \quad (3.7)$$

Then, the two groups, radial and circumferential of natural frequencies (Royston et al., 2000; Soedel, 2005) can be derived as follows:

$$\omega_{n1}^2 = \frac{K_1}{2} \left(1 - \sqrt{1 - \frac{4K_2}{K_1^2}} \right), n = 0, \pm 1, \pm 2, \dots \quad (3.8)$$

$$\omega_{n2}^2 = \frac{K_2}{2} \left(1 + \sqrt{1 - \frac{4K_2}{K_1^2}} \right), n = 0, \pm 1, \pm 2, \dots \quad (3.9)$$

where

$$K_1 = \frac{(n^2 + 1)E}{a^2\rho} \left(\frac{n^2 h^2}{12a^2} + 1 \right) \quad (3.10)$$

$$K_2 = \frac{n^2(n^2 - 1)^2 E^2 h^2}{12a^6 \rho^2} \quad (3.11)$$

ω_{n1} indicates the radial modes and ω_{n2} refers to the circumferential modes. Considering that $\omega_{n2}/\omega_{n1} \approx n^4 E h^2 / 12 a^4 \rho \gg 1$ for most cases, therefore, only ω_{n1} corresponding to the radial modes will be considered. When $n = 0$ and $n = \pm 1$, $\omega_{n1} = 0$ indicates the rigid body motion, so $n = \pm 2$ refers to the fundamental modes in this study. This is especially so when the cylindrical shell is perfectly axisymmetric, $n = \pm 2$ refers to the two modes in one pair which have identical natural frequencies ($\omega_{21} = \omega_{-21}$) but similar mode shapes with a constant phase shift, which can be seen in Fig. 3.2 (Werner, 2005). In fact, it is a typical eigenvalue problem: a square matrix is singular and therefore one eigenvalue which is considered as frequency is corresponding to two different eigenvectors which are regarded as mode shapes. However, if the cylindrical shell is not axisymmetric, the split mode phenomenon happens. That is, the two modes in one pair which initially have identical natural frequencies split and the two identical frequencies become slightly different (Royston et al., 2000).

Fig. 3.2 Vibration modes of a circular ring

It can be seen from Eq. (3.8) that the $\pm n$ th frequencies of the radial modes are functions of dimensions (radius a and thickness h) and material properties (elastic modulus E and

density ρ). When the wall thickness reduces slightly, the $\pm n$ th frequencies can be expressed in the linear form where h_0 is the initial wall thickness:

$$\omega_{n1}(h) = \omega_{n1}(h_0) + \left. \frac{\partial \omega_{n1}}{\partial h} \right|_{h=h_0} (h - h_0) \quad (3.12)$$

where

$$\frac{\partial \omega_{n1}}{\partial h} = \frac{1}{4\omega_{n1}} \left[\frac{\partial K_1}{\partial h} - \frac{1}{\sqrt{K_1^2 - 4K_2}} \left(K_1 \frac{\partial K_1}{\partial h} - 2 \frac{\partial K_2}{\partial h} \right) \right] \quad (3.13)$$

$$\frac{\partial K_1}{\partial h} = \frac{n^2(n^2 + 1)Eh}{6\rho a^4} \quad (3.14)$$

$$\frac{\partial K_2}{\partial h} = \frac{n^2(n^2 - 1)^2 E^2 h}{6a^6 \rho^2} \quad (3.15)$$

Therefore, when the $\pm n$ th natural frequencies are obtained, the wall thickness can be evaluated according to Eq. (3.12).

3.3 Numerical Examples

Because the frequencies of a cylindrical shell are derived from the simplified model of circular ring, hence, it is necessary to verify the simplification that the length of cylindrical shell does not influence the frequencies. In this section, three numerical examples with different dimensions are used to verify Eq. (3.12). Then, several examples with different wall thickness reduction are considered to assess the wall thickness reduction by the frequencies. All the numerical simulations are analyzed using general purpose finite element software, ABAQUS 6.9.

3.3.1 Verification of simplified model

Three cylindrical shells with different dimensions are adopted to verify Eq. (3.12); their dimensions and models are presented in Table. 3.1 and Fig 3.3, respectively.

ABAQUS shell element called S8R is adopted in these three models. Although less elements can guarantee the convergence of numerical calculation, there are still 32 mesh elements used in the circumferential direction for each model and the width and length of each shell element is nearly the same. This can also be seen clearly in Fig. 3.3.

Fig. 3.3 Three FE models with different dimensions used in ABAQUS

Table. 3.1 The dimensions of the cylindrical shells

	Case I	Case II	Case III
Length (mm)	330	330	660
Radius (mm)	80	160	160

The initial wall thickness is $h_0 = 6$ mm and the material properties are as follows: the elastic modulus $E = 2 \times 10^{11}$ N/m² and the density $\rho = 7800$ kg/m³. Then, ± 2 nd natural frequencies of these three cylindrical shells with respect to the reduction of wall thickness can be easily obtained by Eq. (3.12). $\omega_{\pm 1}(h_0)$ which refers to the initial ± 2 nd natural frequencies in Eq. (3.12) is determined by numerical simulations.

(a) Case I

(b) Case II

(c) Case III

Fig. 3.4 The comparison between theoretical solutions and numerical examples

Fig. 3.4 shows the comparison between theoretical solutions and numerical examples on how the ± 2 nd natural frequencies change with respect to wall thickness. It is found that the numerical examples match the theoretical solutions very well and the split-mode phenomenon does not happen, which verifies Eq. (3.12) as well. Moreover, the following two issues have to be addressed here. Firstly, Eq. (3.12) is based on the assumption that the wall thickness reduces slightly which is less than 4% for the three cases. When the wall thickness reduces dramatically, Eq. (3.12) is not suitable and some revisions have to be proposed. Secondly, although the theoretical solutions match the numerical examples very well, the slopes in Eq. (3.13) show small deviations. This might be because the theoretical solutions are based on the simplified model (circular ring), which do not consider the length of the cylindrical shell; while the numerical solutions are based on the entire cylindrical shell as seen in Fig. 3.3.

3.3.2 The relationship between natural frequencies and mass loss

The relationship between frequencies and mass loss is discussed in this sub-section. As reviewed in Chapter 1, general corrosion and line corrosion will be investigated in this study. Either general corrosion or line corrosion can be simulated as reduction of wall thickness at the corrosion area. Because reduction of wall thickness can never be uniform

in practice, therefore, various cases are considered to simulate the practical conditions as shown in Fig. 3.5. The locations and dimension of the mass loss for various cases have been listed in Table 3.2 clearly. For Case 1, there is no reduction of wall thickness and the initial wall thickness is 6 mm; for other cases, the wall thickness is reduced in the red area and the wall thickness of the green part keeps for 6 mm. The radius is 8 mm and the length is 330 mm for all cases.

Case 1

Case 2

Case 3

Case 4

Case 5

Case 6

Case 7

Case 8

Case 9

Case 10

Case 11

Case 12

Case 13

Case 14

Case 15

Fig. 3.5 The FE models used for various cases

Because the locations and the areas of the corrosion or reduction of wall thickness are different for each case, therefore, the reduction of the wall thickness has been converted to the mass loss for easy comparison.

Table 3.2 The locations and dimensions of mass loss for various cases

	Mass loss in longitudinal direction (mm)	Mass loss in circumferential direction
Case 1	Nil	Nil
Case 2	[82.5, 247.5]	$[0, 2\pi]$
Case 3	$[0, 82.5]$ & $[247.5, 330]$	$[0, 2\pi]$
Case 4	$[0, 330]$	$[0, \pi]$
Case 5	[82.5, 247.5]	$[0, \pi]$
Case 6	$[0, 82.5]$ & $[247.5, 330]$	$[0, \pi]$
Case 7	$[0, 330]$	$[0, \pi/2]$
Case 8	[82.5, 247.5]	$[0, \pi/2]$
Case 9	$[0, 82.5]$ & $[247.5, 330]$	$[0, \pi/2]$
Case 10	$[0, 330]$	$[0, \pi/4]$
Case 11	[82.5, 247.5]	$[0, \pi/4]$
Case 12	$[0, 82.5]$ & $[247.5, 330]$	$[0, \pi/4]$
Case 13	$[0, 330]$	$[0, \pi/8]$
Case 14	[82.5, 247.5]	$[0, \pi/8]$
Case 15	$[0, 82.5]$ & $[247.5, 330]$	$[0, \pi/8]$

(a) +2nd natural frequency

(b) -2nd natural frequency

Fig. 3.6 The relationship between (a) +2nd (b) -2nd natural frequencies and mass loss for various cases

The relationship between ± 2 nd natural frequencies and mass loss is illustrated in Figs. 3.6 (a) and (b) respectively. It is found that the 15 cases can be divided into 2 groups: Cases 1-9 are in the first group and Cases 10-15 are in the second group.

For the first group, the ± 2 nd natural frequencies decrease with increase of the mass loss, matching the theoretical solution of Eq. (3.13) very well, but for the second group, the ± 2 nd natural frequencies calculated by FEM show great deviation with the theoretical solution: +2nd natural frequency decreases rapidly with the increase of the mass loss and the -2nd natural frequency decreases slowly when the mass loss is smaller than 2%, and then it decreases dramatically when the mass loss is larger than 2%. This is because the theoretical solution is based on uniform wall thickness. All the cases in the first group have larger area of wall thickness reduction; it can be considered as global and closely uniform wall thickness reduction. But the cases in the second group have smaller area of wall thickness reduction which can be considered as local. Therefore, for the global wall thickness reduction (or general corrosion), the theoretical solution can be used to assess the mass loss according to the ± 2 nd natural frequencies. However, it is still difficult to determine the exact relationship between the ± 2 nd natural frequencies and mass loss for the local reduction of wall thickness.

To further investigate the relationship between the ± 2 nd natural frequencies and the mass loss, all the 15 cases are divided into 5 sub-groups according to the corrosion area in the circumferential direction: Sub-group 1: the corrosions cover 360° (2π) in the circumferential direction (Cases 1-3); Sub-group 2: the corrosions cover 180° (π) in the circumferential direction (Cases 4-6); Sub-group 3: the corrosions cover 90° ($\pi/2$) in the circumferential direction (Cases 7-9); Sub-group 4: the corrosions cover 45° ($\pi/4$) in the circumferential direction (Cases 10-12); Sub-group 5: the corrosions cover 22.5° ($\pi/8$) in the circumferential direction (Cases 13-15).

(a) Sub-group 1

(b) Sub-group 2

(c) Sub-group 3

(d) Sub-group 4

(e) Sub-group 5

Fig. 3.7 The relationship between ± 2 nd natural frequencies and mass loss for 5 sub-groups

The ± 2 nd natural frequencies with respect to mass loss for these 5 sub-groups are illustrated in Fig. 3.7. For Sub-group 1, the corruptions cover 360° (2π) in the circumferential direction and it keeps the cylindrical shell axisymmetric, therefore, in theory, the ± 2 th natural frequencies are identical and the split-mode phenomenon does not happen. From Fig. 3.7 (a), it can be seen that the $+2$ nd natural frequencies and the -2 nd natural frequencies for Cases 1-3 are actually identical respectively. On the other hand, the ± 2 nd natural frequencies of the three cases match the theoretical solutions very well.

For Sub-group 2, the corruptions cover half, 180° (π) in the circumferential direction so that it will not keep the cylindrical shell axisymmetric anymore, therefore, in theory, the split-mode phenomenon happens and the $+2$ nd natural frequencies shows difference from the -2 nd natural frequencies. This can also be demonstrated in Fig. 3.7(b): the $+2$ nd natural frequencies are slightly lower than the -2 nd natural frequencies for Cases 4-6 respectively. One issue should be addressed here: although there is a difference between the ± 2 nd natural frequencies because of the split-mode phenomenon, the difference is quite slight and it can be ignored for the cases where the corruptions cover half of the cylindrical shell in circumferential direction. The same as Sub-group 1, the ± 2 nd natural

frequencies match the theoretical solutions well and it can be used for evaluation of mass loss according to the ± 2 nd natural frequencies.

For Sub-group 3, the corruptions cover a quarter, 90° ($\pi/2$) in the circumferential direction. Similar to Sub-group 2, the axisymmetry of the cylindrical shell has been damaged. From Fig. 3.7(c), it could be seen that the difference between the $+2$ nd and -2 nd natural frequencies rises with the increase of mass loss which demonstrates the split-mode phenomenon. Although the calculated ± 2 nd natural frequencies by FEM show an obvious deviation from the theoretical solution when the mass loss is greater than 2%, the theoretical solutions can still be used to evaluate the reduction of wall thickness.

For Sub-group 4 and Sub-group 5, the corruptions cover only 45° ($\pi/4$) and 22.5° ($\pi/8$) in the circumferential direction respectively. The corrosion in these cases should be considered as local corrosion which shows quite different characteristics. From Fig. 3.7(d), it is found that for Case 10 and Case 13 in which the corrosion runs through the cylindrical shells, the differences between the ± 2 nd natural frequencies increase with the mass loss when it is less than 4%. However, for Cases 11-12 and 14-15 in which the corrosion does not run through the cylindrical shells, the differences between the ± 2 nd natural frequencies increase with the mass loss when it is less than 2% and decrease with the mass loss when it is larger than 2%. Moreover, the calculated ± 2 nd natural frequencies show large deviation from the theoretical solutions. Therefore, the theoretical solutions cannot be used for assessing the severity of local corrosion.

3.4 Summary

In this chapter, the relationship between the wall thickness reduction (or mass loss) and the natural frequencies for various cases are discussed. For general corrosion which reduces the wall thickness on large area (no less than a quarter in the circumferential direction), the differences between the ± 2 nd natural frequencies can be ignored although the split-mode phenomenon occurs. It is also found that the calculated ± 2 nd natural frequencies by FEM match the theoretical solutions very well, and the wall thickness reduction (or mass loss) can be evaluated accordingly. However, for local damage (line corrosion) which reduces the wall thickness on stripe area, the differences between the ± 2 nd natural frequencies are great and they show obvious deviation from the theoretical solutions. Therefore, the theoretical solution cannot be used to evaluate the reduction of wall thickness (or mass loss).

For general corrosion, the damage is global and the evaluation of wall thickness reduction is important; for the line corrosion, the damage is local, and therefore, the location is more important. The theoretical solution proposed in this chapter is not suitable for assessing the location and severity of the local damage; the approximate and effective method will be discussed in Chapters 5 and 6.

CHAPTER 4. THE INFLUENCE OF AUXILIARY MASS ON NATURAL FREQUENCIES

4.1 Introduction

The natural frequencies can be obtained easily by applying the Fast Fourier Transform (FFT) to the vibration signal measured by the accelerometer. However, the mass of the accelerometer cannot be ignored occasionally as it will influence the natural frequencies. For CNG cylinder, its mass is usually not so large that the influence of the accelerometer mass has to be considered.

In this chapter, the influences of auxiliary mass on the natural frequencies of beam and plate-like structures are first reviewed, and then the influence of auxiliary mass on the natural frequencies of cylindrical shell is discussed in details. Subsequently, the approach to eliminate the influence of the auxiliary mass so as to assess the relationship between the mass loss and natural frequencies are demonstrated by numerical simulations and experiments.

4.2 Influence of auxiliary mass on frequencies

The effect of auxiliary mass on natural frequencies of beam and plate-like structures has been investigated by a few researchers. Low (2003) studied the effect of multiple lump masses on the frequencies of beams. Zhong and Oyadiji (2008a) further studied the influence of auxiliary mass on the frequencies of a cracked beam and then they (Zhong and Oyadiji, 2008b) proposed a novel method to detect a crack based on wavelet transform. Later, they (Zhong and Oyadiji, 2009) extended this approach by using the derivative of the natural frequencies which are more sensitive to local damages on beam like structures. Zhang and Xiang (2011) demonstrated analytically the change of natural frequencies with respect to the location of auxiliary mass and they found that the change of frequencies is equivalent to the mode shapes square which is proven sensitive to local damages. Later, they (Zhang et al., 2013b) extended this approach to plate like structures.

4.2.1 Influence of auxiliary mass on natural frequencies of beams

The model of an Euler-Bernoulli beam with a spring-mass system is illustrated in Fig. 4.1. The mass M is supported on a spring of stiffness k at position x_0 of the beam which has bending stiffness EI and mass $\bar{m}$ per unit length.

Fig. 4.1 The interaction system of an Euler-Bernoulli beam and auxiliary mass

The governing equations of this interactive system (Zhang and Xiang, 2011) can be obtained as

$$\bar{m}\ddot{u}(x, t) + EIu''''(x, t) = f(t)\delta(x - x_0) \quad (4.1)$$

$$M\ddot{q}(t) + kq(t) = ku(x_0, t) \quad (4.2)$$

where $q(t)$ and $u(x, t)$ are the vertical displacements of the mass and the beam measured from the static equilibrium position, respectively; $\delta(\cdot)$ is the Dirac delta function and $f(t)\delta(x-x_0)$ is the contact force:

$$f(t) = -M(g + \ddot{q}) \quad (4.3)$$

Using the modal superposition method, the displacement of the beam can be written as

$$u(x, t) = \sum_n \varphi_n(x)q_n(t) \quad (4.4)$$

where $\varphi_n(x)$ is the n th mode shape and $q_n(t)$ is the corresponding modal coordinate.

Substituting Eq. (4.4) into Eq. (4.1) and multiplying $\varphi_m(x)$ on both sides and integrating over the entire beam, one can obtain:

$$\int_0^l \bar{m} \varphi_m \sum_n \varphi_n \ddot{q}_n(t) dx + \int_0^l EI \varphi_m \sum_n \varphi_n'''' q_n(t) dx = \int_0^l \varphi_m f(t) \delta(x - x_0) dx \quad (4.5)$$

Because the auxiliary mass is mounted on the beam directly, hence, the stiffness can be considered infinite. Therefore, the displacement of the mass could be approximated as

$$q(t) \approx u(x_0, t) = \sum_n \varphi_n(x_0) q_n(t) \quad (4.6)$$

Substituting Eqs. (4.3) and (4.6) into Eq. (4.5), and noting the orthogonal property of mode shapes, Eq. (4.5) becomes:

$$\left[1 + \frac{M}{\bar{m} \phi_n} \varphi_n^2(x_0) \right] \ddot{q}_n + \omega_{bn}^2 q_n = -M \frac{g + \sum_{i \neq n} \ddot{q}_i \varphi_i(x_0)}{\phi_n} \varphi_n(x_0) \quad (4.7)$$

where ω_{bn} is the n th natural frequency of the beam and ϕ_n is an integration constant:

$$\phi_n = \int_0^l \varphi_n^2(x) dx \quad (4.8)$$

Thus, the natural frequencies of the interactive system including the beam and auxiliary mass can be approximately written as

$$\omega_n^2 = \frac{\omega_{bn}^2}{1 + \frac{M}{\bar{m} \phi_n} \varphi_n^2(x)} \quad (4.9)$$

It can be found that the natural frequencies of the interactive system are functions of location of the auxiliary mass. If the auxiliary mass can be ignored, it will not influence the natural frequencies of the interactive system. Otherwise, the influence of the auxiliary mass has to be considered.

4.2.2 Influence of auxiliary mass on natural frequencies of plates

The model of a thin plate with auxiliary mass is shown in Fig. 4.2. The mass M is supported on a spring of stiffness k at the position (x_0, y_0) on the plate which has bending stiffness D and mass $\tilde{m}$ per unit area.

Fig. 4.2 The interactive system of a thin plate and auxiliary mass (Zhang et al., 2013b)

Similarly, the governing equations (Zhang et al., 2013b) can be written as

$$\tilde{m}\ddot{u}_p(x, y, t) + D\nabla^4 u_p(x, y, t) = f_p(t)\delta(x - x_0, y - y_0) \quad (4.10)$$

$$\tilde{M}\ddot{q}_p(t) + kq_p(t) = ku_p(x_0, t) \quad (4.11)$$

where $q_p(t)$ and $u_p(x, y, t)$ are the vertical displacements of the mass and the plate measured from the static equilibrium position, respectively; $f_p(t)\delta(x-x_0, y-y_0)$ is the contact force:

$$f_p(t) = -\tilde{M}(g + \ddot{q}_p) \quad (4.12)$$

Using the modal superposition method, the plate displacement can be written as

$$u_p(x, y, t) = \sum_n \sum_m \varphi_{nm}(x, y) q_{nm}(t) \quad (4.13)$$

where φ_{nm} is the (n, m) th mode shape and q_{nm} is the corresponding modal coordinate.

Substituting Eq. (4.12) into Eq. (4.10) and multiplying $\varphi_{nm}(x, y)$ on both sides and integrating over the entire plate, one can obtain:

$$\begin{aligned} \int_0^a \int_0^b \tilde{m} \varphi_{nm} \sum_n \sum_m \varphi_{nm}(x, y) \ddot{q}_{nm}(t) dx dy \\ + \int_0^a \int_0^b EI \varphi_{nm} \sum_n \sum_m \nabla^4 \varphi_{nm}(x, y) q_{nm}(t) dx dy \\ = \int_0^a \int_0^b \varphi_{nm} f_p(t) \delta(x - x_0, y - y_0) dx dy \end{aligned} \quad (4.14)$$

The displacement of the mass could be also approximately written as

$$q_p(t) \approx u_p(x_0, y_0, t) = \sum_n \sum_m \varphi_{nm}(x_0, y_0) q_{nm}(t) \quad (4.15)$$

Substituting Eqs. (4.12) and (4.15) into Eq. (4.14), and noting the orthogonal property of mode shapes, Eq. (4.14) becomes:

$$\left[1 + \frac{\tilde{M}}{\tilde{m} \phi_{nm}} \varphi_{nm}^2(x_0, y_0) \right] \ddot{q}_{nm} + \omega_{nm}^2 q_{nm} = -\tilde{M} \frac{g + \sum_{i,j \neq n,m} \ddot{q}_{ij} \varphi_{ij}(x_0, y_0)}{\phi_{nm}} \varphi_{nm}(x_0, y_0) \quad (4.16)$$

where ω_{nm} is the (n, m) th natural frequency of the plate and ϕ_{nm} is an integration constant:

$$\phi_{nm} = \int_0^a \int_0^b \phi_{nm}^2(x, y) dx dy \quad (4.17)$$

Thus, the natural frequencies of the interactive system including the plate and auxiliary mass can be approximately written as

$$\Omega_{nm}^2 = \frac{\omega_{nm}^2}{1 + \frac{\tilde{M}}{\tilde{m}\phi_{nm}} \phi_{nm}^2(x_0, y_0)} \quad (4.18)$$

It can also be seen that if the auxiliary mass can be ignored, then this will not influence the natural frequencies of the interactive system. However, when it cannot be ignored, the influence of the auxiliary mass has to be considered carefully.

4.2.3 Influence of auxiliary mass on natural frequencies of cylindrical rings

The auxiliary mass fixed on a cylindrical shell changes the natural frequencies of the structure. Although many formulas can be used to describe the interactive system of auxiliary mass and cylindrical shell, it is still very difficult to obtain the explicit solution. In order to get useful information from the explicit solution, a simplified model shown in Fig. 4.3 is adopted instead. In this model, the thin cylindrical shell is simplified as a free circular ring and the auxiliary mass is modeled as a spring-mass system. The ring has bending stiffness EI_r , density ρ_r , cross-sectional area A and radius a . The auxiliary mass M_r is supported at position θ_0 on a spring with large stiffness k_r . For simplicity, the damping factor is ignored in this study.

Fig. 4.3 Axis-symmetric circular ring and an auxiliary mass (Zhang et al., 2014)

Only radial vibration is considered so that the Prescott's equation can be adopted. According to Prescott's theory (Soedel, 2005), the governing equations of the ring and spring-mass system can be written as

$$\frac{EI_r}{a^4} \left(\frac{\partial^4 u_r}{\partial \theta^4} + 2 \frac{\partial^2 u_r}{\partial \theta^2} + u_r \right) + \rho_r A \frac{\partial^2 u_r}{\partial t^2} = f_r(t) \delta(\theta - \theta_0) \quad (4.19)$$

$$M_r \ddot{q}_r(t) + k_r q_r(t) = k_r u_r(\theta_0, t) \quad (4.20)$$

where $q_r(t)$ and $u_r(\theta, t)$ are the vertical and radial displacements of the mass and the ring measured from the static equilibrium position, respectively; $\delta(\cdot)$ is the Dirac delta function and $f_r(t)\delta(\theta-\theta_0)$ is the contact force between the mass and the ring, and

$$f_r(t) = -M_r(g + \ddot{q}_r) \quad (4.21)$$

Using the modal superposition method, the radial displacement of the ring can be represented as

$$u_r(\theta, t) = \sum_{\substack{\pm n \\ n=0,1,2,\dots}} U_{\pm n}(\theta) q_{\pm n}(t) \quad (4.22)$$

where $U_{+n}(\theta) = \cos[n(\theta-\varphi_{+n})]$ and $U_{-n}(\theta) = \sin[n(\theta-\varphi_{-n})]$ are the $\pm n$ th mode shape and $q_{\pm n}(t)$ are the corresponding modal coordinates. In the mode shape expressions, $\varphi_{\pm n}$ are identical and completely arbitrary.

Substituting Eq. (4.22) into Eq. (4.19) and multiplying $U_m(\theta)$ on both sides and integrating over circumference produces:

$$\begin{aligned}
& \int_0^{2\pi} \left[\frac{EI_r}{a^4} \left(U_m(\theta) \frac{\partial^4}{\partial \theta^4} \sum_{\pm n} U_{\pm n}(\theta) q_{\pm n}(t) + 2U_m(\theta) \frac{\partial^2}{\partial \theta^2} \sum_{\pm n} U_{\pm n}(\theta) q_{\pm n}(t) \right. \right. \\
& \quad \left. \left. + U_m(\theta) \sum_{\pm n} U_{\pm n}(\theta) q_{\pm n}(t) \right) + \rho_r A U_m(\theta) \frac{\partial^2}{\partial t^2} \sum_{\pm n} U_{\pm n}(\theta) q_{\pm n}(t) \right] d\theta \\
& = \int_0^{2\pi} U_m(\theta) f_r(t) \delta(\theta - \theta_0) d\theta \quad n = 0, 1, 2, \dots
\end{aligned} \tag{4.23}$$

When the stiffness of the spring k is relatively large, the displacement of the mass could be approximated as

$$q_r(t) \approx u_r(\theta_0, t) = \sum_{\pm n} U_{\pm n}(\theta_0) q_{\pm n}(t) \quad n = 0, 1, 2, \dots \tag{4.24}$$

Substituting Eqs. (4.21) and (4.24) into Eq. (4.23), and noting the orthogonal property of mode shapes, Eq. (4.23) becomes (substituting subscript m by n):

$$\begin{aligned}
& [M_r U_{\pm n}^2(\theta_0) + \rho_r A \Omega_{\pm n}] \ddot{q}_{\pm n} + \frac{EI_r}{a^4} (n^2 - 1)^2 \Omega_{\pm n} q_{\pm n} = -M_r U_{\pm n}(\theta_0) \left(g + \sum_{i \neq \pm n} \ddot{q}_i U_i(\theta_0) \right) \\
& n = 0, 1, 2, \dots
\end{aligned} \tag{4.25}$$

Thus, the natural frequencies of the interactive system, including the ring and the spring-mass system, can be approximated as

$$\omega_{\pm n}^2(\theta_0) = \frac{\frac{EI_r}{\rho_r A a^4} (n^2 - 1)^2}{1 + \frac{M_r}{\rho_r A \Omega_{\pm n}} U_{\pm n}^2(\theta_0)} = \frac{\omega_{\pm rn}^2}{1 + \frac{M_r}{\rho_r A \Omega_{\pm n}} U_{\pm n}^2(\theta_0)} \tag{4.26}$$

where $\Omega_{\pm n} = \int_0^{2\pi} U_{\pm n}^2(\theta) d\theta$ are integral constants.

From Eq. (4.26), it can be seen that the natural frequencies of the interactive system $\omega_{\pm n}^2$ are functions of θ_0 as well. When there is no damage or it is perfectly axisymmetric,

$\omega_{\pm n}^2(\theta_0)$ are constants because $\varphi_{\pm n}$ changes with respect to θ_0 making $U_{\pm n}^2(\theta_0)$ constants. Actually in this situation, $\varphi_{\pm n}$ equal to θ_0 where the auxiliary mass is located, therefore $U_{+n}(\theta) = \cos[n(\theta - \varphi_{+n})] = 1$ and $U_{-n}(\theta) = \sin[n(\theta - \varphi_{-n})] = 0$ are constants. However, the circular ring cannot always be perfectly axisymmetric and it has imperfections or defects, therefore, the split-mode would occur and Eq. (4.26) can be presented as

$$\omega_{\pm dn}^2(\theta_0) = \frac{\omega_{\pm d n}^2}{1 + \frac{M_r}{\rho_r A \Omega_{\pm dn}} U_{\pm dn}^2(\theta_0)} \quad (4.27)$$

where $\omega_{\pm dn}$ are the $\pm n$ th natural frequencies of the new system, $\omega_{\pm d n}$ are the natural frequencies of the damaged ring and $U_{\pm dn}$ are the corresponding mode shapes. Because of defects or imperfections, $U_{\pm dn}$ are determined functions of θ_0 , and therefore, $\omega_{\pm dn}$ change with respect to the position of the auxiliary mass, θ_0 . It can be seen that $\omega_{\pm dn}(\theta)$ are functions of θ , and therefore, they are named the $\pm n$ th Frequency Shift Curve (FSC) respectively in this study. Considering that $U_{+dn}(\theta) \approx \cos[n(\theta - \varphi_{+n})]$ and $U_{-dn}(\theta) \approx \sin[n(\theta - \varphi_{-n})]$ where φ_{+n} and φ_{-n} are determined by the defects or the imperfections respectively, there will be $2n$ peaks and $2n$ troughs on the $+n$ th and $-n$ th FSCs. Usually, φ_{+n} equals to φ_{-n} so that the peaks on the $+n$ th FSC will occur at positions where the troughs on $-n$ th FSC are located.

For a typical cylindrical shell, only the ring-like motion with no nodal points on the longitudinal direction is considered, hence, the representation of Eq. (4.27) will not change. The ring-like motion mode shapes of the cylindrical shell can be written as

$$\phi_{+nq}(\theta, x) = \cos(n[\theta - \varphi_n]) Q_q(x) \quad (4.28)$$

$$\phi_{-nq}(\theta, x) = \sin(n[\theta - \varphi_{-n}]) Q_q(x) \quad (4.29)$$

where the $Q_q(x)$ is an axial function and q indicates the number of nodal points in the longitudinal direction. When $q = 0$, it indicates that the cylindrical shell has only a ring-like motion; $Q_{q=0}(x) = Q_0$ is a non-zero constant, and then $\phi_{\pm n0}(\theta, x) = U_{\pm n}(\theta) Q_0$. For those cases when $q \neq 0$ there are several nodal points in the longitudinal direction, they

are not considered in this study. That is the reason why the developed theory or Eq. (4.27) can be extended to cylindrical shell. In fact, the first two mode shapes of a cylindrical shell usually relates to the ring like motion, that is $\phi_{\pm 2,0}(\theta, x) = U_{\pm 2}(\theta)Q_0$ (Zhang et al., 2014).

For either undamaged or damaged cases, when the auxiliary mass is relatively small ($M_r \ll \rho_r A$), the influence of the auxiliary mass on the natural frequencies can be ignored: no matter where the accelerometer is attached, the measured natural frequencies will be identical. However, when the mass of accelerometer is not always small, its influence on the natural frequencies cannot be ignored. Since the natural frequencies $\omega_{\pm dn}$ fluctuate with respect to the circumferential location of the auxiliary mass, the average of the natural frequencies on several positions along circumferential direction will be used for evaluation.

4.3 Numerical simulations of the influence of auxiliary mass

In this section, several numerical examples are adopted to investigate the influence of auxiliary mass on the natural frequencies. Firstly, the fluctuation of $\omega_{\pm dn}$ in terms of θ_0 will be demonstrated; secondly, the relationship between the mass loss and the average of the natural frequencies will be evaluated. All the numerical examples are simulated by ABAQUS Version 6.9.

4.3.1 Influence of auxiliary mass on natural frequencies

The FEM model is illustrated in Fig. 4.4, and the geometry and material properties are as follows: elastic modulus $E = 200$ GPa, density $\rho = 7800$ kg/m³, radius $a = 80$ mm, thickness $h = 6$ mm, length $L = 330$ mm, mass $M = 0.3$ kg and spring stiffness $k = 2 \times 10^7$ N/m. The cylindrical shell is uniformly meshed into 32 parts in the circumferential direction and 4 parts in the longitudinal direction. The auxiliary mass remains at the same position while the cylindrical shell is rotated by $\pi/16$ each time. Then the natural frequencies of the interactive system can be obtained easily (Zhang et al., 2014).

Fig. 4.4 Finite element model of a cylindrical shell with auxiliary mass (Zhang et al., 2014)

The cylindrical shell with perfectly axis-symmetry is considered first. From the numerical simulation, the ± 2 nd natural frequencies of the perfect cylindrical shell are obtained and plotted in Fig. 4.5. As expected, the ± 2 nd natural frequencies are constants and do not change with respect to the location of auxiliary mass.

Fig. 4.5 The ± 2 nd FSCs of the perfect cylindrical shell (Zhang Y. et al., 2014)

Then, the cylindrical shell with imperfection is considered. An artificial damage is introduced into the cylindrical shell by reducing the thickness from 6 mm to 5.5 mm. The dark area shown in Fig. 4.6 represents the damage located at $[7\pi/16, 9\pi/16]$ in the circumferential direction and $[0, L]$ on the longitudinal direction, respectively.

Fig. 4.6 Finite element model of a damaged cylindrical shell

The auxiliary mass is located along the middle line and the ± 2 nd FSCs of the damaged cylindrical shell are shown in Fig. 4.7. As expected, 2×2 peaks and 2×2 troughs occur on the ± 2 nd FSCs; the peaks on +2nd FSC occur at the positions where the troughs on the -2nd FSC are located. It also demonstrates that +2nd FSC shows a fixed phase difference of $\pi/2$ compared to the -2nd FSC.

Fig. 4.7 The ± 2 nd FSC of the damaged cylindrical shell

4.3.2 Relationship between mass loss and average natural frequencies

Because the natural frequencies fluctuate with respect to the location of auxiliary mass, it cannot be ignored. The average frequencies of FSCs are adopted to assess the relationship of mass loss and natural frequencies.

The same cases are chosen herein, and the FEM models are illustrated in Fig. 3.4. For Cases 1, there is no local reduction of wall thickness and it is perfectly axisymmetric; for Case 2 and Case 3, although the wall thickness reduces at the red part, the cylindrical shell is still perfectly axisymmetric. However, for Cases 4-15, the reduction of wall thickness at the red part damages the axisymmetry of the cylindrical shell. For all cases, the reduction of wall thickness is converted to the mass loss and the relationship between the mass loss and the average natural frequencies is presented and discussed.

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.8 The ± 2 nd FSCs of Case 1

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.9 The ± 2 nd FSCs of Case 2

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.10 The ± 2 nd FSCs of Case 3

Figs. 4.8 to 4.10 show the ± 2 nd FSCs of Cases 1–3. It can be found that the ± 2 nd natural frequencies keep constants and do not change with respect to the angular location of the auxiliary mass. This is because the cylindrical shells in Cases 1–3 are perfectly axisymmetric and the cylinder-mass systems are identical no matter where the auxiliary mass is mounted in the circumferential direction. When the auxiliary mass makes the

cylinder-mass system non-axisymmetric, then the split-mode phenomenon occurs and the ± 2 nd natural frequencies are different. With the increase of mass loss, the ± 2 nd natural frequencies decrease gradually.

(a) $+2$ nd FSCs

(b) -2 nd FSCs

Fig. 4.11 The ± 2 nd FSCs of Case 4

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.12 The ± 2 nd FSCs of Case 5

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.13 The ± 2 nd FSCs of Case 6

For Cases 4–6 (Figs. 4.11 to 4.13), the cylindrical shells have a large area (a quarter to a half) reduction of wall thickness. When the wall thickness only reduces slightly or the mass loss is very small, it seems that the ± 2 nd natural frequencies are not dependent on the location of the auxiliary mass and still remain constant. When the mass loss is larger than 1%, the ± 2 nd FSCs show four peaks and four troughs clearly and the peaks on the

+2nd FSC occur at the positions where the troughs on the -2nd FSC are located, which coincides with the results of numerical examples very well. Moreover, with the increase of the reduction of wall thickness, the fluctuations of ± 2 nd natural frequencies increase accordingly.

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.14 The ± 2 nd FSCs of Case 7

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.15 The ± 2 nd FSCs of Case 8

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.16 The ± 2 nd FSCs of Case 9

For Cases 7–9, the ± 2 nd FSCs are similar to those of Cases 4–6. When the wall thickness reduces slightly, the fluctuations of ± 2 nd FSCs are quite small and can be considered constants; when the mass loss becomes larger, the fluctuations are more obvious. With the increase of wall thickness reduction, the fluctuations of ± 2 nd FSCs increase accordingly.

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.17 The ± 2 nd FSCs of Case 10

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.18 The ± 2 nd FSCs of Case 11

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.19 The ± 2 nd FSCs of Case 12

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.20 The ± 2 nd FSCs of Case 13

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.21 The ± 2 nd FSCs of Case 14

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.22 The ± 2 nd FSCs of Case 15

For Cases 10–15, the ± 2 nd FSCs show great fluctuation even if the mass loss is as low as 0.5%. This is because the wall thickness reduction area is comparatively small, and therefore, the reduction of wall thickness is high, which makes the cylindrical shell more non-axisymmetric. With the increase of mass loss, it seems that the fluctuation amplitudes of the ± 2 nd FSCs remain the same, which is quite different from Cases 1–9.

(a) +2nd natural frequency

(b) -2nd natural frequency

Fig. 4.23. The relationship between averages of (a) +2nd and (b) -2nd natural frequencies and mass loss for various cases

The relationship between the average of ± 2 nd natural frequencies and the mass loss for all the 15 cases are plotted in Figs. 4.23(a) and (b) respectively. It is found that Fig. 4.23 is quite similar to Fig. 3.5, which demonstrates that the average of ± 2 nd natural frequencies can eliminate the influence of auxiliary mass. Compared with Fig. 3.5, the data points in Fig. 4.23(a) decrease by nearly 3Hz because of the auxiliary mass; however, Fig. 4.23(b) is almost identical to Fig. 3.6 which demonstrates that the auxiliary mass does not influence the average of the -2nd natural frequencies although it brings fluctuation to the -2nd natural frequencies. The possible reasons will be discussed further in Chapter 5.

The same conclusion discussed in Chapter 3 can also be obtained from Fig. 4.23: for the global reduction of wall thickness (Cases 1-9), the theoretical solution can be used to

assess the mass loss according to the average of ± 2 nd natural frequencies; for the local reduction of wall thickness (Cases 10-15), it is still difficult to determine the exact relationship between the average of ± 2 nd natural frequencies and mass loss.

Similarly, all the 15 cases are divided into 5 subgroups according to the damage area in circumferential direction. The relationship between the mass loss and the average of ± 2 nd natural frequencies are discussed in details.

(a) Sub-group 1

(b) Sub-group 2

(c) Sub-group 3

(d) Sub-group 4

(e) Sub-group 5

Fig. 4.24 The relationship between averages of the ± 2 nd natural frequencies and mass loss for 5 sub-groups

For Sub-group 1, although the reduction of wall thickness does not damage the axisymmetry of the cylindrical shell, the auxiliary mass makes the cylinder-mass system non-axisymmetric so that the ± 2 nd modes split and the averages of ± 2 nd natural frequencies show variation, which can be seen in Fig. 4.24(a). Moreover, it is observed that the averages of ± 2 nd natural frequencies match the analytical solution well and the difference between +2nd natural frequencies and -2nd natural frequencies barely change with respect to mass loss.

For Sub-group 2, the reduction of wall thickness covers half of the circumferential direction, damaging the axisymmetry of the cylindrical shell. It can also be observed that the averages of ± 2 nd natural frequencies match the analytical solution well, especially when the mass loss is small.

For Sub-group 3, the damages cover only a quarter of the circumferential direction. However, the Cases 7–9 can also be considered as global reduction of wall thickness, and the averages of ± 2 nd natural frequencies also match the analytical solution. When the wall thickness reduces slightly, the averages of ± 2 nd natural frequencies can be predicted by analytical solution very well; as the mass loss increases, the deviation between the

± 2 nd natural frequencies and the analytical solution becomes more pronounced and the differences between $+2$ nd natural frequencies and -2 nd natural frequencies become larger as well.

For Sub-group 4 and Sub-group 5, the damages of Cases 10–15 are considered as local reduction of wall thickness. The averages of ± 2 nd natural frequencies show large deviation from the theoretical solution, therefore, the theoretical solution cannot be used for assessing the severity of local damages.

Moreover, it can be observed that the difference between the ± 2 nd natural frequencies with respect to the mass loss almost remains the same for the global corrosion covering a large area, while it becomes larger for the local corrosion only covering small strip area. This can be used to classify whether the corrosion is global or local and will be further validated in the following sections.

4.4 Experimental study of the influence of auxiliary mass

The experimental examples are used to study the influence of auxiliary mass and the relationship between mass loss and average of ± 2 nd natural frequencies. Because only relationship between the averages of ± 2 nd natural frequencies and mass loss for global reduction of wall thickness matches the theoretical solution well, therefore, large area of wall thickness reduction is introduced to the specimen simulating the global damage. Then, the capacity of the theoretical solution to predict the mass loss according to the ± 2 nd natural frequencies is validated in this section.

4.4.1 Experimental setup

The Experimental setup is shown in Figs. 4.25 and 4.26. The cylindrical shell is supported by soft sponges to simulate a free-free boundary condition. An accelerometer (Wilcoxon Model No. F3/Z602WA) is mounted on top of the cylindrical shell with the help of a small magnetic base. When the cylindrical shell is tapped by a hammer, the vibration signal is converted to the electrical signal which is then amplified through a charger amplifier (Wilcoxon Model No. PA8HF) and sampled at sampling frequency of 2048Hz by data acquisition card (Measurement Computing Model No. USB 2527). Finally, after applying FFT to the acceleration signal, the natural frequencies can be obtained easily. DasyLab ® is used herein for the data acquisition and processing.

Fig. 4.25 The flowchart of experimental set-up

Fig. 4.26 Equipments used in experimental set-up

4.4.2 Experiments of steel cylindrical shells

Two identical steel cylindrical shells are used to validate the theoretical solution. Eq. (3.12) represents the relationship between the reduction of wall thickness converted to mass loss and the average of ± 2 nd natural frequencies. The specimen dimensions are given as follows: radius = 84.5 mm, length = 330 mm and thickness = 6.93 mm. The initial mass of Sample 1 and Sample 2 are 9278 g and 9297 g respectively.

Firstly, the frequencies of these two cylindrical shells are measured, and then artificial reduction of wall thickness is introduced. A grinder is used manually to reduce material from the cylindrical shells. It is hold with a firm grip and moved across the cylindrical shell in the direction of the handle. Only small amount material can be reduced at a time, therefore, the procedure should be repeated several times. Although the surface cannot be extremely smooth, it resembles the erosion surface in practice. For Sample 1, the wall thickness is reduced from the internal surface, similar to the practical situation. For sample 2, the wall thickness is reduced from the external surface which can be repeated on the real CNG cylinder. The reduction of wall thickness is introduced alternatively on the dark gray area and the light gray area shown in Fig. 4.27 to make 0.5% mass loss each time. It can be seen that the wall thickness reduction area is nearly one third of the entire area of the cylindrical shell so that it can be considered as global reduction of wall thickness.

Fig. 4.27 The damaged area of the cylindrical shells

The sequence of photographs showing the two cylindrical shells with different mass loss are given in Fig. 4.28.

Sample 1 (0% mass loss)

Sample 2 (0% mass loss)

Sample 1 (0.5% mass loss)

Sample 2 (0.5% mass loss)

Sample 1 (1.5% mass loss)

Sample 2 (1.5% mass loss)

Sample 1 (2% mass loss)

Sample 2 (2% mass loss)

Sample 1 (2.5% mass loss)

Sample 2 (2.5% mass loss)

Sample 1 (3% mass loss)

Sample 2 (3% mass loss)

Sample 1 (3.5% mass loss)

Sample 2 (3.5% mass loss)

Sample 1 (4% mass loss)

Sample 2 (4% mass loss)

Fig. 4.28 The photographs of cylindrical shells with different mass loss

Figs. 4.29 to 4.30 show the $\pm 2^{\text{nd}}$ FSCs of Sample 1 and Sample 2. It can be observed that the $\pm 2^{\text{nd}}$ FSCs fluctuate with respect to angular location of auxiliary mass and show four peaks and four troughs even for the initial condition (no mass loss). This is because these two cylindrical shells have initial imperfection which cannot be eliminated due to the limitation of fabrication process. With the increase of mass loss, the $\pm 2^{\text{nd}}$ natural frequencies decrease gradually. For Sample 1, the fluctuation amplitudes of $\pm 2^{\text{nd}}$ FSCs are almost identical for different mass loss; for Sample 2, the fluctuation amplitudes of $\pm 2^{\text{nd}}$ FSCs decrease when the mass loss increases and the reason might be that the artificial removal of materials makes the thickness more uniform.

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.29 The ± 2 nd FSCs of Sample 1

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.30 The ± 2 nd FSCs of Sample 2

Fig. 4.31 The relationship between the averages of ± 2 nd natural frequencies and the mass loss for Samples 1 and Sample 2

Then, the relationship between the averages of ± 2 nd natural frequencies and the mass loss for Sample 1 and Sample 2 can be seen in Fig. 4.31. It can be seen that the experimental results match the theoretical solution well for both Sample 1 and Sample 2. For Sample 1 which has an internal reduction of wall thickness, the averages of measured ± 2 nd natural frequencies are a slightly lower than the theoretical solution; for Sample 2 which has an external reduction of wall thickness, the averages of measured ± 2 nd natural frequencies are slightly higher than the theoretical solution. The reason for this slight difference is because the effective radius of the cylindrical shell is larger which will make the natural frequencies lower if the reduction of wall thickness is introduced on the internal surface and vice versa. In fact, from Eq. (3.8), it can also be observed that with the same thickness, the frequency is lower if the radius is larger.

4.4.3 Application on 30 L CNG cylinder

The approach has been applied on a typical 30 L CNG cylinder shown in Fig. 4.32. The CNG cylinder is also divided into 16 parts in the circumferential direction, and then an accelerometer is mounted vertically at a point marked on the cylinder surface. After tapping on the cylinder and measuring its natural frequencies, the cylinder is rotated by 22.5° and the accelerometer is moved to another point.

Fig. 4.32 The photograph of undamaged 30 L CNG cylinder

Firstly, the ± 2 nd frequencies of the undamaged 30 L CNG cylinder are measured. Then, artificial reduction of wall thickness is introduced. The wall thickness is reduced from the external surface alternatively similar to the dark gray area and light gray area shown in Fig. 4.27. It can be seen that the wall thickness reduction area is nearly one third of the entire area so that it can be considered as global.

The sequence of photographs showing the 30 L CNG cylinder with different mass loss can be seen in Fig. 4.33.

0% Mass loss

0.1% Mass loss

0.2% Mass loss

0.4% Mass loss

0.6% Mass loss

0.9% Mass loss

1.3% Mass loss

1.6% Mass loss

1.9% Mass loss

2.2% Mass loss

2.6% Mass loss

2.9% Mass loss

Fig. 4.33 The photographs of 30 L CNG cylinder with different mass loss

Fig. 4.34 shows the $\pm 2^{\text{nd}}$ FSCs of the 30 L CNG cylinder. It can be observed the $\pm 2^{\text{nd}}$ natural frequencies fluctuate with respect to the angular location of accelerometer and show four peaks even for the initial condition. This is because the 30 L CNG cylinder has initial imperfection which cannot be eliminated due to fabrication process. With the increase of mass loss, the natural frequencies decrease gradually.

(a) +2nd FSCs

(b) -2nd FSCs

Fig. 4.34 The $\pm 2^{\text{nd}}$ FSCs of 30 L CNG cylinder

Fig. 4.35 The relationship between the averages of ± 2 nd natural frequencies and the mass loss for the 30 L CNG cylinder

Then, the relationship between the averages of ± 2 nd natural frequencies and the mass loss for the 30 L CNG cylinder are plotted in Fig. 4.35. It can be seen that the experimental results match the theoretical solution well, especially when the mass loss is below 2%. Because the reduction of mass is introduced on the external surface, the averages of measured ± 2 nd natural frequencies are slightly higher than the theoretical solution. It can also be found that the difference between the ± 2 nd natural frequencies almost maintain the same value, which is an important feature of global corrosion and can be used to classify the whether the corrosion is local or global.

4.5 Summary

The influence of auxiliary mass on the natural frequencies is discussed in this chapter. It is found that the auxiliary mass changes the axisymmetry of the cylinder-mass system and the split-mode phenomenon occurs, and the +2nd natural frequency shows a difference from the -2nd natural frequency. If the cylindrical shell is still axisymmetric, the natural frequencies will keep constants no matter where the auxiliary mass is located in the circumferential direction. However, if the cylindrical shell is non-axisymmetric due to reduction of wall thickness, the $\pm n$ th natural frequencies will fluctuate with respect to the angular location of the auxiliary mass resulting in $2n$ peaks and $2n$ troughs.

Therefore, the average natural frequencies are used and the relationship between the averages of ± 2 nd natural frequencies and mass loss is studied.

From the numerical examples it can be concluded from this study that the mass loss can be evaluated well for global reduction of thickness according to the averages of measured ± 2 nd natural frequencies by using the theoretical solution, which coincides with the conclusion obtained previously. Moreover, the experimental results have also validated the conclusion. But for the local reduction of wall thickness, it is still difficult to assess the mass loss according to the averages of measured ± 2 nd natural frequencies.

CHAPTER 5. DETECTION OF LOCAL REDUCTION OF WALL THICKNESS IN CIRCUMFERENTIAL DIRECTION

5.1 Introduction

The relationship between the ± 2 nd natural frequencies and the mass loss is discussed in the previous two chapters. The accelerometer is considered as auxiliary mass influencing the measured natural frequencies, and therefore, the averages of the ± 2 nd natural frequencies along the circumferential direction are adopted to eliminate these influences. For the global reduction of wall thickness, the theoretical solution obtained in Chapter 3 can be used to evaluate the mass loss according to the averages of ± 2 nd natural frequencies. However, the theoretical solution is not suitable for the local reduction of wall thickness. In fact, the evaluation of mass loss for the local reduction of wall thickness is not as important as that for the global reduction of wall thickness. Instead, the location of the wall thickness reduction is more of a concern for local damage.

In this chapter, a method to locate the local reduction of wall thickness on circumferential direction is proposed. Then, it is demonstrated by numerical examples and experimental tests. Moreover, some parameters influencing the effect of this method are also discussed.

5.2 Theoretical Background

The simplified model is illustrated in Fig. 4.3 and the natural frequencies of the interactive system containing defects are obtained from the following equation:

$$\omega_{\pm dn}^2(\theta) = \frac{\omega_{\pm drn}^2}{1 + \frac{M_r}{\rho_r A \Omega_{\pm dn}} U_{\pm dn}^2(\theta)} \quad (4.27)$$

where θ is the circumferential location of the auxiliary mass, $\omega_{\pm dm}$ are the $\pm n$ th natural frequencies of the damaged ring, $U_{\pm dn}$ are the corresponding mode shapes, and $\Omega_{\pm dn} = \int_0^{2\pi} U_{\pm dn}^2(\theta) d\theta$ are corresponding integral constants. It can be seen that $\omega_{\pm dn}^2(\theta)$ is a

function of θ , and therefore, it is named the $\pm n$ th Frequency Shift Curve (FSC) in the previous chapter. According to (Royston et al., 2000), $\omega_{\pm dm}$ and $U_{\pm dn}$ can be presented as

$$\omega_{\pm drn}^2 = \omega_{\pm rn}^2 + \Delta\omega_{\pm drn}^2 \quad (5.1)$$

$$U_{\pm dn}(\theta) = U_{\pm n}(\theta) + \Delta U_{\pm n}(\theta) \quad (5.2)$$

where $\omega_{\pm n}$ are the $\pm n$ th natural frequencies of the undamaged ring and $U_{\pm n}$ are the corresponding mode shapes, $\Delta\omega_{\pm drn}$ and $\Delta U_{\pm n}$ are perturbations of natural frequencies and mode shapes of the intact ring, respectively. The perturbations were used (Royston et al., 2000) to approximate the frequencies and mode shapes of a damaged ring because slight local damage cannot influence the dynamic characteristics very much. Recalling that $U_{+n}(\theta) = \cos[n(\theta - \varphi_{+n})]$ and $U_{-n}(\theta) = \sin[n(\theta - \varphi_{-n})]$, $\varphi_{\pm n}$ are identical and arbitrary if the ring is undamaged and perfectly axisymmetric, which makes the $\pm n$ th FSCs constants. However, $\varphi_{\pm n}$ will be determined by local damage if the ring has a defect, and therefore, the mode shapes of the damaged ring $U_{\pm dn}(\theta)$ perform like sine or cosine functions. As a result, the $\pm n$ th FSCs show $2n$ peaks and $2n$ troughs; and the peaks on the $+n$ th FSC are related to the troughs on the $-n$ th FSC. In particular, it is found that $\varphi_{+dn} = \varphi_{-dn} = \theta_d$ if local damage occurs at θ_d , which demonstrates that trough will occur at θ_d on the $+n$ th FSC.

The results of numerical examples and experimental tests provided by Royston et al. (2000) show that obvious anomaly occurs on the $+n$ th mode shape at θ_d but slight deviation can be found on the $-n$ th mode shape at θ_d . That is, the $-n$ th FCS is not so sensitive as the $+n$ th FSC to local damage, and therefore, only $+n$ th FSC is used herein to detect local damages. In fact, the $-n$ th mode shape at θ_d has value of zero, which means θ_d is a nodal point of the $-n$ th mode; therefore the $-n$ th mode is less sensitive to the damage. Furthermore, from Royston's previous work it can be found that $U_{dn}(\theta_d) = \max|U_{dn}(\theta)|$, which may be because the local stiffness decreases due to local damages and then the area is more flexible. Hence, according to Eq. (4.27), it can be concluded that the local damage usually occurs where there is trough with minimum value on the $+n$ th FSC. Moreover, the anomaly at θ_d on the $+n$ th mode shape becomes more apparent as the severity of local damage increases. Therefore, the trough where the local damage occurs

is lower than others with the increase of the severity of local damage. As a result, an index is proposed to describe the severity of local damage:

$$\Delta = t_i - \frac{\sum_{j=1, j \neq i}^4 t_j}{3} \quad (5.3)$$

where t_i is the lowest trough value and $t_{j, j \neq i}$ are the other three trough values.

Because when $n = \pm 1$, the natural frequencies are zero, and these are related to rigid body motion, so that the natural frequencies of $n = +2$ and $n = -2$ are regarded as the fundamental and second frequencies respectively, which are conventionally used for beams and plates (Zhang, 2014). Moreover, the ± 2 nd natural frequencies can be measured most easily and accurately, so only these two modes are studied.

The above theory can be extended from ring like structures to cylindrical shells if only the ring like motion with no nodal point in the longitudinal direction is considered. In fact, the mode shapes of cylindrical shell can be presented as

$$\phi_{+nq}(\theta, x) = \cos(n[\theta - \varphi_n]) Q_q(x) \quad (4.28)$$

$$\phi_{-nq}(\theta, x) = \sin(n[\theta - \varphi_{-n}]) Q_q(x) \quad (4.29)$$

where $Q_q(x)$ is an axial function to present the mode shapes and q indicates the number of nodal points, both on the longitudinal direction. In particular, the cylindrical shell has only ring-like motion if $q = 0$; in other words, $Q_{q=0}(x) = Q_0$ is a non-zero constant and the mode shape of cylindrical shell becomes $\phi_{\pm n0}(\theta, x) = U_{\pm n}(\theta)Q_0$. The other modes if $q \neq 0$ are more complex and they will not be studied herein. In fact, the first two modes of a cylindrical shell are usually related to the ring like motion, which makes this approach more convenient.

Therefore, the procedure of detecting local damage in the circumferential direction can be concluded as follows. Firstly, the cylindrical shell needs to be meshed uniformly in the circumferential direction and an auxiliary mass is mounted at one point. Tapping the cylindrical shell by a small hammer and measuring the acceleration, after which the ± 2 nd

natural frequencies can be calculated accordingly. Then, the auxiliary mass is attached on another point and the ± 2 nd natural frequencies can be measured again, and finally the ± 2 nd FSCs can be determined. “The trough with minimum value on the $+2$ nd FSC indicates the circumferential location of the damage; the difference between the lowest trough value and the values of the other three troughs is an index representing the severity of the local damage: the larger the difference, the more severe the local damage. Moreover, the difference between the averages of ± 2 nd FSCs can also describe the severity of the local damage” (Zhang et al., 2014).

5.3 Numerical examples

To validate the proposed approach of detecting local damage in the circumferential direction, several numerical examples are simulated by ABAQUS V6.9. The finite element model is illustrated in Fig. 4.4 and “the geometry and material properties are as follows: elastic modulus $E = 200$ GPa, density $\rho = 7800$ kg/m³, radius $a = 80$ mm, thickness $h = 6$ mm, length $L = 330$ mm, mass $M = 0.3$ kg and spring stiffness $k = 2 \times 10^7$ N/m” (Zhang et al., 2014). The cylinder is uniformly divided into 32 parts along circumference and 4 segments along longitudinal direction. The auxiliary mass is modeled as a mass-spring system; it does not move so that it can stay vertically but the cylindrical shell is rotated by $\pi/16$ each time. Then, the natural frequencies can be calculated, and thereafter, the FSCs can be obtained.

5.3.1 Cylindrical shells with different damages

Several artificial damages with various sizes are introduced to the cylindrical shell by reducing the wall thickness by 0.5 mm. The damaged areas are indicated by dark color in Figs. 5.1(a)–(c); they are located at $[7\pi/16, 9\pi/16]$ in the circumferential direction and $[0, L]$, $[0, L/2]$ and $[0, L/4]$ in the longitudinal direction, respectively. In particular, the auxiliary mass is located above the center of the damage in case 1; while it is located above the edge of the damage in case 2 and it is located far away from the damage in case 3.

(a)

(b)

(c)

Fig. 5.1 Finite element model of damaged cylindrical shells with different damages: (a) Case 1 (b) Case 2 (c) Case 3 (Zhang et al., 2014)

Figs. 5.2(a)–(c) shows the ± 2 nd FSCs of the three damaged cases respectively. Fluctuations due to local damage can be observed, which coincides well with the conclusion of previous chapter. In particular, the peaks of the $+2$ nd FSCs are corresponding to the troughs of the -2 nd FSCs. Moreover, it can be found that the troughs with minimum values on the $+2$ nd FSCs occur at the circumferential location of the damages.

(a)

(b)

(c)

Fig. 5.2 The ± 2 nd FSCs for different damage cases: (a) Case 1 (b) Case 2 (c) Case 3
(Zhang et al., 2014)

The $+2$ nd FSCs and -2 nd FSCs for different damaged cases are shown in Fig. 5.3 for better comparison. It can be seen clearly that the difference between the minimum trough value and the other three trough values on one $+2$ nd FSC becomes more apparent with the increase of damaged size, which are 0.11 Hz, 0.83 Hz, 1.37 Hz for damaged case 1, 2, 3 respectively. On the other hand, it can hardly be ascertained in Fig. 5.3(b) the location of the damage although the damage could occur at one of the four peaks, which verifies that the -2 nd FSC is not so sensitive as $+2$ nd FSC.

(a)

(b)

Fig. 5.3 Comparison of +2nd and -2nd FSCs for different cases: (a) the +2nd FSCs (b) the -2nd FSCs (Zhang et al., 2014)

The averages of ± 2 nd FSCs and their difference for different cases are shown in Table 5.1. Because of the damage, the averages of either +2nd or -2nd FSCs decrease. Furthermore, as the damage severity increases, the difference between the averages of

± 2 nd FSCs becomes larger; therefore, it can be regarded as an indicator of damage severity as well.

Table 5.1 (Zhang et al., 2014)

Averages of ± 2 nd FSCs for different damaged cases and undamaged case (Hz)

	+2nd	-2nd	Difference
Damaged case 1	597.603	616.351	18.748
Damaged case 2	599.675	617.820	18.145
Damaged case 3	600.696	618.697	18.001
Undamaged	601.6	619.6	18.000

5.3.2 Effects of the weight of auxiliary mass

The effects of the weight of auxiliary mass are discussed because the FSCs are produced by auxiliary mass. In fact, if the weight of auxiliary mass is so small that it can be neglected, the FSCs will be a straight line which is also the basic idea of conventional vibration testing.

(a)

(b)

Fig. 5.4 The ± 2 nd FSCs for damaged case 1 with difference auxiliary mass: (a) the +2nd FSCs (b) the -2nd FSCs (Zhang et al., 2014)

Auxiliary masses with magnitude $m = 0.01, 0.05, 0.1, 0.2, 0.3$ kg are used and the ± 2 nd FSCs for damaged case 1 are shown in Fig. 5.4. It can be observed that the peak-peak value of the +2nd FSC becomes larger and the damage severity indicator – difference between the minimum trough value and other three trough values – is more apparent with the increase of weight of auxiliary mass. Therefore, larger auxiliary mass is recommended for better performance. In fact, a normal accelerometer with magnetic base weighting usually more than 50 g is sufficient for the experiment so that only a single accelerometer is required in this approach. Moreover, heavier magnetic base can be applied to improve the results.

From Fig. 5.4(b) it can be observed that the peak-peak value of the -2nd FSC becomes larger with the increase of weight of auxiliary mass as well. However, it is still very hard to determine the location of damage through the -2nd FSC.

5.3.3 Effects of initial imperfection

Because perfectly axisymmetric cylindrical shell does not exist due to fabrication limitations, therefore, even a new cylindrical shell has a slight variation in thickness, defined as initial imperfection in this study. The effects of initial imperfection are

discussed herein and the thickness of a cylindrical shell with initial imperfection is specified as (Royston et al., 2000)

$$h = \frac{6}{1 + 0.05 \cos 2\theta} \text{ (unit: mm)} \quad (5.4)$$

Then, the three damaged cases described earlier are analyzed here in which the thickness of damaged area is reduced by 0.5 mm.

Fig. 5.5 shows the ± 2 nd FSCs for undamaged cylinder shells with initial imperfection in different damaged cases. Different from perfectly axisymmetric cylindrical shell, four periodic peaks can be found on the ± 2 nd FSCs of the undamaged cylindrical shell with initial imperfection. When the damage is introduced, the ± 2 nd FSCs change apparently; and the damage is clearly indicated by the trough with minimum value on the $+2$ nd FSC. In particular, the difference between the minimum trough value and the other three trough values is larger while the damage severity increases. Moreover, Table 5.2 shows the averages of the ± 2 nd FSCs, and it can be found that the averages of the ± 2 nd FSCs decreases and their difference increases as the damage becomes more severe.

(a)

(b)

Fig. 5.5 The ± 2 nd FSCs for undamaged cylinder shell with an initial imperfection in different damaged cases: (a) the $+2$ nd FSCs (b) the -2 nd FSCs (Zhang et al., 2014)

Table 5.2 (Zhang et al., 2014)

The averages of ± 2 nd FSCs for different damaged cases and undamaged cylindrical shell with an initial imperfection (Hz)

	+2nd	-2nd	Difference
Damaged case 1	598.702	617.075	18.373
Damaged case 2	600.223	618.276	18.053
Damaged case 3	600.978	618.963	17.985
Initial imperfection	601.948	620.001	18.053

5.4 Experimental studies

The proposed approach of detecting local damage on circumference is then verified experimentally by a steel cylindrical shell and a 40 L CNG cylinder.

5.4.1 Experimental setup and descriptions

Fig. 5.6 shows the experimental setup: an accelerometer with magnetic base (300 g) which is regarded as auxiliary mass is mounted on top of cylindrical shell to measure the vibration signal, and the sponge is used under the cylindrical shell to provide free-free boundary conditions. The cylindrical shell has a diameter $D = 169$ mm, thickness $h = 6.93$ mm and length $L = 330$ mm. There are 16 lines uniformly drawn on the cylinder and the midpoints on each line are marked clearly. Firstly, the accelerometer is attached at the midpoint and the cylinder is tapped by a small hammer. The tapping point is near the accelerometer to improve the signal-to-noise ratio. Then, the vibration acceleration is recorded by the accelerometer and it is converted from time domain to frequency domain by FFT. The natural frequencies are indicated by peaks on the frequency spectrum, and then the ± 2 nd frequencies can be extracted easily. In fact, they are the first two natural frequencies. When one measurement has been completed, the cylindrical shell is rotated by $\pi/8$ (22.5°) and the accelerometer is mounted at another point and the same procedure is repeated. Then, the ± 2 nd FSCs can be obtained after 16 measurements.

Fig. 5.6 The experimental setup used in the test (Zhang et al., 2014)

5.4.2 Experiments of steel cylindrical shells

The undamaged case which has initial imperfection and four damaged cases are tested, can be seen in Fig. 5.7. For damaged case A, the artificial damage is introduced at the area $[7\pi/16, 9\pi/16] \times [3L/5, 9L/10]$ by reducing the thickness by 1.5 mm; for damaged

case B, a new damage is located at the area $[7\pi/16, 9\pi/16] \times [L/10, 2L/5]$ by reducing the thickness by 1.5 mm; for damaged case C, the damage at the area $[7\pi/16, 9\pi/16] \times [L/10, 2L/5]$ becomes more severe by reducing the thickness to 4 mm; for damaged case D, the thickness at the area between the two damages is reduced by 1.5mm to link the two separate damages together (please refer to Figs. 5.7 and 5.8).

Fig. 5.7 The profile of damaged cases: (a) Damaged case A (b) Damaged case B (c) Damaged case C (d) Damaged case D (Zhang et al., 2014)

Fig. 5.8 Four damaged cases: (a) Damaged case A (b) Damaged case B (c) Damaged case C (d) Damaged case D (Zhang et al., 2014)

Fig. 5.9 shows the measured ± 2 nd FSCs for all the tested cases, including the undamaged case and four damaged cases. The results coincide with the previous numerical examples very well. For the undamaged case, there are four periodic peaks on the ± 2 nd FSCs due to initial imperfection. When the artificial damage is introduced by grinding manually, the phases of ± 2 nd FSCs shift and the trough with minimum value on the $+2$ nd FSC occurs where the damage is located. As the damage severity becomes higher, the damage severity indicator, the difference between the trough with minimum value and other three troughs, becomes more apparent.

(a)

(b)

Fig. 5.9 The ± 2 nd FSCs for undamaged cylindrical shell with an initial imperfection and different damaged cases: (a) the +2nd FSCs (b) the -2nd FSCs (Zhang et al., 2014)

Moreover, Table 5.3 represents the averages of ± 2 nd FSCs for different cases. The same conclusion can be found: the difference between the averages of the ± 2 nd FSCs is more apparent with the increase of the damage severity and the averages of ± 2 nd FSCs also decrease as the damage is more severe.

Table 5.3 (Zhang et al., 2014)

The averages of ± 2 nd FSCs for undamaged cylinder shell with an initial imperfection and different damaged cases (Hz)

	+2nd	-2nd	Difference
Initial imperfection	693.965	703.220	9.255
Damaged case A	687.464	700.911	13.447
Damaged case B	681.141	700.016	18.875
Damaged case C	672.760	699.272	26.512
Damaged case D	658.141	698.943	40.802

5.4.3 Experiments of 40 L CNG cylinder

The approach of detecting circumferential location of damage is also applied on a 40 L CNG cylinder shown in Fig. 5.10. The CNG cylinder is divided into 16 parts in the circumferential direction and an accelerometer is mounted vertically at one point marked on the tank. After tapping on the cylinder and measuring the natural frequency, the cylinder is rotated by 22.5° and the accelerometer is then mounted at another point.

Fig. 5.10 The undamaged 40L CNG cylinder

Firstly, the ± 2 nd frequencies of the undamaged 40 L CNG cylinder are measured. Then, artificial reduction of wall thickness is introduced. The wall thickness is reduced from the external surface and it is introduced on the area shown in Fig. 5.11. The artificial damage is strip-like and it could simulate the line corrosion. The photographs of the 40 L CNG cylinder with different mass loss can be seen in Fig. 5.11.

0% Mass loss

0.06% Mass loss

0.17% Mass loss

0.23% Mass loss

0.45% Mass loss

0.73% Mass loss

0.84% Mass loss

1.13% Mass loss

Fig. 5.11 The photographs of 40 L CNG cylinder with different mass loss

Fig. 5.12 shows the measured ± 2 nd FSCs for all scenarios. It can be observed that the ± 2 nd FSCs fluctuate with the circumferential location of accelerometer and show periodic peaks and troughs. The trough with minimum value on ± 2 nd FSCs occurs where the damage is located. As the damage becomes more severe, the difference between the trough with minimum value and the other three becomes more apparent.

(a) Natural frequency ($n=2$)

(b) Natural frequency ($n=-2$)

Fig. 5.12 The ± 2 nd FSCs of 40 L CNG cylinder with different damaged cases: (a) the +2nd FSCs (b) the -2nd FSCs

Fig. 5.13 The relationship between the averages of ± 2 nd natural frequencies and the mass loss for the 40 L CNG cylinder

Furthermore, it can be seen from Fig. 5.13 that the difference between the natural frequencies ($n=+2$ & $n=-2$) increases with the mass loss, which is an important feature of local corrosion.

5.5 Summary

A novel damage detection method based on the natural frequencies is proposed to locate the wall thickness reduction in the circumferential direction. The theoretical formulas have been verified by both numerical simulation and experimental tests. It can be concluded that when there is damage on the cylindrical shell or CNG cylinders, periodic fluctuations corresponding to mode shapes occur. For the +2nd FSC which is the easiest to be measured, the trough with the minimum value indicates the circumferential location of the damage. The initial imperfection has an influence but it can be eliminated by providing the baseline. Moreover, the difference between the natural frequencies ($n=+2$ & $n=-2$) increases with the mass loss, which can be used for classifying the corrosion whether it is local or global.

CHAPTER 6. FREQUENCY SHIFT SURFACE BASED DAMAGE INDEX

6.1 Introduction

The local reduction of wall thickness can be detected in the circumferential direction by +2nd FSC, which has been demonstrated in the previous chapter. In this chapter, the auxiliary mass is used again for detection of local damage in longitudinal direction. Moreover, a new damage index is proposed to accurately locate the local damage, which has been verified by both numerical examples and experimental results.

6.2 Auxiliary mass induced frequency shift in longitudinal direction

A numerical example of a cylindrical shell with a single damage is shown in Fig. 6.1. The properties of the cylinder are as follows: length $L = 330$ mm, radius $r = 160$ mm, thickness $h = 6$ mm, elastic modulus $E = 2 \times 10^{11}$ N/m² and density $\rho = 7800$ kg/m³. Moreover, the spring has stiffness of 1×10^7 N/m and the auxiliary mass is of 0.2 kg. The single artificial damage is introduced at $[7\pi/16, \pi/2] \times [5L/16, 7L/16]$ and the thickness at damage area is reduced by 1 mm. The cylinder is uniformly meshed into 32×14 segments (from $L/16$ to $15L/16$ in longitudinal direction).

Fig. 6.1 Numerical example of a cylindrical shell with a single damage

The auxiliary mass is attached on top of the cylinder, and it passes through three lines, l_1 , l_2 and l_3 respectively. l_1 passes through the damage area while l_2 and l_3 are near the damage. Then, the auxiliary mass induced FSC in longitudinal direction is illustrated in

Fig. 6.2 and only the +2nd (fundamental) mode is considered for convenience. It can be seen that they have quite similar shapes. Although they are all smooth, small anomaly on the FSC of l_1 can also be observed. In fact, the anomaly is a drop on the FSC, which may be because the local damage reduces the local stiffness.

Fig. 6.2 The +2nd FSC of l_1 , l_2 and l_3

Fig. 6.3 The curvatures of frequency curves of l_1 , l_2 and l_3

Then, the curvature used to enhance the effect of anomaly is calculated accordingly. Fig. 6.3 shows the curvatures of FSC of l_1 , l_2 and l_3 respectively, and they are quite similar except an obvious anomaly is found on the curvature of FSC of l_1 . In fact, the apparent peak is just at the location of the artificial damage, therefore, the anomaly on the curvature of FSC can be considered as an indicator of local damage.

As a result, after the circumferential direction of the local damage is determined, the curvature of FSC can be used to indicate the longitudinal location of local damage.

Because the curvature of FSC is important in damage detection, it should as accurate as possible in experiments. Although noise is one of the principal causes of inefficacy of most of the methods based on curvature that have been proposed in literature, it shows very limit influence in this method. This is because the FSC is based on the measurement of natural frequency, which is the least vulnerable to noise in dynamic characteristics. However, the resolution of measured frequency is crucial in this method because it directly determines the accuracy of curvature of FSC; therefore, higher resolution of frequency is expected.

6.3 Damage index based on auxiliary mass induced Frequency Shift Surface

The local damage can be located in the circumferential direction first and then determined in the longitudinal direction. In theory, the Frequency Shift Curve (FSC) can be extended to Frequency Shift Surface (FSS). Recalling that the FSC on circumferential direction is defined in Chapter 4 as

$$\omega_{\pm dn}^2(\theta_0) = \frac{\omega_{\pm drn}^2}{1 + \frac{M_r}{\rho_r A \Omega_{\pm dn}} U_{\pm dn}^2(\theta_0)} \quad (4.27)$$

It can be extended from circumferential direction to the entire surface, and therefore, the Frequency Shift Surface (FSS) can be defined as

$$\omega_{\pm nq}^2(\theta_0, l_0) = \frac{\omega_{\pm c,nq}^2(\theta_0, l_0)}{1 + \frac{M}{\rho h L \Omega_{\pm n,q}} \varphi_{\pm nq}^2(\theta_0, l_0)} \quad (6.1)$$

where $\varphi_{\pm nq}$ are mode shapes of the cylindrical shell, n and q indicate the number of nodal points in the circumferential and longitudinal directions respectively, and

$$\Omega_{\pm n,q} = \int_0^L \int_0^{2\pi} \varphi_{\pm nq}^2(\theta, l) d\theta dl \quad (6.2)$$

Based on FSS, a new damage index is proposed and it uses only the information from the damaged cylinder. Because the local damage at early stage is usually small, therefore, most curvatures of FSCs are smooth and more importantly they have almost the same shape. Therefore, the average of all curvatures of FSCs is expected to be smooth; it can be regarded as “baseline” or reference so that the prior information from undamaged cylinder can be avoided. Finally, the damage index can be proposed as the comparison between the “baseline” and curvature of each FSC.

In particular, if a cylindrical shell is uniformly meshed into $M \times N$ segments in the circumferential and longitudinal directions, there will be M longitudinal FSCs and $N+1$ points on each FSC. Only the +2nd FSC is used herein because the +2nd frequency of the cylinder is actually the fundamental frequency and it can be measured more easily and accurately. The +2nd frequency of the cylinder is presented as $\omega(\theta, l)$ in this chapter and the +2nd frequency on each cross point can be similarly represented as $\omega(\theta_i, l_j)$, $i=1, \dots, M$, $j=1, \dots, N+1$. Therefore, the curvature of each FSC can be calculated by central difference:

$$\omega''(\theta_i, l_j) = \frac{\omega(\theta_i, l_{j-1}) - 2\omega(\theta_i, l_j) + \omega(\theta_i, l_{j+1}))}{(\Delta l)^2}, j = 2, \dots, N - 1 \quad (6.3)$$

Then, the average of all curvatures of FSCs can be obtained as

$$M_\omega(l_j) = \sum_{i=1}^M \omega''(\theta_i, l_j) / M \quad (6.4)$$

Finally, the damage index can be represented as

$$D(\theta_i, l_j) = \left(\omega''(\theta_i, l_j) - M_\omega(l_j) \right)^2 \quad (6.5)$$

The FSS of the finite element model used previously is shown in Fig. 6.4. It is found that in the longitudinal direction, all frequency shift curves have similar shapes and in the circumferential direction, four periodic fluctuations lay as expected. In fact, only the fundamental frequency is used as it can be measured more easily and accurately.

Fig. 6.4 Fundamental FSS of cylindrical shell with a single damage

Fig. 6.5 Damage index of cylindrical shell with a single damage

The proposed damage index of this cylindrical shell with a single damage is shown in Fig. 6.5: a peak indicating this single damage can be seen clearly. Moreover, the damage area can be determined from the right side of Fig. 6.5 as well, which is the contour of the damage index.

For better illustration, another numerical example in which the cylindrical shell has multiple damages is adopted. The FE model is similar to that containing a single damage, only except that another damage is introduced at $[9\pi/8, 19\pi/16] \times [5L/8, 3L/4]$ by reducing the thickness to 5.5 mm. These two damages are marked as “Damage A” and “Damage B” respectively, which can be seen in Fig. 6.6.

Fig. 6.6 Numerical example of a cylindrical shell with multiple damages

The inspection procedure is the same, and therefore, the fundamental FSS can be obtained when the auxiliary mass is probing over the entire surface. Fig. 6.7 shows the fundamental FSS of the cylinder with multiple damages: it appears almost the same as that of the cylinder with only a single damage and no apparent feature corresponding to the damages can be seen. Then, the proposed damage index is calculated accordingly and shown in Fig. 6.8. Two sharp peaks related to these two local damages can be seen clearly and the one indicating “Damage A” is higher than that indicating “Damage B”. This is because although the two damages have the same size, “Damage B” is only half deep as “Damage A”; and therefore, “Damage A” is much more severe than “Damage B”. Furthermore, from the contour of the damage index (right side of Fig. 6.8) it can be seen that the areas of the two peaks are almost the same, which coincides with the FE model well.

Fig. 6.7 Fundamental FSS of cylindrical shell with multiple damages

Fig. 6.8 Damage index of cylindrical shell with multiple damages

6.4 Experimental verification of damage index

Several experiments are conducted herein to validate the proposed method. Firstly, an artificial damage is introduced on the external surface of a steel cylinder by reducing the wall thickness, and then, another artificial damage is introduced on other position to simulate the scenario of multiple damages.

6.4.1 Experimental setup

The experimental setup shown in Fig 6.9 is similar to those described earlier in the previous chapters. Firstly, the specimen is meshed into uniform segments and each cross point needs to be labelled clearly. Then, the specimen is supported by sponges to provide free-free boundary condition. A single accelerometer (Wilcoxon Model No. F3/Z602WA) which is 0.3 kg in weight is mounted at a cross point with the help of magnetic base. Because the accelerometer is heavy enough, therefore, no other auxiliary mass is required. When the specimen is tapped by a small hammer, it vibrates and the acceleration is recorded accordingly. The electrical signal is then amplified through a charge amplifier (Wilcoxon Model No. PA8HF) and sampled at sampling frequency of 2048Hz by data acquisition card (Measurement Computing Model No. USB 2527). Thereafter, the acceleration is transferred into frequency domain by Fast Fourier Transform (FFT), and then the fundamental frequency can be obtained. After one test is completed, the accelerometer is moved to another cross point. When the accelerometer is probing the entire surface the fundamental FSS can be obtained.

Fig. 6.9 The set-up of experimental tests

6.4.2 Steel cylinder with a single damage

A steel cylinder having diameter of 169 mm, thickness of 6.93 mm and length of 420 mm is utilized herein and an artificial damage is introduced at $[7\pi/16, 9\pi/16] \times [120 \text{ mm}, 180$

mm] by reducing the thickness by 2.5 mm, as shown in Fig. 6.10. Moreover, the cylinder is meshed uniformly and 13×16 cross points can be obtained.

Fig. 6.10 The steel cylinder with a single damage

The fundamental FSS can be easily obtained by following the procedure described above, and it is shown in Fig. 6.11. It looks like those in the numerical examples but the single damage can be seen directly from the contour of the FSS. The frequency drops drastically when the auxiliary mass is close to the damage. This is because the local damage reduces the local stiffness, and at the same time, the auxiliary mass increases the local inertia. Fig. 6.12 shows the damage index from which an apparent peak related to the local damage can be found. Moreover, the size of the damage can also be assessed according to the contour of the damage index.

Fig. 6.11. The fundamental FSS of the steel cylinder with a single damage

Fig. 6.12 Damage index of steel cylinder with a single damage

6.4.3 Steel cylinder with multiple damages

Another artificial damage is introduced at $[19\pi/16, 21\pi/16] \times [300 \text{ mm}, 240 \text{ mm}]$ on the same steel cylinder by reducing thickness by 0.5 mm. To distinguish these two damages, they are labelled as “Damage A” and “Damage B” respectively, which can be seen in Fig. 6.13. They have almost the same size, but “Damage A” is five times deeper than “Damage B”.

Fig. 6.13. The steel cylinder with multiple damages

Fig. 6.14 shows the fundamental FSS of the cylinder containing multiple damages, which is quite similar to that with only single damage. “Damage A” is much more severe, so it can be indicated directly by the sudden drop on the fundamental FSS. However, “Damage B” is not so obvious and it can hardly be seen from Fig. 6.14.

Fig. 6.14 The fundamental FSS of the steel cylinder with multiple damages

The corresponding damage index is presented in Fig. 6.15 and two peaks related to these two damages can be seen clearly. In particular, the peak indicating “Damage A” is much higher than that describing “Damage B”, which verifies that this damage index can evaluate the severity of the damage as well. Similarly, the sizes of these damages can be determined through the contour of the damage index.

Fig. 6.15. Damage index of steel cylinder with multiple damages

6.5 Summary

In this chapter, the Frequency Shift Curve (FSC) induced by auxiliary mass is extended from circumferential direction to longitudinal direction, and it can be found that the

curvature of FSC in longitudinal direction is sensitive to local damage. Furthermore, FSC is extended to Frequency Shift Surface (FSS) on the entire surface. In fact, the natural frequency drops suddenly and significantly as the auxiliary mass is approaching the local damage. This is because the local damage reduces the local stiffness and the auxiliary mass increases the local inertia. In other word, the auxiliary mass is capable to amplify the effect of local damage on the frequency, and therefore, it can be used to probe the local damage.

In addition, a novel damage index is proposed, based on the assumption that the FSS is usually smooth and the sudden drop on FSS induced by the auxiliary mass approaching local damage is regarded as anomaly. Moreover, because of the periodicity of FSS in circumferential direction and similarity of FSS in longitudinal direction, the proposed damage index does not require any prior information from undamaged structures.

Finally, the proposed damage index shows good performance in all the numerical examples and experimental tests. Because the curvature of FSC is important in damage detection, it should be as accurate as possible in the experimental results. Although noise is one of the principal causes of inefficacy of many methods based on curvature that have been proposed in literature, it shows very limit influence in this method. This is because the FSC is based on the measurement of natural frequency, which is less vulnerable to noise in dynamic characteristics. However, the resolution of measured frequency is crucial in this method because it determines directly the accuracy of curvature of FSC; therefore, higher resolution of frequency is expected.

CHAPTER 7. STUDY OF INSPECTION OF CRACKS ON CYLINDRICAL SHELL STRUCTURES

7.1 Introduction

Although a crack seldom occurs like corrosion in CNG cylinders, it is still quite dangerous as it may cause catastrophic failure; therefore, it is very important to discover the damage due to crack at the early stage. The approach and damage index proposed in the previous chapter is used herein to study the inspection of crack on cylindrical shell structures. Moreover, several numerical examples and experimental studies are used to validate the proposed method and damage index.

7.2 Numerical examples

A free-free cylindrical shell with various damage scenarios is studied herein to validate the proposed method and damage index. All numerical examples are simulated by ABAQUS, V6.11.

7.2.1 Cylindrical shell with a single crack on internal surface

The FE model of the cylindrical shell with free-free boundary condition is shown in Fig. 7.1. It has length of 400 mm, radius of 103 mm and thickness of 6 mm, and the material properties are as follows: Young's modulus $E = 200\text{GPa}$, density $\rho = 7800\text{kg/m}^3$, Poisson's ratio $\nu = 0.3$. It is meshed uniformly into 32×16 elements, and therefore, there are total 32×17 cross points (Fig. 7.2). The auxiliary mass is simulated as a spring-mass system with mass of 0.2 kg and stiffness of $2 \times 10^7 \text{ N/m}$ and it is mounted at the cross point. For simplicity, the damping is ignored.

Fig. 7.1 FE model of auxiliary mass - cylindrical shell interactive system

Moreover, a single slot with width of 5 mm, depth of 5 mm and length of 50 mm is used to simulate the crack, and its center is located at $(\pi/2, 5L/8)$ which can be seen in Fig. 7.2 clearly.

Fig. 7.2 The specifications of a single crack on internal surface

The auxiliary mass is probing the entire surface, and then the fundamental FSS can be easily obtained. The procedure is the same as that described in Chapter 6. Fig. 7.3 shows the fundamental FSS and four periodic peaks in the circumferential direction can be seen as expected. Because the slot is quite small, therefore, the fundamental FSS is smooth and no anomaly related to the slot can be found.

Fig. 7.3 The fundamental FSS of the cylindrical shell with a single crack on internal surface

Fig. 7.4 The damage index of the cylindrical shell with a single crack on internal surface

Fig. 7.4 shows the damage index of the cylindrical shell mapped to the 2-D coordinate and the sharp peak reveals the location of the slot. Moreover, if the mesh density is higher enough, the area of the slot can be evaluated as well.

7.2.2 Cylindrical shell with a single crack on external surface

In this case, the slot with the same dimensions and location is introduced on the external surface, which is illustrated in Fig. 7.5.

Fig. 7.5 The specifications of a single crack on external surface

The fundamental FSS of this case is shown in Fig. 7.6. Although it is quite similar to that of the previous case, however, the anomaly related to the slot on the external surface can be seen directly from the contour of the fundamental FSS. Fig. 7.7 presents the corresponding damage indices mapped to the 2-D coordinate and the peak indicating the slot is obvious. Moreover, it can be found the peak value in this case is much higher than that in the previous peak, which indicates that the slot on the external surface is more severe than that on the internal surface. In fact, this is because the reduction of thickness from the internal surface apparently induces a lower reduction of local stiffness of the cylinder with respect to the reduction of thickness from the external surface.

Fig. 7.6 The fundamental FSS of the cylindrical shell with a single crack on external surface

Fig. 7.7 The damage index of the cylindrical shell with a single crack on external surface

7.2.3 Cylindrical shell with multiple cracks on internal surface

In this scenario, two slots are introduced on the internal surface, which can be seen in Fig. 7.8. “Slot A” having width of 5 mm, depth of 4 mm and length of 75 mm is located at $(19\pi/16, 13L/32)$; “Slot B” having width of 5 mm, depth of 5 mm and length of 50 mm is located at $(\pi/2, 5L/8)$.

Fig. 7.8 The specifications of multiple cracks on internal surface

Fig. 7.9 presents the fundamental FSS of the cylindrical shell with two cracks on the internal surface. It looks quite similar to those in the previous cases, however, the trough

with minimum value occurs at $19\pi/16$ in the circumferential direction, where “Slot A” locates. This may be because “Slot A” is 50% longer than “Slot B”, therefore, it is considered more severe.

Fig. 7.9 The fundamental FSS of the cylindrical shell with multiple cracks on internal surface

Fig. 7.10 The damage index of the cylindrical shell with multiple cracks on internal surface

The corresponding damage index is shown in Fig. 7.10. Two peaks that relate to “Slot A” and “Slot B” respectively are clearly seen, and the peak indicating “Slot B” is higher than that indicating “Slot A”. This is because the damage index is based on curvature, which is more sensitive to damage with small area.

7.2.4 Cylindrical shell with multiple cracks on external surface

In this scenario, two slots with the same dimensions and locations as those in the previous case are introduced on the external surface, which can be seen in Fig. 7.11.

Fig. 7.11 The specifications of multiple cracks on external surface

The fundamental FSS of the cylindrical shell with multiple cracks on external surface is shown in Fig. 7.12, which is similar to that in the previous scenario. However, two anomalies related to these two cracks can be seen directly from the contour of the FSS, which may be because the slots on external surface are more severe than those on internal surface.

Fig. 7.13 shows the damage indices mapped to 2-D coordinate, and the two peaks indicating these two slots are apparent. The peak indicating “Slot A” is lower than that related to “Slot B”. However, both of the two peak values are higher than those of cylindrical shell with two cracks on internal surface, which also means that this damage index is more sensitive to external cracks.

Fig. 7.12 The fundamental FSS of the cylindrical shell with multiple cracks on external surface

Fig. 7.13 The damage index of the cylindrical shell with multiple cracks on external surface

7.3 Experimental verification

Two simple experiments are carried out to verify the proposed method and damage index. Firstly, a steel cylinder is used and an artificial slot is introduced; then a 7 L CNG cylinder with a sharp slot is tested. The experiment setup is the same as that in Chapter 6.

7.3.1 Steel cylinder with a single slot

Fig. 7.14 shows the photograph of a steel cylindrical shell, which has diameter of 169 mm, thickness of 6.93 mm and length of 330 mm. It is supported by sponges to simulate a free-free boundary condition, and it is meshed uniformly into 16×11 elements (Fig. 7.15). The accelerometer used in Chapter 6 is placed on top of the cylindrical shell and it is regarded as the auxiliary mass itself. A slot which has depth of 5 mm and length of 80 mm is introduced on the external surface.

Fig. 7.14 The steel cylinder with a single slot

Fig. 7.15 The specifications of artificial slot on external surface of cylinder

The experimental procedure is exactly the same as that described in Chapter 6. When the accelerometer is probing the entire surface of the cylinder, the fundamental FSS and corresponding damage index can be obtained accordingly.

Fig. 7.16 The fundamental FSS of the cylindrical shell with artificial slot on external surface

Fig. 7.17 The damage index of the cylindrical shell with artificial slot on external surface

Fig. 7.16 shows the fundamental FSS of the cylinder with a single slot. It is not so smooth as compared to those in numerical examples because of measurement error and sparse grids. However, the anomaly related to the slot can be seen clearly without applying the damage index. The frequency drops drastically when the accelerometer is closed to the slot; it is because not only the slot decreases the stiffness but the accelerometer also increases the local inertia.

The corresponding damage index is shown in Fig. 7.17 and the peak revealing the slot is quite sharp. Moreover, the size of the slot can also be assessed according to the contour of the damage index.

7.3.2 CNG cylinder with artificial crack

A 7 L CNG cylinder with an artificial crack is tested herein. The dimensions of the cylindrical shell part of the CNG cylinder are as follows: external diameter $D = 170$ mm, thickness $h = 6$ mm and length $L = 320$ mm. It has been meshed uniformly into 16×10 elements, and therefore, there are 16×11 cross points.

A slot with width of 1 mm and depth of 5 mm is cut using an EDM machine to simulate the crack. For generality, the slot is located at $(7\pi/8 + \pi/40) \times [80 \text{ mm}, 240 \text{ mm}]$ which can be seen in Fig. 7.18, and therefore, no cross point is located on the slot. The cylindrical shell part of this CNG cylinder vibrates slightly different from the cylinder with fixed boundary condition; it is because the two ends of the CNG cylinder are much thicker, and hence, they provide a fixed end boundary condition to the cylindrical shell part. In fact, when the cylindrical shell vibrates freely, the two ends are almost stationary.

Fig. 7.18 The CNG cylinder containing a crack damage

The fundamental FSS of this 7 L CNG cylinder is then measured, which is presented in Fig. 7.19. It looks different from other cases because of the fixed end boundary condition: the frequency is higher at the boundary while it is lower at the middle which is just opposite to other cases with free-free boundary condition. In fact, the local stiffness at the boundary area is higher due to the fixed end boundary condition, while the local stiffness at the middle area is much lower. In addition, the crack reduces the local stiffness so drastically that the frequency drops apparently when the auxiliary mass is approaching the crack; therefore, the crack can be found directly from the fundamental FSS. The corresponding damage index is shown in Fig. 7.20 as well. The crack is indicated clearly and the length can be evaluated accordingly.

Fig. 7.19 The fundamental FSS of the CNG cylinder containing a crack damage

Fig. 7.20 Damage index of the CNG cylinder containing a crack damage

7.4 Summary

In this chapter, several numerical examples and experiments are studied to validate the effect of proposed method and damage index on damage of cracks; both numerical examples and experimental studies show good performance. Not only the locations of cracks can be determined according to the damage index, but the lengths of cracks can also be evaluated. Especially, a 7 L CNG cylinder with a single crack is tested herein. Although the boundary condition of the cylindrical shell part of the CNG cylinder is considered as fixed end which is different from the free-free boundary condition, the fundamental FSS and the damage index can give accurate result of inspection of crack.

This is because the fundamental vibration mode of the cylindrical shell part is the ring like mode and there is no nodal point in the longitudinal direction.

CHAPTER 8. CONCLUSIONS AND FUTURE WORKS

8.1 Summary of the main findings

The main objective of this research work is to investigate the influence of corrosion and crack on the vibration characteristics of CNG cylinders and general cylindrical shells, and develop a quick and effective technique to locate and evaluate the corrosion at the internal surface of CNG cylinders. The main findings concluded in this dissertation are summarized as follows:

(1) The corrosion reduces the wall thickness of CNG cylinders resulting in the decrease of the natural frequencies. Different types of corrosion, for example, line corrosion and general corrosion have different influence on the natural frequencies of CNG cylinders. In particular, the severity of general corrosion can be evaluated according to the fundamental natural frequency.

(2) As the weight of the accelerometer has influence on the natural frequencies of CNG cylinders, it cannot be ignored and it is regarded as auxiliary mass in this research. Generally, the natural frequencies decrease slightly due to an increase of mass. Moreover, because of the accelerometer, the cylindrical shell is never axisymmetric. Hence, the natural frequencies change with respect to the location of the accelerometer, and the averages of measured frequencies at different location are adopted to approximate the natural frequencies. Furthermore, split mode phenomenon occurs once the cylindrical shell is never axisymmetric. It is especially so for the first two (± 2 nd) identical natural frequencies; if the cylindrical shell is not axisymmetric then they become different. For general corrosion having a large area, the difference is nearly constant as the severity of corrosion increases. However, for line corrosion having only small strip area, the difference rises with the increase of severity of corrosion. Therefore, the difference between the first two natural frequencies can be used to classify the type of corrosion.

(3) The natural frequency of CNG cylinder changes with respect to the location of auxiliary mass. When the auxiliary mass is probing along a line, the natural frequency is a function of the location of auxiliary mass, which is defined as Frequency Shift Curve (FSC) in this study. In particular, the fundamental (± 2 nd) FSC in the circumferential

direction shows four periodic peaks and troughs, and it is found that the trough with minimum value indicates the circumferential location of the local corrosion. Furthermore, the curvature of FSC in the longitudinal direction is capable of indicating the longitudinal direction of local corrosion.

(4) The idea of FSC is then extended to the entire surface, and the Frequency Shift Surface (FSS) is defined accordingly. Moreover, a novel damage index based on FSS is proposed and it does not require prior information from undamaged structure. Not only a single local corrosion, but multiple defects of local corrosion can also be identified accurately. Moreover, the size and severity of local corrosion can be evaluated as well.

(5) The damage due to a crack on a cylindrical shell is studied as well. The cracks, on either internal or external surface, can be inspected accurately by the proposed method and the damage index. In fact, the natural frequency of the cylindrical shell drops drastically when the auxiliary mass is approaching the crack; it is because the auxiliary mass increases the local inertia while the crack reduces the local stiffness.

8.2 Original contributions

The original contributions of this study are summarized as follows:

(1) The relationship between corrosion and natural frequencies of cylindrical shell is investigated for the first time. It provides fundamental basis for judging the type of corrosion and assessing the severity of general corrosion.

(2) The influence of auxiliary mass on the vibration characteristics of cylindrical shell is studied first, based on which the idea of FSC is proposed. The circumferential location of local corrosion can be easily determined through the fundamental (+2nd) circumferential FSC.

(3) The curvature of fundamental (+2nd) longitudinal FSC is investigated and anomaly would occur at the location of local corrosion. Then, the longitudinal location of local corrosion can be determined accordingly.

(4) The FSS is defined first, based on which a novel damage index is proposed. It does not require prior information from undamaged structure. Assuming that the undamaged

structure is smooth, the locations, sizes and severities of local corrosions can be evaluated by using this damage index.

(5) Inspection of crack on cylindrical shell is investigated. It reveals that the proposed damage index is sensitive to crack. In fact, the natural frequency of the cylindrical shell drops significantly when the auxiliary mass is approaching the crack; it is because the auxiliary mass increases the local inertia while the crack reduces the local stiffness.

8.3 Future works

This study has accomplished the research scopes and objectives. However, several research gaps are also found and preliminary ideas for the future work are suggested:

(1) Through the fundamental circumferential FSC, the circumferential location of local corrosion can be determined, and then the longitudinal direction can be indicated by the curvature of fundamental longitudinal FSC. This quick inspection method is effective for a single corrosion. When multiple corrosions occur, it is effective for the most severe corrosion; for other less severe corrosions, the FSS and corresponding damage index are required. However, it is somehow time-consuming, and therefore, how to improve this method for multiple corrosions should be investigated further.

(2) Although FSS and the corresponding damage index show good performance for inspection of multiple damages, however, the mesh should be sufficiently refined to determine the sizes of damages accurately, making the job inconvenient and time consuming. Hence, optimization of the mesh and grid is an important issue needed to be researched further.

(3) This method is specifically proposed for Type-1 all metal CNG cylinder. Other types of CNG cylinders such as composite CNG cylinder have also been used widely; therefore, extending this technique to other types of CNG cylinders should be carried out for future works.

(4) This method is effective for general cylindrical shells like CNG cylinders. However, its capability should be extended to other important large structures such as pressure vessels and oil storage tanks.

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