EXPERIMENTAL AND NUMERICAL STUDY ON
PROPAGATION BUCKLING IN SUBSEA PIPE-IN-PIPE SYSTEMS

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ABSTRACT

Subsea pipe-in-pipe (PIP) systems are being used extensively in the design of high-pressure and high-temperature (HP/HT) flowlines because of their outstanding thermal insulation performance. A typical PIP system consists of concentric inner and outer pipes, bulkheads, and centralizers. The inner pipe (flowline) conveys production fluids, and the outer pipe (carrier pipe) protects the system from high external pressure (water depths up to 3000 m) and mechanical damage. These two pipes are isolated by centralizers at joints and connected through bulkheads at both ends of the pipeline. The annulus (the space between the tubes) is either empty or filled with non-structural insulation material such as foam or water.

The external pressure in the vicinity of local dents or ovality in the outer pipe-wall can cause a local collapse, which may catastrophically propagate along with the PIP system, known as propagation buckling. The lowest pressure required to sustain such a buckle propagation is known as the propagation pressure $P_p$, which is only 15–30% of the initiation pressure $P_i$ of the intact pipe. The propagation buckling (or buckle propagation) has been extensively investigated in single pipelines using analytical, experimental and numerical methods.

Unlike single pipelines, buckle propagation and collapse mechanisms in PIPs have only been marginally addressed. Therefore, this research program aims to investigate propagation buckling of subsea pipe-in-pipe (PIP) systems under hydrostatic pressure; experimentally and numerically. By using experimental results, a finite element (FE) model is validated and is used to conduct a comprehensive parametric study to evaluate the effect of geometric and material parameters affecting the propagation buckling pressure of PIP system. The experimental study performed herein showed that the buckling mechanisms and collapse pressure of the inner pipe of a PIP system are different from those of a single pipeline. Therefore, a chapter of the thesis is dedicated to the investigation of the collapse mechanisms of the inner pipe of a PIP system. Moreover, a parametric study is conducted and the collapse mechanisms of PIPs with various combinations of outer and inner pipes in a broad practical range of diameter-to-thickness ratios ($D/t$) are evaluated. The thesis is concluded by proposing and investigating the appropriateness of a novel carbon fibre buckle arrestor.
The thesis makes the following contributions to the knowledge and practice in the area of offshore pipelines. A more accurate simplified analytical expression is proposed to predict the propagation buckling pressure of PIP system. The buckle propagation modes of PIPs are identified and discussed. Subsequently, empirical expressions for each mode are proposed to predict the propagation buckling pressure of PIP systems. Moreover, a parametric study is conducted on the collapse mechanisms of the inner pipe of a PIP system. Comprehensive empirical expressions are proposed for the collapse pressure of the inner pipe, and the upper and lower pressure capacity bounds are quantified. Design charts are proposed to estimate the collapse pressure of the inner pipe and its corresponding collapse modes. In the last part of this research, CFRP buckle arrestors are proposed for subsea pipelines and are shown to achieve higher arresting efficiency compared with the conventional buckle arrestors.
STATEMENT OF ORIGINALITY

This work has not previously been submitted for a degree or diploma in any university. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made in the thesis itself.

(Signed) _________________________________

Date: 23.01.2020

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ACKNOWLEDGEMENT OF PUBLISHED PAPERS

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Listed below are co-authored publications resulted from this study and are presented in this thesis as Chapters 3, 4, 5, and 6. My contribution to each co-authored publication is outlined in the beginning of the relevant chapter. The bibliographic details of the published/submitted papers are given below:

Chapter 3:


Chapter 4:


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LIST OF PUBLICATIONS

The bibliography of all peer reviewed publications is presented below to disseminate the methodology and outcomes of this research:

Book Chapters:


Journal Papers:


Conference Papers:


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NOTATIONS

$D_o$ outer pipe diameter
$D_i$ inner pipe diameter
$D$ nominal outside diameter
$D_s$ solid inner pipe diameter
$t_o$ wall thickness of the outer pipe
$t_i$ wall thickness of the inner pipe
$h, h_a$ thickness of the buckle arrestor
$h_{cm}$ minimum thickness of the buckle arrestor
$L, L_a$ length of the buckle arrestor
$\eta$ buckle arrestor efficiency
$\eta_{CFRP}$ CFRP arrestor efficiency
$\eta_{UC}$ upper bound of buckle arrestor efficiency
$\eta_{PC}$ lower bound of buckle arrestor efficiency
$\theta$ orientation of the CFRP arrestor
$E_o$ modulus of elasticity of the outer pipe
$E_i$ modulus of elasticity of the inner pipe
$E'_o$ tangent modulus of the outer pipe
$E'_i$ tangent modulus of the inner pipe
$\sigma_Y$ yield stress
$\sigma_{Yo}$ yield stress of the outer pipe
$\sigma_{Yi}$ yield stress of the inner pipe
$\sigma_{Ya}$ yield stress of the buckle arrestor
$\sigma_u$ ultimate stress
$\sigma_o$ yield stress of the pipe
$\sigma_{oa}$ yield stress of the buckle arrestor
\( \sigma_{ly} \) lower yield stress
\( \sigma_{uy} \) upper yield stress
\( \sigma_{lu} \) lower ultimate stress
\( \sigma_{uu} \) upper ultimate stress
\( m_p \) plastic moment
\( W_{ex} \) external work done by the net hydrostatic pressure
\( W_{in(f)} \) internal work due to the circumferential flexure
\( W_{in(m)} \) internal work due to the membrane effect
\( \nu \) Poisson’s ratio
\( P_p \) buckle propagation pressure of a single pipe
\( P_l \) buckle initiation pressure of a single pipe
\( P_{l2} \) buckle initiation pressure of outer pipe of the PIP system
\( P_{ps} \) buckle propagation pressure of the PIP system with a solid inner pipe
\( P_{cr} \) critical collapse pressure of a single pipe
\( P_c \) collapse pressure of a single pipe
\( P_{co} \) collapse pressure of the outer pipe
\( P_{ci} \) collapse pressure of the inner pipe
\( P_{ini} \) initial pressure at the onset of contact between outer and inner pipes
\( P_{p2} \) propagation pressure of the PIP system
\( P_o \) yield pressure of a single pipe
\( P_{ico} \) in-elastic collapse pressure of a single pipe
\( P_x \) cross-over pressure
\( P_{RST} \) Pressure from RST
\( V_o \) initial internal volume of the PIP system
\( \Delta V \) volume change of the PIP system
\( \Delta_o \) ovalization ratio
\( \phi \) geometric imperfection parameter
\[ \Delta l \quad \text{change in the circumferential length} \]
\[ \Delta A \quad \text{change in the cross-section area} \]
\[ U \quad \text{total energy dissipated in the RST} \]
\[ F_o \quad \text{load level at the four plastic hinges in the RST} \]
\[ L_{RST} \quad \text{length of the RST sample} \]
\[ l_t \quad \text{transition zone length in single pipe} \]
\[ l_{te} \quad \text{transition zone length in PIPs} \]
\[ V_f \quad \text{volume fraction of the CFRP} \]
\[ \Delta t_d \quad \text{time gap between the onset of the buckle initiation and time at which the buckle crosses the arrestor} \]
1.1 Background

The high demand for hydrocarbons as the primary source of the world's energy is driving the oil and gas companies to explore deep and ultra-deep offshore areas worldwide. Most of the onshore oil resources are being depleted, as the global need for oil products increases. Currently, offshore oil and gas productions represent one-third of the global crude oil and one-fourth of global gas productions, and these percentages are expected to increase in the next few years.

The subsea pipelines are increasingly being used in the oil and gas production and transportation industry, both for onshore or offshore areas. New designing challenges have emerged for subsea pipelines to operate in deep offshore areas, at high hydrostatic pressures and high-temperature flow. Since the operating life for pipelines is around 30 years or more, the pipeline is a large asset and also has a high capital cost. Once the pipeline is established, the maintenance and operation costs are relatively small.

A large number of oil reserves have been located offshore in the last 40 years (the Gulf of Mexico, North Sea, the Persian Gulf, East coast Brazil, Indonesia, Malaysia, Northwest Australia). In the early 1970s, the first reserves were located in the Gulf of Mexico, the North Sea and West coast California at a significant distance from the shore with few hundred feet water depths. Since that time, offshore operations have increased rapidly and gradually moved to deeper waters. Today, production has reached approximately 2,130 m, while exploration is proceeding at a water depth of 3,400 m in offshore Uruguay (Schuler, 2016). Long tubular structures including pipelines, risers and flowlines are the primary means of gathering and
transporting the oil and gas in offshore production operations. As the water depth increases, larger size pipes and higher operation pressure are required to increase the transporting efficiency.

Onshore pipelines are designed for the internal pressure requirements, and have diameters ranged between 35 to 65 inches. In terms of diameter-to-thickness ratios ($D/t$), they have $D/t$ range from 40 to 80. However, offshore pipelines typically have diameters less than 35 inches and diameter-to-thickness ratios ($D/t$) ranging from 15 to 40. Offshore pipelines are designed to resist the installation loads and the operational loads safely and to survive several long-term design conditions. During installation and operation, the empty pipe is prone to collapse under large external hydrostatic pressure. Moreover, the hydrocarbons can be at high internal pressure and temperature as high as 180°C. Long-term design conditions that can impact pipeline design include the development of a buckle propagation; impact by any foreign object causing a local buckling; the survivability of the pipeline in case of a sea storm, earthquake, etc. (Bai and Bai 2005; Kyriakides and Corona, 2007; Karampour and Alrsai, 2019)

As a result, all the aforementioned conditions increase the chance of pipeline failure, particularly buckle propagation failure. However, the subsea pipeline can experience a number of structural instabilities, such as lateral buckling, upheaval buckling, span formation and propagation buckling (Karampour et al., 2013; Albermani et al., 2011; Karampour et al., 2013; Karampour and Albermani, 2014). The lateral buckling failure occurs in a horizontal plane when the axial compression force in the pipeline is revived at an imperfection point. This can be led to uncontrolled lateral buckling causing the pipeline failure and rupture which is affecting the pipeline’s structural integrity. The upheaval buckling of offshore pipelines occurs as a result of axial compression force introduced along the pipelines in vertical plane due to high internal pressures and substantial temperature differences. Among these, propagation buckling is the most critical failure, particularly in deep and ultra-deep waters, and can rapidly affect many kilometres of pipeline.

In deep water applications, excessive buckling failure from installation and external hydrostatic pressure impose a considerable hazard for the mechanical integrity of the pipeline. To attain flow assurance in deep water and avoid catastrophes, pipe-in-pipe systems (PIPs),
which are comprised of an inner pipe, conveying the hydrocarbons at high-temperature and high-pressure and an outer pipe resisting the external hydrostatic pressure, are widely used in offshore pipeline applications. Typically, the annulus (the space between the two pipes) is either empty or filled with non-structural insulation materials such as polyurethane foam, mineral wool, granular or microporous materials or water (Karampour et al. 2017; Karampour and Alrsai, 2019; Kyriakides, 2002; Sriskandarajah et al. 1999).

Pipe-in-pipe systems are generally divided into two categories in the offshore industry: compliant and noncompliant. In a compliant or fully bonded system, the inner pipe and the outer pipe are attached, and the entire annulus is filled with insulation material; in the noncompliant or unbonded system, the pipes are connected only through bulkheads at discrete locations, and insulation pads are wrapped onto the inner pipe. In a compliant system, the relative movement between the inner and outer pipes is arrested and can deform uniformly, whereas the inner and outer pipes can move relative to each other in the noncompliant system. Due to their excellent thermal insulation performance, PIP systems are well exploited in the subsea developments to transport the hydrocarbons at high pressure and high temperature (HP/HT). Today, pipe-in-pipe systems are widely used in the Pacific, the Gulf of Mexico, the North Sea, and Africa (Bai and Bai 2005; Kyriakides and Corona, 2007; Bi and Hao, 2016).
1.2 Problem Statement

Numerous advancements have been made in the production of pipe-in-pipe systems in recent years, and pipe-in-pipe systems have been used in some pipeline projects. The safe use of pipe-in-pipe systems requires a comprehensive understanding of their performance in all possible limit states. A primary design concern for the subsea pipe-in-pipe system is collapse under external hydrostatic pressure. A second concern, often of equal importance, is the survival of the line, in case a buckle is accidentally propagated in the pipeline. In the presence of local damage in the outer pipe-wall (in the form of dents or corrosion, etc.) local collapse can be initiated in the pipeline. Once initiated, the circular cross-section of the pipe transforms into a dog-bone shape that propagates at high velocities and has the potential of rapidly destroying the entire pipeline. The lowest pressure required to propagate the buckle is termed propagation pressure \( (P_P) \), which is typically only 15% of the collapse pressure \( (P_c) \) (Kyriakides et al., 1998). Therefore, propagation buckling due to external hydrostatic pressure is one of the most challenging limit states of subsea pipelines that is affected by different aspects, including the geometric and material characteristics of the pipe-in-pipe system. Based on previous studies (Palmer & Martin 1975; Pasqualino & Estefen 2001; Albermani et al. 2011; Gong et al. 2012; Karampour et al. 2013; Karampour & Albermani 2014; Tang et al. 2014; Xue et al. 2015; Stephan et al. 2016; Karampour & Albermani 2016; Karampour et al. 2017), one of the geometric properties affecting the propagation buckling of pipelines is the outer diameter to wall thickness ratio \( (D/t) \). Therefore, in a pipe-in-pipe system with different \( D/t \) in the outer and inner pipes, the propagation buckling is expected to be different from that of single pipes. Besides, the material properties including yield stress of pipe-in-pipe systems make them substantially different from conventional single pipes. Therefore, considering the geometric and material properties, the current knowledge about pipe-in-pipe buckling behaviour should be updated.

Comprehensive studies of this subject should include finite element studies because it is unable to test the pipe-in-pipe system with all conceivable combinations of geometry, material, and operation conditions. Previous research on single pipes has resulted in abundant developments in the finite element modelling of pipelines in terms of appropriate mesh generation, element type, boundary conditions, etc. The main difficulty in finite element
analyses of the pipe-in-pipe system defines the contact and target elements for the inner and outer pipe. The primary requirement for accurate finite element modelling of a pipe-in-pipe system is a suitable isotropic geometry model. This model should be able to simulate the material response in all possible stress paths and the range associated with the considered limit state.

The next subject that needs to be explored is the buckle collapse modes through which the geometry and material properties of the pipe-in-pipe system affect the behaviour of the pipe in different conditions. Due to the complexity, the number of parameters required to define the pressure response in the pipe-in-pipe system is higher compared to that of single pipe models. Correlating these parameters to the pressure response from experimental and FE models in a simple and comprehensible manner is worthwhile research. Once this study is accomplished, the results can be used in other investigations to obtain the desired geometry properties of the pipe-in-pipe system for design.

Designing pipelines based on the propagation pressure is overly conservative and impractical. The preferred alternative is to design based on the collapse pressure and install buckle arrestors at certain intervals along the pipeline. Such arrestors are wrapped around the pipeline to limit the damage and safeguard the downstream section of the pipeline (Netto and Estefen, 1996; Kyriakides et al. 1998; Lee and Kyriakides, 2004). Existing buckle arrestors are usually stiff rings which locally enhance the circumferential stiffness of the pipeline and thus provide an obstacle in the path of a buckle propagation. However, the existing buckle arrestors have certain limitations in practice.

Recently, Carbon Fibre Reinforced Polymer (CFRP) composites are increasingly being used in the repairing, retrofitting and strengthening of existing structures especially in the offshore structures. The strength or stiffness of any structure member can improve by bonding a thin CFRP layer to the external surface (Zhang and Teng, 2010). Rehabilitation of oil and gas pipelines is one of the recent application of CFRPs. Employing these materials to repair the damaged pipes provides unique advantages over the conventional repair techniques (Cerccone, L., and Lockwood, J. D. 2005; Duell, J. et al. 2008; Esmaeel, R. et al. 2012; Shamsuddoha et al. 2013). Therefore, the CFRP repair system was successfully applied on offshore pipelines to avoid the costs and dangers of underwater welding or suspending production (Green, 2010).
Therefore, CFRP will be used in this research program to manufacture a new buckle arrestor for subsea pipelines to achieve high arresting efficiency. Since the CFRP arrestor is a new buckle arrestor proposed to subsea industries, feasibility, suitability and efficiency studies should be done first on single pipeline system in order to understand the arresting mechanisms, before extending the application to the pipe-in-pipe system. Due to the time frame for the PhD program, the CFRP arrestor is proposed and studied for the single pipeline system only in this research program, and further investigations will be recommended for the pipe-in-pipe system.

The research program is divided into five phases. In the first phase, a literature review has concluded that a pipe-in-pipe system is a satisfactory approach for pipelines application, and the propagation buckling is the most critical one of the structural instabilities, particularly in deepwater applications. In the second phase, experimental methodologies have been developed to predict the propagation buckling in the pipe-in-pipe system, including the hyperbaric chamber tests, ring squash tests (RST) and confined ring squash tests (CRST). In the second phase, the finite element (FE) model was developed in order to simulate the propagation buckling failure in pipe-in-pipe systems. By using the FE model, a numerical study and parametric analysis performed in the third phase to capture the effects of parameters affecting the propagation buckling pressure in pipe-in-pipe systems. In the fourth phase, collapse mechanisms and capacity of a non-pressurised inner pipe within the pipe-in-pipe system were investigated through a combination of experimental and numerical studies. In the last phase of this program, a new design concept of buckle arrestors for subsea pipelines by using Carbon Fibre Reinforced Polymer (CFRP) is proposed and is shown to achieve high arresting efficiency.

1.3 Objectives and Scope

This research project is designed to address the concerns stated in section 1.2. The primary objective of the proposed research project is to understand how the propagation buckling occurs in the pipe-in-pipe system under quasi-static steady-state conditions and observe the post-buckling deformed configurations of the PIP system. Then, design expressions are
proposed to predict the propagation and collapse pressures of the pipe-in-pipe system. Finally, a new buckle arrestor concept is proposed and evaluated.

Therefore, to achieve the objectives stated above, the following specific steps were considered:

- Design experimental set-up to predict buckle propagation pressure and buckle failure modes in PIP systems.
- Create 3D numerical Finite Element (FE) models by using commercial finite element package (ANSYS) and validate the FE model based on the experimental study.
- Perform numerical study and parametric analysis on propagation buckling of PIP systems to capture the effect of the parameters affecting the buckle propagation pressure and its collapse modes in the PIP systems.
- Perform a parametric analysis on collapse mechanisms of the inner pipe of the PIP system to predict the collapse pressure of the inner pipe.
- Evaluate the relationship between the propagation pressure of PIP system and collapse pressure of the inner pipe, and then propose empirical formulas based on the numerical results.
- Propose and experimentally validate a new concept of buckle arrestors for subsea pipelines by using Carbon Fibre Reinforced Polymer (CFRP).

1.4 Research Methodology

In this research project, the experimental and numerical approach will be followed in order to achieve the objectives. The motivation of this project stems from the lack of design guidelines for the subsea pipelines, particularly, PIP systems and lack of understanding of the propagation buckling behaviours of PIP systems under external hydrostatic pressure. In this section details on how this research project is going to unfold this gap is discussed.

1.4.1 Propagation buckling of PIP systems

In this part, propagation buckling responses of PIP systems under external hydrostatic pressure are investigated. Unlike the previous studies in the literature review, PIP system
comprising of carrier pipes with a diameter-to-thickness ($D_o/t_o$) ratio in the range 26-40 are examined in this research program. A set of experimental tests using the hyperbaric chamber and ring squash test (RST) will be performed followed by three-dimensional (3D) finite-element analysis. The experimental results will be compared with simplified two-dimensional (2D) analytical solution and the FE results. This part consists of:

Experimental study

In this experimental study, a number of PIPs sample and single carrier pipes will be tested at Griffith University Engineering Lab using the hyperbaric chamber. The samples will be subjected to external hydrostatic pressure. Results of this experimental work will be used later to validate the FE model. This experimental study is aiming for:

- Capturing the propagation pressure of PIPs and single carrier pipes.
- Capturing the buckle propagation modes.
- Plotting the pressure inside the chamber against the normalized change in volume of the carrier and inner pipes.

Previous studies (Kamalarasa and Calladine 1988; Albermani et al. 2011) have shown that the ring squash test (RST) is a satisfactory approach that gives a lower bound estimate of the buckle propagation pressure in single pipeline systems. The RST is performed on a short segment of the pipe with a length of $2.5D$, approximately in such a way as to produce the dog-bone shape of the deformed pipe observed in the hyperbaric chamber. In this experimental study, the RST will be adapted to estimate the propagation pressure of the pipe-in-pipe system.

Finite Element Analysis (FEA)

A Finite Element model will be established and validated by the results obtained from experimental investigations. Finite element simulations of 1.6 m long samples of PIPs used in the hyperbaric chamber tests will be conducted using ANSYS 17.1 (ANSYS 17.1 Release) in order to investigate the propagation buckling response of PIPs and single carrier pipes and the deformed shape with $D_o/t_o$ ratio in the range 26-40.
1.4.2 Parametric dependences of propagation buckling pressure of PIPs

The previous studies (Kyriakides and Vogler, 2012; Gong and Li, 2015) have only covered propagation pressure of PIPs with thick and moderately thick carrier pipes with diameter-to-thickness ($D_o/t_o$) ratio smaller than 25. However, in this study, the range of diameter-to-thickness ($D_o/t_o$) ratio has been extended to capture the effects of thin and moderately thin carrier pipes with $D_o/t_o$ of 30 and 40. Based upon the experimental observations and extensive numerical results with various $t/t_o$, $D_i/D_o$ and $\sigma_{yi}/\sigma_{yo}$, the collapse propagation modes of the PIPs will be identified. Moreover, a more accurate empirical formula for the propagation pressure $P_{ps}$ of the PIPs with solid inserts and a better empirical prediction for propagation pressure $P_{ps2}$ of the PIP systems with thin and moderately thin carrier pipes will be proposed in this study. Furthermore, a separate empirical formula will be proposed for thick and moderately thick carrier pipes of PIP systems.

1.4.3 Parametric dependences of collapse pressure of the inner pipe of PIPs

The hyperbaric chamber results suggest that the collapse pressure of the inner pipe of the PIP system is a function of geometric and material parameters of both inner and outer pipe. Therefore, a comprehensive parametric study will be performed in this section with a range of diameter-to-thickness ratios ($D/t$) between 15 and 40. Then, empirical expressions will be proposed for the collapse pressure of the inner pipe. The upper and lower bounds of the collapse pressure of the inner pipe will be identified. The proposed empirical equation will be compared to the experimental results of the tested PIPs.

1.4.4 CFRP buckle arrestor

A new buckle arrestor concept for the subsea pipeline will be introduced in this part. The concept involves using the Carbon Fibre Reinforced Polymer (CFRP) around the pipeline in order to achieve high arresting efficiency and increase the propagation buckling capacity of the pipeline. Since the CFRP arrestor is a new buckle arrestor proposed to subsea industry, feasibility, and efficiency studies will be done on single pipe system in this study in order to
understand the arresting mechanisms, and further investigations will be suggested to the pipe-in-pipe system. Therefore, a number of experiments will be conducted on single pipe systems inside the hyperbaric chamber to study the effect of the CFRP arrestor parameters; includes thickness, length and orientation as well as the manufacturing techniques. Also, the efficiencies will be established and compared to the existing buckle arrestors.

1.5 Thesis Outline

This thesis is structured as the following:

- Chapter 1 presents an introduction to the thesis and briefs the research objectives and methodologies.
- Chapter 2 reviews the literature relevant to this thesis. The pipe-in-pipe systems and behaviour of propagation buckling, as well as the buckle arrestors, are presented.
- Chapter 3 is a published journal paper, which experimentally and numerically investigates propagation buckling of subsea pipe-in-pipe system.
- Chapter 4 is a published journal paper, which represents numerical study and parametric analysis of the propagation buckling of subsea pipe-in-pipe system.
- Chapter 5 is a published journal paper, which studies the collapse mechanisms on the inner pipe of the pipe-in-pipe system.
- Chapter 6 is a submitted journal paper, which proposes and tests the CFRP buckle arrestor concept and its efficiency for subsea pipelines.
- Chapter 7 summarises the thesis findings and provides recommendations for future research.
2.1 Parameters of Subsea Pipe-in-pipe Systems (PIPs)

Over the past few decades, the pipe-in-pipe (PIP) systems have become a vital part of the subsea developments and broadly used in offshore pipeline applications in which thermal insulation of the pipeline is necessary. Due to its high insulation performance, in a PIP system heat losses between the transported fluid and the environment are much less than the conventional subsea pipelines. This is achieved using thermal insulation materials of very low thermal conductivity, such as aerogel, installed in dry atmospheric conditions between the inner pipe or "flowline," which conveys the fluid, and the outer pipe or "carrier pipe," which protects the system from external mechanical damage. In deep water applications, the carrier pipe is designed to resist the high ambient external pressure (typically in water depths more than 3,000 m) while the design of the inner pipe is mainly based on the internal pressure of the hydrocarbons at a temperature as high as 180°C (Jukes et al., 2009). The high hydrostatic pressure in deep waters may trigger a local collapse in the outer pipe. The collapse in the outer pipe can cause a buckle which if the pressure is maintained at certain level, will propagate rapidly along the length of the PIP. The collapse and its propagation in the outer pipe may trigger a collapse in the inner pipe as well.

This thesis will study the collapse mechanism and the buckle propagation in a PIP system and will propose an efficient product that can arrest the buckle in a single pipeline (either outer or inner pipes).

The collapse of single pipes with low $D/t$ under external pressure has been extensively investigated using analytical, experimental, and numerical methods, and is quite well understood (Park and Kyriakides, 1996; Tae-Dong, 1996; Estefen, 1999). Additionally; analytical studies which account for all major parameters affecting collapse are presented
(Stephens and Kulak, 1982). Consequently, a pipe-in-pipe system for deep waters can be safely designed against collapse failure. Nevertheless, as the same scenario for single pipelines, during the installation and operation processes of the pipeline structures, off-design conditions can develop resulting in local collapse (Park and Kyriakides, 1996; Kyriakides et al., 1998).

Local collapse, regardless of how it is introduced to the pipeline, usually will initiate a propagation buckling which may catastrophically propagate and collapse significant sections of the pipeline structure (Palmer and Martin, 1975; Kamalarasa and Calladine, 1988; Xue and Fatt, 2001; Albermani et al., 2011; Karampour et al. 2013; Karampour & Albermani 2014). For single pipeline systems, this problem is again quite well understood. The extent of the buckle propagation of the system is controlled by using buckle arrestors at intermittent locations along the length of the pipeline (Netto and Estefen, 1996; Park and Kyriakides, 1996; Kyriakides et al., 1998). To date, structural instabilities in PIPs have only been marginally addressed (Kyriakides and Vogler 2002; Kyriakides 2002; Gong and Li 2015; Zheng et al. 2014).

The structural behaviour of subsea pipelines was reviewed thoroughly by Stephens et al. (1982) in relation to the analytical developments of elastic and inelastic local buckling. Their report reviewed researches conducted from the 1920s to 1970s. As a result of their work, the elastic buckling of the pipe and the corresponding buckled shapes have shown to depend upon both geometric parameters such as the length to radius \( (L/R) \) ratio, the diameter to wall thickness \( (D/t) \) ratio, material properties including modulus of elasticity \( (E) \) and Poisson’s ration \( (\nu) \). Also, the ultimate strength equation of pipe buckling was examined and developed in their report. The main concern for most researches conducted before the 1970s was the strength of the pipe buckling. However, Jirsa et al. (1972) investigated experimentally concrete coated steel pipes with a diameter ranging from 10.75 to 20 inches, in order to measure the influence of ovality on flexural behaviour when tested under four points bending. Afterwards, the researchers were focused on studying the diameter to wall thickness \( (D/t) \) to investigate the pipeline buckling.

Tae-Dong (1996) asserted that the factors influencing the collapse are the pipe's diameter to thickness ratio \( (D/t) \), its material properties (Young's modulus, yield stress and hardening...
parameters), initial geometric imperfections like ovalization ratio and thickness variations, and anisotropies in yielding and residual stresses introduced in the manufacturing process. Dyau and Kyriakides (1993) developed the analysis tools which can take these factors into account and can predict the collapse pressure \((P_c)\). It is also well known that collapse in such structures, should it occur, is local. However, under favourable conditions, the local collapse can develop into a propagating buckle. In a constant pressure environment, a buckle propagates if the ambient external pressure is higher than the second characteristic pressure of the pipe, its propagation pressure \((P_p)\). A buckle can be initiated at any pressure between the collapse and propagation pressures: \(P_p < P_l < P_c\). Propagation of the buckle will continue until the externally applied pressure reaches a level lower than the propagation pressure, or until an arresting device is encountered.

In research series, Kyriakides (Kyriakides 2002; Kyriakides and Vogler 2002) have identified the geometric parameters (Figure 2.1) and material properties of the pipe-in-pipe system through a combination of experiments and numerical investigations. In their studies, PIP systems with the outer pipe having \(D/t\) values of 24.1, 21.1 and 16.7 have been investigated, and the propagation pressure of the PIP system \((P_{p2})\) has been quantified parametrically. Their proposed expression for \(P_{p2}\) shows that the thickness ratio \((t_i/t_o)\) has the major effect on \(P_{p2}\) and diameter ratio \((D_i/D_o)\) comes second, also their studies show that the material properties have less effect than geometry parameters of the PIP system.
2.2 Collapse Performance of Subsea Single Pipes and PIP Systems

The design considerations of the offshore pipeline are divided into two general performance categories: (i) short-term and (ii) long-term design conditions. The primary design consideration for short-term stability is to design the pipeline to safely sustain first the installation loads, and second, the operational loads. Where control of pipeline curvature and tension is critical for the prevention of buckling. Since the oil or gas fields’ life may exceed 20 to 30 years, pipelines are also designed to survive various long-term design conditions such as natural disasters (earthquakes, sea storms, etc.), corrosion and accidental damage by a foreign object.

Over the past few decades, many researchers have conducted extensive studies on the collapse and failure of the offshore pipeline based on theoretical analysis, collapse experiments and finite-element models (Demars et al. 1977; Strating, J., 1981; Kyriakides and Babcock, 1982; Lancaster and Palmer, 1993; Park and Kyriakides, 1996; Xue and Hoo Fatt 2002; Netto et al. 2007; Netto, T.A. 2010; Albermani et al. 2011). Damage in the pipe-wall (in the form of dents or imperfections) is one of the most common causes of local collapse in pipelines. Dents can occur due to accidental impact by foreign objects as well by local buckling induced during...
transportation of the pipe, or during installation, or during the operation of the line (Demars et al. 1977; Strating, J., 1981). Such dents will significantly reduce the local collapse pressure capacity of the pipe and will trigger a propagation buckling along the pipeline (Kyriakides and Babcock, 1982) (see Figure 2.2). Furthermore, for gas pipelines under large internal pressure, dents, sometimes accompanied by gouges, often introduce burst failures (Lancaster and Palmer, 1993; Park and Kyriakides, 1996).

![Figure 2-2: Initiation of a propagation buckling in an offshore pipeline by impact of a foreign object. (Park T., 1996)](image)

The main parameters affecting the collapse of the pipelines are the pipe geometry represented by the value of $D/t$, the material properties such as strength and stiffness and geometric imperfections represented by the initial ovality parameter, $\Delta_o$ and wall thickness variation parameter. For different combinations of these parameters, the pipe collapse can occur in the elastic or plastic state of the material. However, a thin-walled pipe with a high enough value of $D/t$ and perfect geometry can often induce the elastic buckle, and the corresponding critical buckling pressure $P_{cr}$ can be evaluated from the following (Timoshenko and Gere 1961):

$$P_{cr} = \frac{2E}{(1-v^2)} \left( \frac{t}{D_o} \right)^3$$  \hspace{1cm} (2.1)
For the pipe collapse to occur in the plastic state of the material, the corresponding yield pressure \( P_o \) can be evaluated as the following (Timoshenko and Gere 1961):

\[
P_o = 2\sigma_y \left( \frac{t}{D_o} \right)
\]  

(2.2)

where \( E \) is Young’s modulus, \( \sigma_y \) is the yield stress, \( \nu \) is the Poisson ratio of the material, \( D_o \) is the mean diameter of the tube and \( t \) is the wall thickness.

Previous researches (Albermani et al., 2011; Chater and Hutchinson, 1984; Kyriakieds et al., 1984) have shown that the collapse pressure is a combined elastic-plastic failure. The inelastic expression \( P_{eco} \) suggested by Timoshenko (Timoshenko and Gere 1961) can be used to predict the collapse pressure of a tube under uniform external pressure, based on the yield pressure \( P_o \), elastic critical buckling pressure \( P_{cr} \), and geometric imperfection parameter \( \varphi \):

\[
P_{eco} = \frac{1}{2} \left\{ (P_o + \varphi P_{cr}) - \left[ (P_o + \varphi P_{cr})^2 - 4 P_o P_{cr} \right]^{0.5} \right\}
\]  

(2.3)

\[
\varphi = 1 + 3 \Delta_o \frac{D}{t}
\]  

(2.4)

The primary geometric imperfection parameters of the pipe are its initial ovality and the measure of the ovality of the most deformed cross-section, \( \Delta_o \), defined as follows:

\[
\Delta_o = \frac{D_{max} - D_{min}}{D_{max} + D_{min}}
\]  

(2.5)

where \( D_{max} \) is the maximum distance across the deformed cross-section and \( D_{min} \) is the minimum distance across the convex part of the deformed cross-section, as shown in Figure 2-3.
Kyriakides et al. (Kyriakides et al. 1984), through experimental investigations, studied and measured the collapse pressures of steel pipes. These pipes were dented with various degrees using indentors of several different shapes and dimensions. The steel pipes had $D/t$ ratios in the range of 33 to 43. Their study concluded that the collapse pressure of the dented steel pipes is related to the "ovalization" of the most deformed cross-section of the damaged pipe section. All other details, such as the shape and dimension of the dents, were found to play a secondary role. Later on, Park and Kyriakides (Park and Kyriakides, 1996) extended the work done by (Kyriakides et al. 1984) through a combination of experiment and numerical analyses from pipes with lower $D/t$ ratios (18.9, 24.2 and 33.6) in order to cover the needs for deep water applications. The main interesting thing in their study was that the collapse is relatively insensitive to the detailed geometric parameters of the dent, however, is critically affected by the maximum ovalization ratio. Also, the collapse pressure was found to approach the buckle propagation pressure when larger dents induced.

Yeh and Kyriakides (1986) developed a two-dimensional nonlinear analytical solution based on beam element and the solution method to calculate the collapse pressure of long thick-walled pipes under external pressure, their analytical solution includes the initial geometric imperfections of the pipe cross-section such as initial ovality and wall thickness variations, as well as, the effects of residual stresses and initial inelastic anisotropy are considered. Subsequently, they (Yeh and Kyriakides 1988) conducted a series of small-scale collapse experiments of steel pipes with $D/t$ ratios between 10 and 40 and explored the different trend in parameters with respect to collapse pressure. Xue and Hoo Fatt (2002) proposed analytical
solutions for elastic buckling of a non-uniform (two regions, one with a nominal thickness and the other with reduced wall thickness), long cylindrical pipe subjected to external hydrostatic pressure, and the buckling modes of the pipe cross-section were found to occur depending on the relative thickness of two different regions.

The effect of corrosion or erosion defects on the collapse pressure of subsea pipelines subjected to external hydrostatic pressure was investigated, respectively by Netto et al. (2007, 2009) through the combination of experiments and nonlinear numerical analyses. Their studies have shown different collapse modes, which are related to the defect geometry in the pipeline. Furthermore, they noticed that there is an interaction between pipe ovality and defect, which is affecting the collapse pressure of the pipeline. Also, the defect type (internal or external) and how far is the defect from the ovalized cross-section can affect the collapse pressure. Sakakibara et al. (2008) studied the effect of the internal corrosion defects on the collapse of pipelines under external hydrostatic pressure. The Experiments were performed on stainless-steel pipes with $D/t$ values of 21.0 and 18.7 with axially uniform gouges of different thicknesses. Their study concluded that the gouge depth has a significant effect on the collapse pressure, where the collapse pressure can be reduced by approximately 50% when the gouge depth reached 50% of the pipe wall thickness.

Unlike single pipeline systems, collapse mechanisms of PIP systems have only been marginally addressed (Kyriakides 2002; Kyriakides and Vogler 2002; Gong and Li 2015; Li 2018). Moreover, these studies have been purely focused on the buckle propagation pressure ($P_{p2}$) of the PIP systems. The existing knowledge on buckling of single-walled pipelines under external pressure can be used to predict the collapse pressure of the outer pipe of a PIP system. However, the buckling mechanisms of the inner pipe and its collapse pressure are different from those of a single pipeline. Moreover, to the knowledge of the author, there is no existing study on the collapse of the inner pipe of a PIP system under external hydrostatic pressure.
2.3 Propagation Buckling in Subsea Single Pipes and PIP Systems

The techniques of pipeline installation depend on a combination of three main factors such as the water depth, the diameter and the weight of the pipeline. The installation techniques are mainly divided into the following categories: (1) S-lay; (2) J-lay; and (3) Reeled. The S-lay method is the most common method of pipeline installation in shallow water. The pipeline starts in a horizontal position and deforms in an S-shape on the way to seabed, and during this operation, the pipeline is subjected to bending combined with high tension, and external pressure and the maximum curvature occurs in the vicinity of seabed in the presence of maximum external pressure as illustrated in Figure 2-4.

Figure 2-4: Overbend and sagbend regions in the pipe during an S-lay method.
For deepwater applications, J-lay method was invented since it requires less tension on the pipeline due to shorter suspended length and the pipeline starts in a vertical position and lowered to the seabed. The maximum bending and external pressure occur in sagbend. The J-lay configurations are shown in Figure 2-5. In the Reel lay method, several kilometres of pipe is wrapped around a large diameter reel and is carried to the installation site. Reeled pipelines can be installed up to 10 times faster than conventional pipelay. The pipeline is subjected to high plastic bending curvatures during spooling and unspooling, and repetitive bending cycle ovalizes the pipe cross-section which can be caused permanent (residual) stresses, influencing the pipe material properties also, affecting the structural performance of the reeled pipe. Figure 2-6 shows the typical reeling method.

**Figure 2-5:** Sagbend region in the pipe during a J-lay method.
Same aforementioned pipe-laying methods are used to install the pipe-in-pipe system from a barge into the water; however, a PIP system is generally much heavier than a single-wall pipeline. Therefore, the carrier pipe in the PIP system is usually subjected to a large tension from the handling devices combined with external hydrostatic pressure and bending moment. Under such loads, a local buckle is likely to occur on the carrier pipe, especially in the sagbend region which is close to the seafloor, and the overbend region which is close to the barge.

Local buckle, no matter how it is induced to the pipeline, usually will initiate a propagation buckling which can travel in a short time flattening significant sections of the pipeline structure as shown in Figure 2-7. The elastic buckling is followed by a plastic collapse and change in the cross-section of the tube from circular to oval and finally a dog-bone shape. If the pressure is maintained, the buckle will propagate quickly along the length of the pipe. Offshore pipelines normally experience high service external pressure; therefore, the buckle will propagate through the length, forcing the flow line to be shut. The lowest pressure that maintains propagation is known as the propagation pressure ($P_P$) and is much smaller than the collapse pressure.
The problem has been studied extensively so that today the factors influencing the propagating buckle are well understood for single pipe system (Mesloh et al. 1973; Palmer and Martin 1975; Yeh and Kyriakides 1986; Yeh and Kyriakides 1988; Xue and Fatt, 2001; Albermani et al., 2011; Karampour and Albermani, 2015; Stephan et al. 2016). These are the pipe's diameter to thickness ratio ($D/t$), its material properties (Young's modulus, yield stress and hardening parameters), initial geometric imperfections such as ovality and thickness variations, and anisotropies in yielding and residual stresses introduced during the manufacturing process (Kyriakides et al. 1994).

The propagation buckling phenomenon was initially investigated by Mesloh et al. (1973) in a single pipeline, and then Palmer and Martin (1975) proposed the first analytical solution to predict the propagation pressure in subsea pipelines. In their proposed solution, only the initial and final configurations of the pipe cross-section were considered, and the material was assumed to be rigid-perfectly plastic with yield stress $\sigma_y$. By equating the plastic work dissipated in the cylinder due to circumferential bending at the four plastic hinges (internal...
work) (see Figure 2-8), to the work done by external hydrostatic pressure due to the change in the pipe volume, the propagation pressure of a single pipe was found to be

$$P_{\text{pui}} = \pi \sigma_y \left( \frac{t}{D} \right)^2$$  \hspace{1cm} (2.6)

where $D$ and $t$ are the outside diameter and wall thickness of the pipe respectively.

\begin{figure}
\centering
\includegraphics[width=0.5\textwidth]{figure2-8}
\caption{A schematic of 2D deformation stages in propagation buckling.}
\end{figure}

The first experimental investigations on the propagation buckling were performed by Mesloh et al. (1976) and Johns et al. (1978), where the propagation buckling was investigated for pipes with different $D/t$ ratios, tested in different arrestor geometries. Early studies on the buckling behaviour of pipelines generally involved experimental and analytical approaches. Kamalarasa and Calladine (1988) performed physical testing of stainless steel, and aluminium alloy pipes ranging in radius from 11.66 to 13.85 mm with thicknesses between 0.31 and 1.63 mm to provide expressions for buckle propagation pressure. Their experimental results provided a simple improvement of Palmer and Martin’s equation.
Kyriakides and Babcock (1982) performed a number of tests on aluminium and steel pipes to measure the propagation pressure. They also conducted a number of parametric studies on the collapse of elastoplastic rings under external pressure where they observed the influence of key parameters in the ring pressure just before contact of the two opposite internal surfaces. Using the ring collapse pressure, they (Kyriakides and Babcock, 1982) showed that the propagation pressure depends on the diameter-to-thickness ($D/t$) ratio, yield stress and the tangent modulus of the pipe material. Subsequently, Chater and Hutchinson (1984) derived the following expression (Eq. 2.7) for the propagation buckling pressure by considering the elastic-plastic collapse under the cylinder in-plane strain condition

$$\hat{P}_p = \frac{2\pi}{\sqrt{3}} \sigma_y \left( \frac{t}{D} \right)^2$$

(2.7)

However, the aforementioned formulae gave underestimations of the propagation pressure by some 15%. Also, previous experimental studies (Albermani et al. 2011; Kyriakides and Babcock 1982) proved that Eq. (2.6) underestimates the propagation pressure of the pipeline, especially at lower $D/t$ ratios. Therefore, Albermani et al. (2011) developed an experimental test program using ring squash tests and hyperbaric chamber tests to investigate the propagation buckling in deep subsea pipelines. The experimental investigations were performed using aluminium pipes with diameter-to-thickness ($D/t$) ratio in the range 20-48. In this study, a textured cylindrical geometry pipe was proposed. Preliminary Finite Element analysis of the textured pipe indicated that a substantial increase in buckling capacity can be achieved for the same thickness. Also, their study proposed improvements for Palmer and Martin formula (Eq. 2.6) by accounting for the circumferential membrane as well as flexural effects in the pipe wall

$$P_r = \frac{3}{2.515} \left[ \frac{\pi}{4} \sigma_y \left( \frac{t}{r} \right)^2 \right] = 1.193 \hat{P}_p^{PM}$$

(2.8)

Gong et al. (2012), suggested that the material property plays an essential role in propagation buckling pressure. The primary influencing parameters are the strain hardening modulus $E'$ and the yield stress $\sigma_y$. Their study showed that the pipes with larger yield stress and strain...
hardening modulus always provided a higher propagation pressure. Also, the effects of yield stress and strain hardening modulus were shown to be more pronounced in the pipes with lower \(D/t\) ratios compared with larger \(D/t\) ratios. Thus, steel with higher strength and toughness should be preferred to enhance the propagation buckling capacity in practical application, specifically for deep water pipelines. On the other hand, Kyriakides proposed some empirical expressions to predict the propagation pressure of a single pipe (Dyau and Kyriakides, 1993b; Kyriakides and Lee, 2005; Lee and Kyriakides, 2004), all their proposed expressions are only based on \(D/t\) ratio and yield stress.

As mentioned in the previous paragraphs, many researchers have thoroughly investigated the propagation pressure of single pipelines using analytical solutions, experimental and numerical methods; however, only marginal research has been conducted on propagation buckling pressure of pipe-in-pipe systems. Kyriakides (2002) conducted an experimental study on propagation buckling of stainless steel PIP systems with 2-inch diameter carrier pipes with diameter-to-thickness (\(D_o/t_o\)) ratios of 24.1, 21.1, and 16.7 and inner pipes with various diameter-to-thickness (\(D_i/t_i\)) ratios ranging between 15 and 37. In his study, two buckling modes were observed. In the first mode, the local collapse of the outer pipe led to simultaneous collapse of the inner pipe, whereas in the second mode the carrier pipe collapsed over the inner pipe without affecting it.

Kyriakides and Vogler (2002) studied the collapse mechanisms of the pipe-in-pipe system, using in-plane strain condition and by adopting the aforementioned four plastic hinges mechanism in the carrier and inner pipes (Figure 2.9). Their study considered strain hardening in the material model and proposed an equation for propagation pressure of the pipe-in-pipe system (\(P_{p2}\)):

\[
P_{p2} = \frac{2\pi}{\sqrt{3}} \sigma_{yo} \left( \frac{t_o}{D_o} \right)^2 \left[ 1 + \frac{\sigma_{yi}}{\sigma_{yo}} \left( \frac{t_i}{t_o} \right)^2 \right]
\]

(2.9)

where subscripts \(o\) and \(i\) correspond to the outer pipe and inner pipe respectively.
Kyriakides (2002), developed an empirical formula (Eq. 2.10) for buckle propagation pressure of a PIP system based on an extensive experimental study with $D_o/t_o$ values of 24.1, 21.1 and 16.7 and inner tubes of several diameters and wall thicknesses. Furthermore, Gong and Li (2015) proposed a different empirical equation (Eq. 2.11) based on finite element (FE) simulations which estimate the propagation pressure of the PIP system. It is worth mentioning that both of those studies only included thick and moderately thick pipes, with $D_o/t_o$ ratios smaller than 25 in almost similar range; however, their proposed equations are different. Again, the practical range of $D_o/t_o$ ratios of pipelines exploited in subsea engineering is between 15 and 40. This suggests the necessity of an investigation to be performed on buckling of PIPs with $D_o/t_o$ ratios larger than 25.

$$\frac{P_{P2}}{P_P} = 1 + 1.095 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^{0.4} \left( \frac{D_i}{D_o} \right) \left( \frac{t_i}{t_o} \right)^2$$  (2.10)

$$\frac{P_{P2}}{P_P} = 1 + 0.970 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^{0.8} \left( \frac{D_i}{D_o} \right)^{0.3} \left( \frac{t_i}{t_o} \right)^2$$  (2.11)

Recently, Zheng, et al. (2014), performed a numerical investigation on subsea single pipe and pipe-in-pipe systems subjected to external pressure, their study concluded that by decreasing the internal pressure of the dented pipe, the chance of the propagation buckling to occur in the single pipe system is higher than it is for PIP system. Buckle initiation pressures for single
pipe and PIP systems were found to be closely correlated to geometric imperfections (in the form of dents or ovality of the outer pipe). Also, the inner pipe is insensitive to the external hydrostatic pressure, and therefore a buckle is less likely to propagate.

2.4 Buckle Arrestors

Subsea pipelines, including the PIP systems, can experience several structural instabilities, such as lateral buckling, upheaval buckling, span formation and propagation buckling. As explained in the previous literature, propagation buckling is a major concern, particularly in deep water, because it can rapidly destroy many kilometres of the pipeline. Therefore, in order to prevent the propagation of buckling in subsea pipelines, appropriate devices known as buckle arrestors need to be installed.

Since the propagation pressure ($P_p$) is generally much lower than the collapse pressure ($P_c$), designing offshore pipelines based on the propagation pressure is overly conservative and impractical. Moreover, lifting and substituting the damaged section of a pipeline is extremely costly as well. Therefore, the preferred alternative is to design based on the collapse pressure and install buckle arrestors at certain intervals along the pipeline. Buckle arrestors have been increasingly adopted to arrest propagation buckling to a limited length of the pipeline. In other words, should a propagating buckle occur, it must be arrested in a short distance from the initiation point so that the downstream section of the pipeline is maintained intact.

Existing buckle arrestors are devices which can increase the circumferential stiffness of the pipeline in the hoop direction and thus provide an obstacle in the path of a propagation buckling. They typically take the form of stiff rings which are wrapped around the pipeline at certain intervals. The fundamental types of buckle arrestors can mainly be categorized as; (1) slip-on arrestors (Kyriakides and Babcock, 1980), (2) spiral arrestor (Kyriakides S, Babcock, 1982), (3) integral arrestors (Netto and Estefen, 1996; Kyriakides et al. 1998) and (4) clamped arrestors (Kyriakides, 2002). The slip-on arrestor where the arrestor slipped over the pipe has been used in several pipelines in relatively shallow waters, and the integral arrestor where it
welded between two strings has been preferred for pipelines installed in moderately deep and deep waters.

The integral arrestor is a heavier wall section of pipe that needs special welding techniques to avoid cracking of the welds when they are deformed by a propagation buckling (Kyriakides et al. 1998). This makes it perhaps the most expensive of the arrestor concepts. The cross-over mechanisms of the integral buckle arrestors can be categorized into two modes; flattening mode and flipping mode, the occurrence of these modes is determined by the geometry of the pipeline and the arrestor (Toscano et al. 2008). The spacing between arrestors is usually determined by practical considerations and can be optimised based on the cost-benefit balance between the expected cost of the intervention and repair and the cost of installation. However, the spacing of 90-240m (300-800ft) has been typical to date (Netto, 1998; Netto and Kyriakides, 2000 Part I). The detailed design features of the integral arrestor are shown in Figure. 2-10.

![Figure 2-10: Integral buckle arrestor design features. (Netto, 1998)](image)

The fact that slip-on buckle arrestors do not require welding is an advantage; however, arresting efficiency of 100% is not always achievable (Kyriakides and Babcock, 1980; Lee and Kyriakides, 2004). Slip-on arrestors can be more effective if they are in close contact with the external surface of the pipe wall. Hence, they are often grouted either with cement or more
recently by a stiff polyurethane grout to ensure the highest efficiency (Langner, 1999). Buckles may cross over the slip-on arrestors in three different modes. In relatively short and thin arrestor, the buckle will cross in a flattening mode as illustrated in Figure 2-11a. If the arrestor is long and thick, the buckle will cross over in a U-shape mode (Figure 2-11b). Whereas the downstream section buckles in the flipping (flip-flop) mode if the arrestor is relatively short but thick rings (Figure 2-11c).

![Crossover modes of slip-on arrestors](image)

**Figure 2-11**: Crossover modes of slip-on arrestors; (a) flattening mode; (b) U shape mode; and (c) flip-flop mode. (Kyriakides, 2002)

The buckle arrestors were studied experimentally and/or numerically by Kyriakides and Babcock (1980, 1981), Kyriakides (1980), Netto and Estefen (1995), Park and Kyriakides (1997), Kyriakides et al. (1998), Netto and Kyriakides (2000, Part I), Netto and Kyriakides (2000, Part II), Kyriakides and Netto (2004), Gong and Li, 2017, Lee and Kyriakides, 2004 and Toscano et al. (2008). The combined experimental and numerical results were used to generate new design guidelines for such devices. The external pressure required for propagating the collapse pressure through the buckle arrestors is termed as the cross-over pressure ($P_c$); therefore, the buckle arrestor performance is evaluated by measuring the arrestingor’s cross-over pressure.
In 1996, Netto and Estefen tested two buckle arrestor configurations (ring and cylinder) welded on steel alloy pipes with $D/t$ ratios of 16 and 23 under quasi-static conditions in a hyperbaric chamber. Their study asserted that the cross-over pressure $P_X$ can be determined experimentally and mainly depends on geometrical and material parameters of both pipes and buckle arrestor. Toscano et al. (Toscano et al. 2008) performed an experimental investigation on the cross-over mechanisms by using welded integral arrestors. In their study, they found that the cross-over ratio $P_X/P_P$ was much higher for the flipping case than for the flattening case. Furthermore, in previous research conducted for slip-on buckle arrestors, Kyriakides and Babcock (1980) concluded that the cross-over pressure expressed as function by

$$\frac{P_X}{P_P} = F \left( \frac{E}{\sigma_o}, \frac{\sigma_{oa}}{\sigma_o}, \frac{D}{t}, \frac{h_a}{t}, \frac{L_a}{t} \right)$$

An approximate expression was obtained by Netto and Estefen (1996) by using the fitting line of the experimental data through the non-linear least square approach as the following expression

$$\frac{P_X}{P_P} = 11.46 \left( \frac{\sigma_o}{\sigma_{oa}} \right) \left( \frac{h_a}{D} \right) \left( \frac{L_a}{t} \right)^{0.4}$$

where $P_X$ is the cross-over pressure, $P_P$ is the propagation pressure, $\sigma_o$, $\sigma_{oa}$ is the pipe and buckle arrestor yield stresses, respectively, $h_a$ is the buckle arrestor thickness, $L_a$ is the buckle arrestor length, $D$ is the pipe outside diameter, and $t$ is the pipe wall thickness.

The most popular measure of the effectiveness of such arrestors is Kyriakides’ arresting efficiency $\eta$, and was proposed by (Kyriakides S. 1980). An arrestor efficiency can be used as a basis for comparing the merits of various arrestor designs; a lower bound for the arresting efficiency of an arrestor is given by:

$$\eta = \frac{P_X - P_P}{P_C - P_P} \quad 0 \leq \eta \leq 1$$

where, $P_X$ is the cross-over pressure, $P_P$ is the propagation pressure and $P_C$ is the collapse pressure.
Clearly, an arrestor which allows a buckle, propagating at $P_p$, to go through has an efficiency of zero, and one that has a cross-over pressure equal to $P_c$ has an efficiency of 1. The integral arrestor was studied experimentally beside efficiency by Park and Kyriakides (1997) using two major series of integral arrestor. The first series had fixed thicknesses $h \approx 2.1t$ and lengths which varied from $0.5D \leq L_a \leq 2.0D$ and the second series in their study involved seven arrestors with fixed length $L_a \approx 1.25D$ and thicknesses that varied from $1.7t \leq h \leq 3.3t$. The cross-over pressure was found to increase with arrestor length, however, the relationship was nonlinear. Also, the efficiency was seen to be slightly nonlinearily correlated to the arrestor thickness. In this instance, the arrestor with the maximum thickness ($h > 2.7t$) can achieve an arresting efficiency of 1.0. It is interesting to note that the efficiency of a buckle arrestor effected by arrestor’s thickness at a higher level than the length (Netto and Estefen 1996). Furthermore, Kyriakides et al. (1998) have been developed an empirical design equation for integral arrestor based on a parametric study, which is combined between arresting efficiency and the major problem variables for pipe and arrestor. The results of this process produced the following equation:

$$\eta \approx A_I \left( \frac{P_{CO}}{P_p} - 1 \right)$$

where, $\eta$ is arrestor efficiency, and $A_I$ is a constant determined empirically.

In 2008, Lee et al. (Lee et al. 2008) have improved the empirical design formula of Kyriakides et al. (Kyriakides et al. 1998) for integral arrestors. The experimental cross-over pressures enhanced with numerically generated values were used to update Eq. (2.15), it is given by:

$$\eta = A_I \left( \frac{P_{CO}}{P_p} - 1 \right)$$
Kyriakides and Park (Park and Kyriakides 1997; Kyriakides et al. 1998) performed a series of experimental and numerical studies on the design of the integral buckle arrestor. In their studies, they produced the minimum thickness ($h_{cm}$) for high-performance arrestors. This is the lowest thickness that a long arrestor must have in order for its propagation pressure ($P_p$) to be equal to the collapse pressure of the pipe. Thus, $h_{cm}$ can be evaluated as

$$P_p(h_{cm}) = P_{CO}$$

(2.17)

Recently, Gong and Li (Gong and Li, 2017) conducted broad parametric studies to identify the mechanism ruling the arresting performance of integral arrestors. In their study, more reasonable empirical design formulas for the cross-over pressure and efficiency were proposed as the following;

$$\eta = 1.944 \left( \frac{E}{\sigma_{0.2}} \right)^{0.87} \left( \frac{\sigma_{0.2}}{E} \right)^{0.94} \left( \frac{t}{D} \right)^{0.46} \left( \frac{L_a}{D} \right)^{0.8} \left( \frac{h}{t} \right)^{3.05} \frac{E}{E}^{0.14} \frac{E_a}{E}^{0.16} \right)$$

(2.18)

$$\frac{P_X}{P_p} = 1 + 1.944 \left( \frac{E}{\sigma_{0.2}} \right)^{0.87} \left( \frac{\sigma_{0.2}}{E} \right)^{0.94} \left( \frac{t}{D} \right)^{0.46} \left( \frac{L_a}{D} \right)^{0.8} \left( \frac{h}{t} \right)^{3.05} \frac{E}{E}^{0.14} \frac{E_a}{E}^{0.16} \right)$$

(2.19)

Kyriakides and Lee (Lee and Kyriakides, 2004; Kyriakides, 2002) measured the cross-over and propagation pressures of 22 stainless-steel pipes with two slip-on buckle arrestors on each pipe, in various $D/t$ ranged from 14 to 94. Their experimental results showed that the efficiency of the slip-on arrestor changes with the corresponding tube’s $D/t$ value. Based on the experimental results from their study, they proposed empirical expressions to provide the bounding arresting efficiencies of slip-on arrestor; lower and upper bound (Figure 2.12). Based on the studied $D/t$ ratios; the efficiency of slip-on arrestor at $D/t \approx 28$ was ranged between 68%-81% and at $D/t \approx 40$, the efficiency of slip-on arrestor ranged between 78%-100%. Also, in steel slip-on arrestors and regardless of the $D/t$ ratio of the tube, $h \geq 2.5t$ is required to provide arresting efficiency close to lower bound (Lee and Kyriakides, 2004).
In addition, the experimental results in (Lee and Kyriakides, 2004) were used to establish an empirical expression for slip-on arrestor efficiency as follows;

\[
\eta \approx 0.3211 \left( \frac{\sigma_o}{\sigma_{oa}} \right)^{0.8} \left( \frac{t}{D} \right)^{0.75} \left( \frac{L_o}{t} \right)^{0.98} \left( \frac{h}{t} \right)^{2.1} \left( \frac{P_{CO}}{P_p} - 1 \right)
\]

(2.20)

It is worth mentioning that all the aforementioned empirical expressions are not capable of providing an accurate estimation for efficiency, especially with those arrestors have \( \eta > 0.7 \). Therefore, Lee and Kyriakides in (Lee and Kyriakides, 2004) reported that their empirical fit must be used in conjunction with one or both of the efficiency bounds in Figure 2-12. Also, in (Gong and Li, 2017), their study shows that the data points for arrestors with \( \eta > 0.7 \) exhibited a wide scatter and there is no rule to follow. They suggested using the lower bound which proposed in their study to roughly estimate the arresting efficiency of integral arrestors.

![Figure 2-12: Arresting efficiency bounds for slip-on buckle arrestors based on \( \eta_{IC} \) (Upper bound) and \( \eta_{PC} \) (Lower bound). (Lee and Kyriakides, 2004)](image)

A new buckle arrestor concept was introduced for pipe-in-pipe systems by Olso and Kyriakides (2003). The idea comprises either one single ring or a series of closely packed thin rings placed in the annulus between the outer and inner pipes (Figure 2.13). The experiments
involved 2 inches diameter carrier pipes of three different $D/t$ ratios and internal rings of various dimensions (length and thickness). It is interesting to notice that the inner pipe had only a minor effect on the cross-over pressure of this type of arrestors. Olso and Kyriakides (2003) and Kyriakides and Netto (2004) have asserted that the gap between the internal ring arrestors and the inner surface of the carrier pipe is a parameter that required special attention and plays an important role regards the efficiency. Then, the experimental data were used to develop an empirical formula for arresting efficiency expressed as a function of the key nondimensional variables, and the parameters varied included the arrestor length, thickness and yield stress as well as the $D/t$ of the carrier pipe as:

$$\eta \approx 1.0145 \left( \frac{\sigma_{sat}}{\sigma_o} \right)^{0.65} \left( \frac{t}{D} \right) \left( \frac{L_a}{h} \right) \left( \frac{h}{t} \right)^{2.2} \frac{P_{CO}}{P_p} - 1$$

(2.21)

It should be mentioned that this formula was developed based on results involving just the carrier pipe. Their study shows that the inner pipe generally will cause some small increase in the cross-over pressure of such arrestors. Therefore, this increase was treated in their study as an added safety factor to the arrestor design. However, Gong et al. (2018) proposed empirical expressions for cross-over pressure and efficiency of inward integral buckle arrestor for the pipe-in-pipe system. Their expressions account for geometric and material parameters for both inner and carrier pipes as well as the arrestor. For arrestors with $\eta > 0.8$, their formulas are irrelevant, because the flipping cross-over mode will predominate, and their formulas only applicable to flatting cross-over mode.
Figure 2-13: Internal ring buckle arrestor for pipe-in-pipe systems proposed by Olso and Kyriakides (2003).

2.5 Applications of CFRP in Subsea Pipelines

Offshore pipelines experience number of mechanical damage and deterioration including; corrosion (Ellyin et al., 2000; Shouman and Taheri, 2011; Shamsuddoha et al., 2013), structure damage in the form of local buckling caused by external pressure (Mesloh et al., 1976; Kamalarasa and Calladine, 1988; Xue and Fatt, 2001), and excessive conditions generated by earthquakes, mudslides, sea storms, etc. Corrosion is a metallurgical phenomenon that reduces the wall thickness of the pipe. The corroded wall usually reduces the structural integrity of the pipe, and under high internal pressure, failure can result in the form of either leaks or ruptures in the pipe wall (Shamsuddoha et al., 2013). Local buckling, irrespective of how it is introduced to the pipeline, usually will initiate a propagation buckling which can rapidly flatten and destroy significant sections of the pipeline structure. This requires the pipelines to be repaired or replaced to continue operation. The most recent repair
method is by using the fibre reinforced polymer (FRP) (Ellyin et al., 2000; Shouman and Taheri, 2011; Shamsuddoha et al., 2013) and the extent of the propagating buckle is usually limited by the periodic placement of buckle arrestors along the length of the line as explained in the previous section.

Due to its excellent properties, such as high specific strength and stiffness, corrosion resistance, performance to weight ratio and thermal stability (Smith and Clough, 2010; Wonderly et al., 2005; Keller et al. 2013; Goertzen and Kessler, 2007; Sen and Mullins, 2007), the carbon fibre reinforced polymer (CFRP) is primarily preferred for repairing metallic components and tubular pipes. Previous researches have already proven the effectiveness of using the CFRP to improve the corrosion and mechanical damage of the pipelines (Shamsuddoha et al., 2013; Duell et al., 2008; Seica and Packer, 2007). CFRP composites are increasingly being used in the strengthening, repairing and retrofitting of any existing structures. The strength or stiffness of any structure member can be increased by bonding a thin CFRP plate to the external surface (Zhang and Teng, 2010). One of the recent applications for CFRPs in the oil and gas industry is the rehabilitation of the subsea pipelines. Therefore, using these materials especially for repairing the defect pipelines can offer unique advantages over the conventional repair techniques (Cercone and Lockwood, 2005; Esmaeel et al., 2012; Duell et al., 2008). On the other hand, Sharma et al., 2015 have stated that the CFRP is electrically conductive, which may cause corrosion, especially in metallic components and tubular pipes. Also, in their study mentioned that GFRP wrapped samples exhibited higher electrical resistance than CFRP samples and hence, glass fibres impede corrosion better than carbon fibres. Therefore, more future research needs to undertaking to investigate the conductivity issue for CFRP.

An industry study (Koch et al., 2001) shows that CFRP repair systems offer an economy solution on average 24% cheaper than welded steel sleeve repairs and 73% less expensive than substituting the damaged section of the steel pipelines. Moreover, the CFRP repair system can offer numerous advantages such as eliminating the risk of explosion due to welding, preventing the pipeline shut down during the repair and simplicity of application due to the lightweight of CFRPs. Therefore, the CFRP repair system was successfully applied to offshore pipelines to avoid the costs and dangers of underwater welding or suspending
production (Green, 2010). Another recent application of CFRP for repairing the damaged pipeline was reported by Lukacs et al. (Lukacs et al., 2010; Lukacs et al., 2011). Based on the experimental and numerical investigations, their studies show how using CFRP in repairing the subsea pipelines can enhance the capacity of the damaged sections in carrying compression, bending, tension and torsional loads, in both quasi-static and cyclic loading scenarios. Therefore, the applications of CFRP for strengthening and repairing of onshore and offshore oil and gas pipelines are growing quickly, and the extent and depth of research on their potential future applications and design methods are growing similar as well.

In most of the previous studies conducted on offshore pipeline repair (Meniconi, L. et al. 2002; Freire, J. et al. 2007; Kessler et al. 2004), only considered the internal pressure conditions to study the effectiveness of the CFRP repair system. However, in the real scenario, the external pressure conditions combined with internal pressure might have significant effects on CFRP behaviour. Therefore, Shouman and Taheri (2009) examined experimentally and numerically the behaviour of pipelines with FRP repair system subjected to combined loading conditions (internal and external pressures). Their FRP repair system comprised of eight-layer unidirectional carbon and E-glass fibres and epoxy resin wrapped around the pipe in the hoop direction, pipes had external defects in different dimensions. The repaired pipe was found to buckle at a location adjacent to the FRP wrap. Also, their FE study concluded that the maximum value for strain occurred in the unrepaired section of the pipeline rather than in the post-repaired part. Their parametric study has shown that increasing the wrap thickness will not improve the pipe strength, particularly in the axial direction however prevented the pipe yielding in the defect section. Also, by increasing the repair (wrap) length, the axial performance of the repaired pipe will be improved.

Toutanji and Dempsey (Toutanji and Dempsey, 2001) showed that carbon fibre reinforced polymers (CFRPs) performed much better than composites reinforced with glass (GFRP) or aramid (AFRP) fibres in that, they enhanced the ultimate internal pressure capacity of pipes at different levels as shown in Figure 2.14, as well as improving the durability, strength, and corrosion properties of the pipeline. Moreover, carbon fibres exhibit high strength, and stiffness, low density and superior fatigue performance based on the recent studies
(Elanchezhian et al., 2014; Wonderly et al., 2005). Therefore, the carbon fibres were found superior to glass fibres in these studies and recommended for subsea pipeline repairs.

![Comparison between different repair systems and internal pressure capacity.](image)

**Figure 2-14:** Comparison between different repair systems and internal pressure capacity.

(Toutanji and Dempsey, 2001).

Previous studies showed that there is a general lack of parametric studies related to the types of composite repair system, manufacturing techniques and its laminate orientation. Fibre orientation can determine the required directional strength and stiffness for any particular application. Alexander and Ochoa (2010) utilised axially oriented pre-cured half shells with circumferential carbon and E-glass wraps for reinstating capacity to offshore pipelines and risers. Shouman and Taheri (2011) employed a hoop oriented unidirectional glass epoxy in their numerical modelling and experimental work of the composite repair system. Toutanji and Dempsey (Toutanji and Dempsey, 2001) utilised a filament wounded unidirectional
carbon fibre reinforced composite in the hoop direction as a repair solution for damaged pipelines.

Application of the unidirectional laminates provides strength and stiffness for the structural element in the required direction, which offers better control over the desired strength and modulus. When the pipeline is subjected to a combination of hoop, axial and bending load conditions, it tends to approach failure strains at a much lower loading compared with the load being applied individually and separately. Chan, et al. (Chan et al., 2014) found in their study, that the addition of off-axis plies can affect the final properties of the repaired offshore pipeline, also, found that laminates with \([90^\circ/\pm30^\circ]\) and \([90^\circ/\pm45^\circ/0^\circ]\) orientations provided sufficient strength rehabilitation to the pipelines in both hoop and axial directions. On the other hand, other researchers studied the collapse and buckling behaviour of the composite pipeline under external pressure. Their studies show that the fibre orientation angle plays an important role in the buckling pressure capacity and they found that composite pipeline made with fibre orientation between \(40^\circ\) to \(50^\circ\) provides much higher buckling pressure capacity.

With regards to the external hydrostatic pressure in the offshore pipeline, hoop direction and axial direction stress components cause the propagation buckling along the pipeline. By wrapping the CFRP around the circumference of the pipe in different directions, this will allow strengthening the pipe structural against the propagation buckling and increase the buckling capacity. Unlike the repairing applications of CFRP in the offshore pipeline, there are no studies used the CFRP for arresting the propagation buckling in the offshore pipeline. Therefore, by combining the literature from conventional buckle arrestors and CFRP application in subsea pipeline, a new buckle arrestor is proposed for subsea pipelines by using carbon fibre reinforced polymer (CFRP), in order to provide high arresting efficiency and solve the limitations of existing buckle arrestors. It should be noted that the CFRP arrestor is a new buckle arrestor proposed to subsea industries, therefore, feasibility, suitability and efficiency studies should be done first on single pipe system in order to understand the arresting mechanisms for CFRP arrestors, before expanding the application to the pipe-in-pipe system.
2.6 Literature Review Findings

This chapter presented a detailed review of the analytical, experimental and numerical investigations into the propagation buckling failure of subsea pipe-in-pipe systems as well as the existing buckle arrestors and applications of CFRP in subsea pipelines. Most significant points are:

- Propagation buckling due to external pressure is the most critical failure affecting the structural integrity of subsea pipelines. At pressure level much lower than collapse pressure, the buckle can propagate along the pipeline.

- Collapse under external pressure is a major design concern for subsea pipelines. The local collapse capacity of the pipe can reduce due to the damage in the pipe-wall (in the form of dents or corrosion, etc.) and can initiate the buckle propagation. Pipe-in-pipe systems are broadly used in offshore pipeline applications especially in deep and ultra-deep waters in which thermal insulation of the line is necessary. The structural integrity of the PIP system in the event of an accidental collapse of the carrier pipe is an issue of concern for pipeline designers.

- Despite extensive investigations performed on the integrity of single pipeline systems, to date, PIP instabilities have only been marginally addressed. Previous studies have only focused on propagation buckling of PIP systems having carrier pipes with low $D/t$ values.

- Numerous analytical solutions have been suggested in the literature to estimate the propagation pressure of a single pipe. However, only one analytical solution was proposed in (Kyriakides and Vogler, 2002) for PIP propagation pressure ($P_{P2}$) and showed underestimation for $P_{P2}$.

- Kyriakides (Kyriakides, 2002) performed an experimental study to predict the propagation buckling pressure of stainless steel PIPs with carrier pipes having $D/t$ values of 24.1, 21.1 and 16.7 and inner pipes of various $D/t$ ratios ranging between 15 to 37. Based on his experimental study, an empirical expression for propagation buckling pressure, $P_{P2}$, of PIPs was proposed. In addition to that, an extensive finite
element study of propagation buckling pressure of PIPs using carrier pipes with $D/t$ of 25, 20 and 15 and inner pipes having $D/t$ of 15 and 20 was conducted and presented by Gong and Li (Gong and Li, 2015) and another empirical expression for propagation buckling pressure of PIPs was proposed as well. However, both studies covered a similar $D/t$ range of the carrier pipes, the empirical expressions suggested in those studies are different.

- Unlike the single pipeline system, collapse mechanisms of PIPs have only been marginally addressed (Kyriakides, 2002; Kyriakides and Vogler, 2002; Gong and Li, 2015; Li, 2018). Moreover, these studies have been purely focused on the propagation buckling pressure ($P_{F2}$) of the PIP systems. The existing knowledge on buckling of single-walled pipelines under external pressure can be used to predict the collapse pressure of the outer pipe of a PIP system. However, the buckling mechanisms of the inner pipe and its collapse pressure are different from those of a single pipeline. There is no existing study on the collapse of the inner pipe of a PIP system under external pressure.

- Designing subsea pipelines based on the propagation pressure is overly conservative and impractical. The preferred alternative is to design based on the collapse pressure and install buckle arrestors at certain intervals along the pipeline. However, the existing buckle arrestors have shown certain limitations in practice as shown in the previous literature.

- The CFRP buckle arrestors can be a promising candidate to replace the conventional buckle arrestors for single subsea pipelines and pipe-in-pipe systems, due to their simplicity of application, lightness and durability, etc.
Statement of contribution to co-authored published paper

This chapter includes a co-authored and peer-reviewed paper. The bibliographic details of the co-authored paper, including all authors, are:


The paper has been reformatted to meet the guidelines of the thesis. Minor explanation has been added in the paper for further clarity.

My contribution to this paper involved: literature review, experimental works, numerical modelling, result analysis, and editing.

(Signed) _________________________________ Date: 23.01.2020
PhD candidate: Mahmoud Alrsai.

(Countersigned) ___________________________ Date: 23.01.2020
Principal Supervisor: Dr Hassan Karampour.

(Countersigned) ___________________________ Date: 23.01.2020
Associate Supervisor: Dr Sanaul Chowdhury.

(Countersigned) ___________________________ Date: 23.01.2020
External Supervisor: Professor Faris Albermani.
Propagation Buckling in Subsea Pipe-in-pipe Systems

Abstract:

This chapter investigates propagation buckling of subsea pipe-in-pipe (PIP) systems under hydrostatic pressure. Unlike in previous studies, PIP systems consisting of carrier pipes with a diameter-to-thickness ($D_o/t_o$) ratio in the range 26-40 are examined here. Experimental results from ring squash tests (RSTs), confined ring squash tests (CRSTs), and hyperbaric chamber tests are presented and compared with a modified two-dimensional (2D) analytical solution and with numerical results using three-dimensional (3D) finite-element (FE) analysis. The comparison indicates that the proposed modified analytical expression provides a more accurate lower-bound estimate of the propagation buckling pressure of PIP systems compared with the existing equations, especially for higher $D_o/t_o$ ratios. The novel RST and CRST protocols proposed for PIP systems give lower-bound estimates of the propagation pressure. The FE analysis outcomes demonstrate that the lengths of PIP system transition zones are almost twice the corresponding lengths in single pipes. New modes of buckling are discovered in the hyperbaric chamber tests of PIP systems with $D_o/t_o = 26$.

Keywords:

Pipe-in-pipe systems; Propagation buckling; Confined buckling; Ring squash test.

3.1 Introduction

Pipe-in-pipe (PIP) systems are being used extensively in the design of high-pressure and high-temperature (HP/HT) flowlines because of their outstanding thermal insulation. A typical PIP system consists of concentric inner and outer pipes, bulkheads, and centralizers. The inner pipe (flowline) conveys production fluids, and the outer pipe (carrier pipe) protects the system from external pressure and mechanical damage. These two pipes are isolated by centralizers at joints and connected through bulkheads at both ends of the pipeline. The annulus (the space between the tubes) is either empty or filled with non-structural insulation material such as foam or water (Bai and Bai, 2005).
Pipe-in-Pipe systems (Figure 3.1) are normally divided into two categories: compliant and noncompliant. In a compliant system, the inner pipe and the carrier pipe are attached at close intervals either every two pipe joints by tulip or by donut plate (Bokaian, 2004); in a noncompliant system, the pipes are connected only through bulkheads at discrete locations. The relative movement between the inner and outer pipes is arrested in a compliant system, whereas the two pipes can move relative to each other in a noncompliant system. Pipe-in-Pipe systems are exploited in subsea developments, where the carrier pipe is designed to resist high hydrostatic pressures (water depths up to 3,000 m) and the inner pipe is designed to transmit hydrocarbons at temperatures as high as 180°C (Jukes et al., 2009). The HP/HT flow can cause global upheaval (Wang et al., 2015) or lateral buckling (Vaz and Patel, 1999) in the system. Furthermore, high hydrostatic pressure may trigger a local collapse in the carrier pipe, such as propagation buckling or buckle interaction (Karampour et al., 2013a, b; Karampour and Albermani, 2014, 2015; Karampour et al., 2016). The structural integrity of the PIP system under external pressure is an issue of concern because the collapse of the carrier pipe may result in the collapse of the inner pipe.

In a single pipeline under external pressure, a local dent or ovalization in the pipe wall can initiate a buckle that rapidly transforms the pipe cross-section into a dog-bone shape. The buckle then travels along the pipeline as long as the external pressure is high enough to sustain propagation. The lowest pressure required to perpetuate the buckle is termed propagation pressure, $P_p$, which is only a fraction of the buckle initiation pressure. The collapse and propagation of buckling in single pipelines have been extensively investigated using analytical, experimental, and numerical methods. Most notable are the early analytical studies by Mesloh et al. (1973) and Palmer and Martin (1975) and the experimental and numerical investigations by Kyriakides and Babcock (1981) and Albermani et al. (2011). Recently Karampour et al. (2013b) and Karampour and Albermani (2014) investigated the possible interaction between global buckling of water pipelines, such as upheaval and lateral buckling (Karampour et al., 2013a), and propagation buckling. They suggested a novel design for ultradeep pipelines (Karampour and Albermani, 2015; Karampour et al., 2016) that increases the propagation buckling capacity of the pipeline without increasing wall thickness.
Despite extensive investigations performed on the integrity of single pipelines, to date, PIP instabilities have only been marginally addressed. Kyriakides (2002) conducted a thorough experimental study on propagation buckling of steel PIP systems with 2-in.-diameter carrier tubes with diameter-to-thickness \((D_o/t_o)\) ratio values of 24.1, 21.1, and 16.7 and inner pipes with various diameter-to-thickness \((D_i/t_i)\) ratio values ranging between 15 and 37. Kyriakides (2002) observed two dominant modes of buckling.
In the first mode, the local collapse of the outer pipe led to simultaneous collapse of the inner pipe, whereas in the second mode the carrier pipe collapsed without affecting the inner pipe. Based on their experimental study and three-dimensional (3D) finite element (FE) analyses, Kyriakides and Vogler (2002) suggested an empirical formula for PIP propagation buckling pressure, $P_{p2}$. Gong and Li (2015) carried out a finite element study of propagation buckling of PIP systems with carrier pipes having $D_o/t_o$ values of 25, 20, and 15 and inner tubes having $D_i/t_i$ values of 15 and 20. Although both studies (Gong and Li 2015; Kyriakides and Vogler 2002) covered similar carrier pipe $D_o/t_o$ ranges, two different empirical expressions were suggested.

Previous studies have focused on propagation buckling of PIP systems having carrier pipes with low $D/t$ values. For a subsea pipeline, the use of buckle arrestors (Lee and Kyriakides, 2004) is more economical than the use of a carrier pipe with a thick wall (low $D_o/t_o$ value) to resist propagation buckling. The self-weight of the PIP system has to be kept low to ensure that the pipeline is installable. Increasing the wall thickness of the carrier pipe also significantly amplifies the axial force developed in the PIP system due to the internal pressure and temperature which in turn raises the risk of upheaval and lateral buckling in the system (Bokaian, 2004). To address the aforementioned issues, this chapter aims to investigate the collapse modes and propagation buckling pressures of PIP systems (see Table 3.1) with carrier pipes having higher $D_o/t_o$ values than those in Gong and Li (2015) and Kyriakides and Vogler (2002).

A lower-bound analytical solution for PIP system propagation pressure is proposed herein. Hyperbaric chamber tests are conducted on 1.6-m aluminum (Al-6060-T5) PIP systems. Ring squash test (RST) and confined ring squash test (CRST) protocols are proposed to provide estimates of propagation pressure in the PIP systems with respect to buckling modes observed in the experiments. The chapter concludes with a discussion of the FE results with emphasis on failure modes.

### 3.2 Analytical Solution of Propagation Pressure

Numerous analytical solutions have been suggested to estimate the propagation pressure of a single pipe. Unlike propagation pressure, initiation pressure is very sensitive to initial
imperfections such as local dents or ovalizations. Propagation pressure is related to the plastic properties of the pipe and is only a fraction of the buckle initiation pressure. Both buckle initiation pressure and propagation buckling pressure are related to the pipe’s ratio of diameter to wall thickness; however, previous studies have suggested that there is no evident relationship between the two (Albermani et al., 2011; Mesloh et al., 1973). The simplest propagation pressure model was established by Palmer and Martin (1975), which only considered the initial and final configurations of the pipe cross-section. Figure 3.2(a) shows the four plastic hinges developed in the pipe at different stages of propagation buckling. Palmer and Martin assumed a rigid–perfectly plastic material with yield stress $\sigma_{y0}$. By equating the plastic work expanded in the four hinges (internal work) to the work done by hydrostatic pressure due to the change in pipe volume, they found the following expression for single-pipe propagation pressure ($P_p$):

$$P_p^{PM} = \pi \sigma_y \left( \frac{t}{D} \right)^2$$

(3.1)

where $D$ and $t = \text{outside diameter and wall thickness of the carrier pipe, respectively.}$. Experimental studies (Albermani et al., 2011; Kyriakides and Babcock, 1981) showed that Eq. (3.1) underestimates the propagation pressure of the pipeline, specifically at lower $D/t$ values. Considering the collapse under the plane strain condition, propagation buckling of a single pipe can be expressed as (Chater and Hutchinson, 1984; Kyriakides et al., 1984)

$$\hat{P}_p = \frac{2\pi}{\sqrt{3}} \sigma_y \left( \frac{t}{D} \right)^2$$

(3.2)

Kyriakides and Vogler (2002) investigated the collapse of PIP systems using the plane strain condition and adopting a four-hinge mechanism in the carrier pipe and in the inner pipes. They considered the strain-hardening behaviour in the material model of the plastic hinges and proposed the following expression for PIP propagation pressure:

$$\hat{P}_{p2} = \frac{2\pi}{\sqrt{3}} \sigma_{y0} \left( \frac{t_o}{D_o} \right)^2 \left[ 1 + \frac{\sigma_{yi}}{\sigma_{y0}} \left( \frac{t}{t_o} \right)^2 \right]$$

(3.3)
where subscripts \( o \) and \( i \) = outer pipe and inner pipe, respectively. Albermani et al. (2011) proposed a modification to the lower-bound Palmer and Martin (1975) solution by accounting for both the circumferential membrane and the flexural effects in the pipe wall

\[
W_{ex} = (W_{in})_f + (W_{in})_m
\]

(3.4)

where \( W_{ex} \) = external work done by the net hydrostatic pressure; and \( W_{in} \) = internal work due to the circumferential flexure, \( f \), and membrane, \( m \), effects. The initially circular cross-section of the pipe (Figure 3.2(a)) deforms into a dog-bone (Figure 3.2(b)) and eventually a nearly flat segment (Figure 3.2(c)). Accordingly, Eq. (3.4) can be written as

\[
\bar{P}_p (\Delta A) = 3\pi m_p + (pr)(\Delta l)
\]

(3.5a)

where \( \Delta A \) = change in cross-sectional area (Figure 3.2(a–c)); \( \Delta l \) = change in circumferential length; and \( m_p \) = plastic moment; these are given by

\[
\Delta A = \pi r^2
\]

(3.5b)

\[
\Delta l = 2\pi r - 4r\sqrt{2} = 0.626r
\]

(3.5c)

\[
m_p = \sigma_y \frac{t^3}{4}
\]

(3.5d)

Substituting Eqs. (3.5b) - (3.5d) into Eq. (3.5a), the propagation pressure, \( \bar{P}_p \), is obtained as

\[
\bar{P}_p = \frac{3}{2.515} \left[ \pi \sigma_y \left( \frac{t}{D} \right)^2 \right]^{0.5}
\]

(3.6)

The values of \( \hat{P}_p \) and \( \bar{P}_p \) suggest a 15 and a 19% increase, respectively, in the propagation pressure of the single pipe compared with Palmer and Martin’s (1975) expression, \( \bar{P}_p \), regardless of the \( D/t \) value of the pipe. The analytical lower-bound solution to propagation buckling of a single pipe given by Eq. (3.6) can be extended to pipe-in-pipe systems by accounting for the membrane and flexural effects of the outer and the inner pipes.
where $W_{ex}$ = external work done by hydrostatic pressure; and $W_{in}$ = internal work due to circumferential flexure, $f$, and membrane, $m$, effects. Based on experimental observations from the hyperbaric chamber and ring squash tests, the initially circular cross-section of the outer pipe (Figure 3.2(d)) has been shown to deform into the shape in Figure 3.2(e). Further increase in external pressure causes the PIP system to eventually deform into the dog-bone shape (Figure 3.2(f)). Thus Eq. (3.7) can be written as

$$\tilde{P}_{p2} = 3\pi \left( m_{po} + m_{pi} \right) + \tilde{P}_{p2} \left( r_o \Delta l_o + r_i \Delta l_i \right)$$  \hspace{1cm} (3.8a)

where $\Delta A = \text{change in cross-sectional area}$; $\Delta l = \text{change in circumferential length}$; and $m_p = \text{plastic moment}$ (Albermani et al., 2011); these are given by

$$\Delta A = \pi r_o^2$$  \hspace{1cm} (3.8b)

$$\Delta l_o = 0.626 r_o; \quad \Delta l_i = 0.626 r_i$$  \hspace{1cm} (3.8c)

$$m_{po} = \sigma_{yo} \frac{t_o^2}{4}; \quad m_{pi} = \sigma_{yi} \frac{t_i^2}{4}$$  \hspace{1cm} (3.8d)

where subscript $o = \text{outer pipe}$; and subscript $i = \text{inner pipe}$. Substituting Eqs. (3.8b) - (3.8d) into Eq. (3.8a), the propagation pressure, $\tilde{P}_{p2}$, of the PIP system is obtained as

$$\tilde{P}_{p2} = \left[ \frac{3\pi}{2.515} \sigma_{yo} \left( \frac{t_o}{D_o} \right)^2 \right] \left[ 1 + \frac{\sigma_{yi}}{\sigma_{yo}} \left( \frac{t_i}{t_o} \right)^2 \right] \left[ \frac{1}{1 - \left( \frac{D_i}{2D_o} \right)^2} \right]$$  \hspace{1cm} (3.9)

When $D_i = t_i = 0$, Eq. (3.9) yields the propagation pressure of a single pipe as given by Eq. (3.6). Unlike Eq. (3.3), Eq. (3.9) accounts for the effect of $D/D_o$ along with that of $t_i/t_o$ and $\sigma_{yi}/\sigma_{yo}$. Propagation buckling pressures of single pipes (outer pipes) and PIP systems determined from the analytical expressions in Eqs. (3.2), (3.3), (3.6), and (3.9) are listed in Table 3.2 and compared with the experimental results ($P_\rho$ for a single pipe and $P_{p2}$ for a PIP system) obtained from the hyperbaric chamber tests.
3.3 Hyperbaric Chamber Tests

The experimental protocol comprises end-sealing concentric PIP systems, with parameters given in Table 3.1 and a length of 1.6 m (L/D > 20), pressurised inside the hyperbaric chamber shown in Figure 3.3. The chamber has an inner diameter of 173 mm and a length of 4 m and is rated for a working pressure of 20 MPa (2,000-m water depth). The intact PIP system was sealed at both ends by gluing on thick aluminum disks to ensure that the inner pipe was completely sealed from the outer pipe. Two valves were connected to each end of the PIP system, one on the carrier pipe and the other on the inner pipe. One valve was used for bleeding the pipe while filling it with water. The second valve was used to vent the carrier and inner pipes and to collect water from the inner pipe and the cavity between the inner and outer pipes during propagation buckling [through the hoses shown in Figure 3.3(b)]. Volume-controlled pressurization with a high-pressure pump [shown in Figure 3.3(a)] was used, and the pressure was increased until collapse of the system due to external pressure under quasi-static steady-state conditions. By maintaining a low rate of pumping, the chamber pressure was stabilized at propagation pressure, $P_{p2}$, with buckling longitudinally propagating along
the PIP system sample accompanied by water flow from the vents. The change in volume of
the system \((\Delta V)\) during the test was calculated by measuring the weight of water being
discharged from the inner pipe and the cavity between the pipes separately using the digital
weighing scales shown in Figure 3.3(a). Control tests using a single pipe (outer pipe) were
c Conducted first.

Table 3.1: Properties of PIP Systems.

<table>
<thead>
<tr>
<th>Identifier</th>
<th>Carrier pipe</th>
<th>Inner pipe</th>
<th>(D_o/t_o)</th>
<th>(D_i/t_i)</th>
<th>(D_o/D_o)</th>
<th>(t_i/t_o)</th>
<th>(E) (MPa)</th>
<th>(E' ) (MPa)</th>
<th>(\sigma_{yo}) (MPa)</th>
<th>(\sigma_{yi}) (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PIP-1</td>
<td>OD= 80, t= 2</td>
<td>OD= 40, t= 1.6</td>
<td>40.0</td>
<td>25.0</td>
<td>0.50</td>
<td>0.80</td>
<td>69,000</td>
<td>1.01</td>
<td>169</td>
<td>0.93</td>
</tr>
<tr>
<td>PIP-2</td>
<td>OD= 60, t= 2</td>
<td>OD= 40, t= 1.6</td>
<td>30.0</td>
<td>25.0</td>
<td>0.75</td>
<td>0.80</td>
<td>69,000</td>
<td>0.97</td>
<td>139</td>
<td>1.12</td>
</tr>
<tr>
<td>PIP-3</td>
<td>OD= 80, t= 3</td>
<td>OD= 40, t= 1.6</td>
<td>26.7</td>
<td>25.0</td>
<td>0.50</td>
<td>0.53</td>
<td>69,000</td>
<td>1.02</td>
<td>209</td>
<td>0.75</td>
</tr>
</tbody>
</table>

Note: All dimensions are in millimetres; \(OD = \) outer diameter; \(t = \) thickness.

Figure 3-3: Experimental setup at Griffith University: (a) hyperbaric chamber, high-pressure pump, scales, pressure gauge, and vents; (b) pipes and fittings.

Figures 3.4–3.6 show the experimental results of the propagation buckling response. The pressure inside the chamber is plotted against the normalized change in volume of the carrier
pipe (60 × 2 mm) of PIP-2 in Figure 3.4(a). The chamber is gradually pressurized until the initiation pressure, \(P_I\), is reached at which a section of the pipe collapses, resulting in a drastic drop in the chamber’s pressure. The pressure is then maintained at the propagation pressure, \(P_p\), with the dog-bone buckle shape, longitudinally propagating along the length of the pipe. The propagation buckling response of the PIP-2 system is shown in Figure 3.4(b). The change in system pressure is plotted against the normalized change in volume of the inner pipe, 40 × 1.6 mm, and the outer pipe, 60 × 2 mm (the space between the two pipes). Buckle is initiated first \((P_{I2})\) on the outer pipe; then the energy is released through ovalization of the outer pipe until the outer pipe touches the inner pipe.

Buckle initiation pressures, \(P_I\) and \(P_{I2}\), have been shown to be closely related to geometric imperfections in the form of dents or ovality of the outer pipe (Karampour and Albermani, 2014; Zheng et al., 2014). Because the main focus of this chapter is propagation buckling pressures, the parameters affecting buckle initiation pressure are not discussed. Following the contact between the carrier pipe and the inner pipes of PIP-2, the inner pipe collapses, and the buckle propagates longitudinally as long as the pressure is maintained at \(P_{p2}\). When the stiff end caps fall within the vicinity of the buckle transition zone, higher pressure is required to perpetuate the buckle which corresponds to the stiffening part of the PIP-2 response shown in Figure 3.4(b). A dog-bone buckle shape similar to that observed in carrier pipe propagation buckling (Figure 3.4(a)) was detected in the PIP-2 hyperbaric chamber test (Figure 3.4(b)). Changes in volume of the outer and inner pipes are plotted against the test time in Figure 3.4(c). The time history shows an initial discharge from the outer pipe that is higher than that from the inner pipe. However, after the outer pipe touches the inner pipe at \(\Delta V/V_o = 0.1\) (shown in Figure 3.4(b)), discharge from the inner pipe is triggered and at \(\Delta V/V_o > 0.2\) (shown in Figure 3.4(c)) the discharge rate of the inner pipe exceeds that of the outer pipe. This shows that the collapse of the outer pipe is rapidly transferred to the inner pipe and is then followed by the longitudinal propagation of the buckle in both carrier and inner pipes. The rate of discharge in the carrier pipe and the inner pipe gradually decays as time lapses because of the introduction of the end caps in the buckle zone.

The hyperbaric chamber propagation buckling results for the 80 × 2-mm carrier pipe and the PIP-1 system are shown in Figure 3.5. A small dent was made in the PIP-1 carrier pipe in the
single-pipe test, which explains the lower buckle initiation pressure of the carrier pipe compared with that of PIP-2. As shown in Figure 3.5(b) following the collapse of the carrier pipe the pressure inside the chamber drops drastically until the carrier and inner pipes come into contact. Subsequently, a dog-bone buckle shape propagates along the PIP while the pressure is maintained at $P_{p2}$. Hyperbaric chamber tests of PIP-1 and PIP-2 were repeated twice each, and no significant disparities were observed in the results.

Figure 3-4: Propagation buckling response inside the hyperbaric chamber: (a) pressure versus normalized change in volume of the 60 × 2-mm carrier pipe; (b) pressure versus normalized change in volume of PIP-2; (c) normalized volume versus time for PIP-2.
Results for PIP-3 with $D_o/t_o = 26.7$ from three hyperbaric chamber tests are shown in Figure 3.6. Unlike the responses of PIP-1 and PIP-2, three distinctive modes of buckling were observed in PIP-3: (1) the dog-bone buckle shape (flat mode), shown in Figure 3.6(a); (2) the confined buckle shape (U mode), shown in Figure 3.6(b); and (3) a combination of dog-bone and U-shaped buckle, shown in Figure 3.6(c). The dog-bone mode of buckling is similar to the responses observed in PIP systems with high $D_o/t_o$ values (PIP-1 and PIP-2). In this mode of failure, PIP-3 remains straight after failure and a flat mode of buckling propagates through
its length; however, the deformed shape of the inner pipe is not symmetric in the cross section (shown in Figure 3.6(a)).

In the second hyperbaric chamber test of PIP-3, shown in Figure 3.6(b), a confined buckle shape is observed. The confined buckle mode is propagated along the length of the PIP while the pressure in the chamber is escalated followed by rapid discharge of water from the outer and inner pipes. This U-shape buckling mode was previously observed in confined-buckling tests of steel and aluminum tubes (Lee and Kyriakides, 2004; Stephan et al., 2016). Stephan et al. (2016) performed an experimental investigation of the collapse of 3m long aluminum pipes inserted in a 2m long confining steel pipe. They observed a flat mode (dog-bone buckle shape) in the unconfined section of the aluminum pipe and a U-mode buckle shape in the confined section. Their experiments showed that within the studied range \(16 < D/t < 48\), the confined buckle shape consistently propagated at higher pressure compared with the dog-bone buckle shape. However, a comparison of Figures 3.6(a and b) shows that in PIP-3 the U-shape buckling propagation \(P_{p2} = 1,820\) kPa is initiated at a slightly lower pressure than the propagation pressure of the dog-bone buckle shape \(P_{p2} = 2,044\) kPa. In the third test, PIP-3 showed a dog-bone failure mode that had flipped into U mode. The average \(P_{p2}\) results from the hyperbaric chamber tests are summarized in Table 3.2.

### Table 3.2 Comparison of hyperbaric chamber, analytical, RST and FE results.

<table>
<thead>
<tr>
<th>Identifier</th>
<th>Hyperbaric Chamber</th>
<th>Analytical</th>
<th>RST</th>
<th>FE</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(P_p) (kPa)</td>
<td>(P_{p2}) (kPa)</td>
<td>(\hat{P}_p)</td>
<td>(\hat{P}_{p2})</td>
</tr>
<tr>
<td>PIP-1</td>
<td>700</td>
<td>780</td>
<td>0.55</td>
<td>0.78</td>
</tr>
<tr>
<td>PIP-2</td>
<td>900</td>
<td>1,620</td>
<td>0.62</td>
<td>0.59</td>
</tr>
<tr>
<td>PIP-3</td>
<td>1,400</td>
<td>2,020(^a)</td>
<td>0.76</td>
<td>0.64</td>
</tr>
</tbody>
</table>

\(^a\) Corresponds to dog-bone buckle shape shown in Figure 3.6(a).

\(^b\) Dog-bone buckle shape.

\(^c\) U-shape buckle.
Figure 3-6: Propagation buckling response inside the hyperbaric chamber: (a) propagation buckling response of PIP-3 with dog-bone buckle shape; (b) propagation buckling response of PIP-3 with confined buckle shape; (c) propagation buckling response of PIP-3 showing the interaction between dog-bone and confined buckle shape.
3.4 Ring Squash Tests

Previous studies (Albermani et al., 2011; Kamalarasa and Calladine, 1988) have shown that the RST is a satisfactory approach that gives a lower-bound estimate of propagation buckling pressure in single pipelines. The RST is conducted on a ring cut from the pipe specimen in such a way as to produce the dog-bone shape of the deformed pipe observed in the hyperbaric chamber. Figure 3.7 shows the RST setup for a single pipe (60 × 2 mm). In this test, carried out in a universal testing machine, a short segment of the pipe with a length \( l = 150 \text{ mm} \) (approximately \( 2.5D \)) is squashed between two rigid cylinders of the same diameter and length as those of the pipe being tested. The two repeated tests are referred to as Tests 1 and 2 (Figure 3.7(a)). The force, \( F_o \), required to compress the ring is plotted against the deformation of the pipe right under the load (\( \Delta \)) in Figure 3.7(a). The total energy dissipated in the RST process, \( U \), can be evaluated by calculating the area under the force-displacement curve shown in Figure 3.7(a). The pressure \( (P_{RST}) \) associated with the energy required for plastic deformation of the ring can be calculated from the following equation:

\[
P_{RST} = \frac{U}{l \times \Delta A}
\]

(3.10)

where \( \Delta A = \text{difference in area between the original circular shape and the final dog-bone configuration.} \)

In this chapter, the RST has been modified to determine the propagation pressure of the PIP system. Short segments of the outer and inner pipes with lengths of \( 2.5D_o \) \((l = 150 \text{ mm})\) are cut from the PIP specimen. The two pipes are held concentric during the test using soft foam in the space between them. The setup and the results of the RST for PIP-2 are shown in Figure 3.8. The \( F-\Delta \) relationships for the two repeated tests (Tests 1 and 2) agree well, as evident in Figure 3.8(a). Collapse States 1 and 2 (Figures 3.8(b and c)) correspond to the onset of plastic hinge development in the outer pipe and inner pipe, respectively. At the ultimate collapse state (State 3) shown in Figure 3.8(d), four plastic hinges are developed in each pipe. Eq. (3.10) is used to determine the RST pressure of the PIP system \((P_{RST}^{PIP})\). Two ring squash tests were conducted for each PIP system, and the averages of the results from the two tests are
listed in Table 3.2 in form of $P_{p_2}^{RST} / P_{p_2}$. The ring squash test can be used to determine the yield stress, $\sigma_y$, of the pipe using the following expression:

$$\sigma_y = \frac{F_o D}{2Ll^2}$$

(3.11)

where $F_o =$ load level at which the four plastic hinges are shown in Figure 3.7(a) are developed in the pipe wall. The effective yield stress calculated from the ring squash test implicitly accounts for the strain-hardening response of the material. Yield stresses, $\sigma_y$, of the carrier pipes obtained from the ring squash tests, are summarized in Table 3.1.

**Figure 3-7:** (a) Ring squash test results for single pipe (60 × 2 mm); (b) Collapse State 1; (c) Collapse State 2.
A confined-buckling ring squash test (CRST) protocol was proposed by Stephan et al. (2015) to investigate the buckling of a soft pipe encased in a stiff pipe. That procedure is modified here to model the U-shape buckle observed in the hyperbaric chamber tests of PIP-3. In the CRST, a 150 mm long segment of PIP-3 is positioned inside semi-cylindrical solid confinement of diameter $D_o$ and diametrically compressed by a solid cylindrical indenter with a diameter of $(D_o - 2t_o - 2t_i)$. The softening part of the CRST response between Collapse States 1 and 2, shown in Figure 3.9, corresponds to the development of the U-shape buckle in the outer pipe. The drop in response just before the end of Test 2 at Collapse State 3 is due to fracture of the pipe wall at one of the fold lines of the U shape. Results of the ring squash tests

**Figure 3-8:** (a) Ring squash test results for PIP systems (outer pipe: 60 × 2 mm; inner pipe: 40 × 1.6 mm); (b) Collapse State 1; (c) Collapse State 2; (d) Collapse State 3.
of the PIP systems and the confined ring squash test of PIP-3 (U-shape buckle) are summarized in Table 3.2.

![Graph showing force (F) vs. displacement (Δ) for different tests.](image)

**Figure 3-9:** (a) Confined ring squash test results for PIP systems (outer pipe: $80 \times 3$ mm; inner pipe: $40 \times 1.6$ mm; (b) Collapse State 1; (c) Collapse State 2; (d) Collapse State 3.
3.5 Comparison of Analytical Solution and Experimental Results

Table 3.2 compares the analytical and experimental results. Pressures $P_p$ and $P_{p2}$ correspond to the propagation buckling pressures of the single pipe (carrier pipe) and the PIP system, respectively, obtained from the hyperbaric chamber tests. The variables $\hat{P}_p$ and $\hat{P}_{p2}$ refer to the analytical solutions proposed by Kyriakides and Vogler (2002) for the propagation buckling pressure of the single pipe and the PIP system given in Eqs. (3.2) and (3.3), respectively. The analytical expressions derived for propagation buckling pressure of single pipes by Albermani et al. (2011) and PIP systems in the current study are listed in Table 3.2 as $\hat{P}_p$ and $\hat{P}_{p2}$, which correspond to Eqs. (3.6) and (3.9), respectively. The propagation pressures from the RSTs of the PIP systems (and the CRST for PIP-3) are represented as $P_{p2}^{RST}$. To make meaningful comparisons, the analytical and ring squash results are normalized to the hyperbaric chamber results.

As represented in Table 3.2, the analytical solutions provide lower bounds for the propagation buckling of the single pipes and PIP systems. Propagation pressures of single pipes predicted from the analytical solutions, $\hat{P}_p$ and $\hat{P}_{p}$, are from 55 to 79% of the propagation pressure, $P_p$, from the hyperbaric chamber results. In single pipes, the difference between the analytical and hyperbaric chamber results decreases with a corresponding reduction in the $D_o/t_o$ ratio, whereas in PIP systems the difference between the analytical and hyperbaric chamber results increases with a corresponding decrease in the $D_o/t_o$ ratio. Comparison of the analytical and experimental results in Table 3.2 indicates that Eq. (3.9) predicts the propagation buckling of the PIP system better than Eq. (3.3) regardless of the $D_o/t_o$ ratio.

It can be further observed from Table 3.2 that the ratio of propagation pressure obtained from the ring squash tests ($P_{p2}^{RST}$) to those from the hyperbaric chamber tests varies from 0.49 to 0.76 for the PIP systems. Albermani et al. (2011) and Kamalarasa and Calladine (1988) suggested average ring squash–to–hyperbaric chamber pressure ratios of approximately 0.73 for single pipes. It appears that the ring squash test provides a highly conservative prediction of the propagation pressure of PIP systems with low $D_o/t_o$ ratios. Interestingly, the
propagation pressure of PIP-3 from the CRSTs (U-shape buckle) is only 2\% lower than the U-shape propagation pressure from the hyperbaric chamber tests.

3.6 Finite-Element Analysis

3.6.1 Mesh sensitivity and convergence study

There are several options for element types that are available in the ANSYS element library. In this chapter, two shell element types, including four-node SHELL (181) and eight-node SHELL (281) were investigated. The mesh sensitivity and convergence study were implemented in three stages; firstly, the element sizes were determined based on the number of divisions in the circumferential ($mc$) direction and length of element in the longitudinal ($ml$) direction for each shell type. Secondly, by using the element size determined in the first stage, the number of integration points through the thickness was investigated for each shell type. Lastly, the propagation pressure ($P_{p2}$) predicted using these two shell elements is compared with test results for all PIPs in Table 3.1, in order to select the shell type provides more result accuracy and less computational time of the analyse.

The convergence study started by increasing the number of divisions in the circumferential ($mc$) direction and reducing the length of element in the longitudinal ($ml$) direction for each shell type until the propagation pressure ($P_{p2}$) stabilizes in value. Suitable sizes of the finite element mesh for inner and outer pipes were selected depending on the result accuracy and the computational time of the analyses. Sample PIP-3 was chosen to perform this study. Table 3.3 shows the comparison between shell types based on $mc$ and $ml$. Also, Figure 3.10 shows the propagation pressure ($P_{p2}$) versus mesh size ($mc$ and $ml$) for shell 181 and shell 281. It can be seen from the figure and table that the propagation pressure ($P_{p2}$) started to stabilize at $mc = 24$ elements and $ml = 5$mm for both shell types with satisfying computational time.
Table 3.3: Element size comparison between shell types for PIP3.

<table>
<thead>
<tr>
<th>Shell Type</th>
<th>mc (No.)</th>
<th>ml (mm)</th>
<th>$P_{p2}$ (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SHELL 181</td>
<td>6</td>
<td>10</td>
<td>2061</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>7</td>
<td>1952</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>5</td>
<td>1939</td>
</tr>
<tr>
<td></td>
<td>48</td>
<td>3</td>
<td>1940</td>
</tr>
<tr>
<td>SHELL 281</td>
<td>6</td>
<td>10</td>
<td>1990</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>7</td>
<td>1916</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>5</td>
<td>1896</td>
</tr>
<tr>
<td></td>
<td>48</td>
<td>3</td>
<td>1895</td>
</tr>
</tbody>
</table>

Figure 3-10: Propagation pressure ($P_{p2}$) versus mesh size; (a) Shell 181, and (b) Shell 281.
In the second stage of convergence study, the effect of number of integration points through the wall-thickness on the propagation pressure of PIP-3 was investigated for each shell type. The integration points increased for each shell type until the propagation pressure ($P_{p2}$) stabilized. Figure 3.11 illustrates the propagation pressure ($P_{p2}$) versus the number of integration points for shell 181 and shell 281. In this study the element size was selected based on the previous element size study, where $mc = 24$ elements and $ml = 5$mm. It can be seen from the figure that the propagation pressure ($P_{p2}$) started to stabilize at 7 integration points for shell 181, however, for shell 281 the propagation pressure ($P_{p2}$) stabilized at 5 integration points through the thickness.

**Figure 3-11:** Propagation pressure versus the number of integration points in PIP-3; (a) Shell 181, and (b) Shell 281.
Lastly, the propagation pressures ($P_{p2}$) predicted using shell 181 and shell 281 were compared with test results for all PIPs to measure the accuracy for each shell type. It should be noted that the element size and number of integration points used in this study are the same for each shell type. FE models were discretized using $mc = 24$ elements and $ml = 5\,\text{mm}$ with 7 integration points through the thickness of the carrier pipe and the inner pipe. Table 3.4 shows the predicted propagation pressures ($P_{p2}$) using shell 181 and shell 281 and experimental propagation pressures ($P_{p2}$). As shown in the table, shell 281 predictions for PIP-1 and PIP-2 propagation pressure represent 114% and 89%, respectively of the experimental results, which slightly closer to experiments than those predicted by shell 181. However, the propagation pressure obtained from shell 181 for PIP-3 is close to experimental results than shell 281 with less than 3% difference between both types. It worth mentioning that the experimental results for propagation pressures ($P_{p2}$) in Table 3.4 are an average of three repeated tests for each PIPs, reasonable CoVs (shown inside brackets in Table 3.4) were observed for PIP-2 and PIP-3, however, CoV was higher for PIP-1. Overall, it seems that both shell types with comparable element size can provide a good prediction for propagation pressure particularly for PIP-2 and PIP-3. However, the large difference for PIP-1 might be due to inaccuracy test results which is shown with high CoV.

In this chapter, therefore, by considering the results accuracy and computational time for analyses, the carrier pipe and the inner pipe were modelled using four nodes shell element (181) with a fully integrated scheme and seven integration points through the thickness in order to provide high prediction accuracy, also, advanced curved-shell formulation option was selected for shell 181 to account for the curvature of the pipeline.
3.6.2 Finite element models development

Finite-element simulations of 1.6-m-long samples of the PIP systems used in the hyperbaric chamber tests were conducted using ANSYS. Thin four-node shell elements (181) were used to model the carrier pipe and the inner pipe based on mesh sensitivity and convergence study done in the previous section. Frictionless contact and target elements (174 and 170) were used in three pairs to define the nonlinear contact between the carrier and inner pipes and the inner surfaces of the inner pipe wall. Because of symmetry, a one-half model of the pipe wall (180°) was discretized using 24 elements in the circumferential direction with seven integration points through the thickness of the carrier pipe and the inner pipe. To better facilitate the nonlinear analysis, a small ovalization ratio \( \Delta_o \) (Eq. (3.12)) of 0.5% was introduced at midlength on the carrier pipe in the FE model

\[
\Delta_o = \frac{D_{\text{max}} - D_{\text{min}}}{D_{\text{max}} + D_{\text{min}}} \tag{3.12}
\]

where \( D_{\text{max}} \) and \( D_{\text{min}} \) = maximum and minimum outer diameters along the pipe length, respectively. The nodes at either end of the PIP systems were restrained from translation in all directions. A von Mises elastoplastic (bilinear) material definition with isotropic hardening was adopted. The modulus of elasticity, \( E \), and tangent modulus, \( E' \), used in the FE models are also listed in Table 3.1 and are based on the stress-strain curves obtained from the tensile longitudinal coupons taken from the pipe wall, Figure 3.12(a) shows the experimental and FE

<table>
<thead>
<tr>
<th>Shell Type</th>
<th>PIP’s</th>
<th>( P_{p2} ) (kPa)</th>
<th>( P_{p2} ) (Exp.) (kPa)</th>
<th>( P_{p2} ) (FE) / ( P_{p2} ) (Exp.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SHELL 181</td>
<td>PIP-1</td>
<td>998</td>
<td>780 (2.8%)</td>
<td>1.28</td>
</tr>
<tr>
<td></td>
<td>PIP-2</td>
<td>1393</td>
<td>1620 (0.87%)</td>
<td>0.86</td>
</tr>
<tr>
<td></td>
<td>PIP-3</td>
<td>1939</td>
<td>2020 (0.24%)</td>
<td>0.96</td>
</tr>
<tr>
<td>SHELL 281</td>
<td>PIP-1</td>
<td>883</td>
<td>780 (2.8%)</td>
<td>1.14</td>
</tr>
<tr>
<td></td>
<td>PIP-2</td>
<td>1435</td>
<td>1620 (0.87%)</td>
<td>0.89</td>
</tr>
<tr>
<td></td>
<td>PIP-3</td>
<td>1895</td>
<td>2020 (0.24%)</td>
<td>0.94</td>
</tr>
</tbody>
</table>

Table 3.4: Propagation pressure (\( P_{p2} \)) comparison for all PIPs between shell types.
stress-strain curves for outer pipe of PIP-1. The yield stresses used in the FE models are taken
from the ring squash tests using Eq. (3.11) and are listed in Table 3.1 as $\sigma_{yo}$ and $\sigma_{yi}$ for the
outer pipe and inner pipe, respectively. The FE predictions for PIP-2 and PIP-3 propagation
pressure in Table 3.2 represent 86% and 96%, respectively, of the experimental results.
However, the propagation pressure obtained from the FE analysis overestimates the
experimental results for PIP-1.

The pressure response and the deformed shape of PIP-1 and PIP-2 from the FE analyses are
shown in Figures 3.13(b) and 3.14, respectively. The pressure is plotted against the
normalized ovalization of the carrier and inner pipes ($\Delta D/D$). By increasing the hydrostatic
pressure, the carrier pipe of PIP-1 in Figure 3.12(b) gradually deforms from an intact shape
(Stage I) to a deformed shape (Stage II). At this stage, the outer and inner pipes come into
contact, and a small deformation is observed in the inner pipe. The local collapse in the inner
pipe is arrested, which is followed by a slight increase in pressure. The collapse is then
propagated in the outer pipe until detained by the end caps, as depicted in the deformed shape
(Stage III). In the vicinity of the end-caps, higher pressure is required to perpetuate the
collapse in the outer pipe. However, the increase in pressure causes a collapse in the inner
pipe at the pressure level (Stage IV) and initiates a buckle that is propagated through the
length. This mode of buckling, in which the collapse propagates over the inner pipe, was
reported by Gong and Li (2015) and Kyriakides (2002) as occurring in a PIP system where
the inner pipe is stiffer (has a larger thickness and yield stress) than the outer pipe. However,
in the present study, this buckling mode was observed in PIP-1, in which the inner pipe is
softer than the outer pipe.

The buckling response of PIP-2 is shown in Figure 3.13. Following the touchdown (Stage II),
the pressure is slightly increased, and the collapse is propagated in the outer pipe in a length
of approximately $8D$. The inner tube is then collapsed, and propagation buckling is distributed
along the length of PIP-2 (Deformation Stages IV and V). Knowing that the inner pipes in
PIP-1 and PIP-2 are identical, comparison of the buckling responses in Figures 3.13 and 3.14
demonstrates that the collapse pressure of the inner pipe in PIP-2 is significantly higher (68%)
than that of the inner pipe in PIP-1. This difference in collapse pressures of identical inner
pipes in different PIP systems is an important factor in the design of PIP systems where the inner pipe is intended to safely carry the internal fluids.

The pressure response and the deformed shape of PIP-3 from the FE analyses are shown in Figure 3.14. This dog-bone buckle mode is similar to the failure mode shown in Figure 3.6(a), which was obtained from the hyperbaric chamber tests. To explain the U-shape buckling modes observed in the hyperbaric chamber tests of PIP-3, shown in Figures 3.6(b and c), some scenarios were assumed in the FE models. To account for potential manufacturing defects, nonconcentric PIP models were created, and the inner pipe was located 1–5% off-centre from the outer pipe in the horizontal and vertical directions. No significant change in propagation pressure of the eccentric PIP systems was observed, and all FE models exhibited dog-bone failure modes.
Figure 3-12: (a) Experimental and FE stress-strain curves for outer pipe of PIP-1; (b) FE results showing pressure against normalized ovality and corresponding PIP-1 deformed shapes.
Figure 3-13: FE results showing pressure against normalized ovality and corresponding PIP-2 deformed shapes.

To account for the rigid foundation boundary condition in the nonlinear FE model, the nodes at the very bottom of the carrier pipe were constrained against translation along with the left one-third of PIP-3’s length. The FE response of the PIP-3 model is shown in Figure 3.15. propagation buckling is triggered in the middle of PIP-3 and is then propagated in a dog-bone shape to the right of the specimen. The pressure required to propagate this dog-bone buckle mode, shown in Stage III in Figure 3.15, is close to the pressure measured in the hyperbaric chamber test shown in Figure 3.6(c). When the dog-bone buckle mode reaches the end cap at the right end of the PIP system, the U-shape mode is triggered at the left end of PIP-3, as
shown in Stage IV. The U-shape buckle then propagates toward the left end cap and is accompanied by an increase in system pressure, as suggested by the transition from Stage IV to Stage V in Figure 3.15.

Figure 3.14: FE results showing pressure against normalized ovality and corresponding PIP-3 deformed shapes.

To further investigate nonlinear buckling response, another FE model for PIP-3 with different boundary conditions was created. The nodes at the very bottom of the carrier pipe were restrained from translation along the entire length of the PIP system. The FE results are shown in Figure 3.16 and depict a U-shape buckle that is initiated in the carrier pipe and quickly transferred to the inner pipe. The propagation pressure and failure mechanism of PIP-3 from the FE results, shown in Figure 3.16, are similar to the response observed in the hyperbaric
chamber test shown in Figure 3.6(b). The global curvature observed in the inner pipe in Stage V of the FE response was also detected in the failed PIP-3 specimen from the hyperbaric chamber test. It is worth mentioning that the collapse pressure of the inner pipe in the U-shape buckling mode of Figure 3.16 is higher than the corresponding collapse pressures in the dog-bone and flip-flop modes shown in Figures 3.15 and 3.16, respectively.

![Pie chart with annotations](image)

**Figure 3-15:** FE results showing pressure against normalized ovality and corresponding PIP-3 deformed shapes with bottom constraint on a third of the length.
Figure 3-16: FE results showing pressure against normalized ovality and corresponding PIP-3 deformed shapes with bottom constraint along the entire length.
The length of the transition zone in the carrier pipes \((l_t)\) from the finite-element models are listed in Table 3.5. Kamalarasa and Calladine (1988) performed a dimensional analysis and found the following expression for the length of the transition zone in single pipes:

\[
\hat{l}_t = 3.6 \sqrt{\frac{r^3}{t}} \tag{3.13}
\]

Carrier pipe transition lengths obtained from the FE models are on average 13% greater than those predicted by Eq. (3.13). The average transition length of a single pipe (approximately 8\(D\)) obtained from the current FE analysis is close to 10\(D\), as reported by Chater and Hutchinson (1984), and 6\(D\), as reported by Albermani et al. (2011). With regard to the PIP systems, two distinctive transition lengths can be defined from the FE results: one corresponds to the onset of first contact between the carrier pipe and the inner pipe \((l^{i}_{t})\), and the other represents the length of the transition zone in full dog-bone propagation buckling mode in the PIP system \((l^{F}_{t})\). The transition lengths \(l^{i}_{t}\) and \(l^{F}_{t}\) are important parameters for efficient design of buckle arrestors in PIP systems (Kyriakides and Netto 2004). A value of \(l^{i}_{t} / l_t > 1\) essentially means that buckling is propagated along the carrier pipe without affecting the inner pipe, which is important from a design point of view. The full propagation transition lengths of the PIP systems \((l^{F}_{t})\) listed in Table 3.5 range from 10 to 20\(D_o\) with an average of 14 \(D_o\) — almost twice the corresponding transition lengths in single pipes.

<table>
<thead>
<tr>
<th>Identifier</th>
<th>(D_o/t_o)</th>
<th>(\hat{l}_t) (mm)</th>
<th>(l_t) (mm)</th>
<th>(l_t/D_o)</th>
<th>(l^{i}_{t})</th>
<th>(l^{F}_{t})</th>
</tr>
</thead>
<tbody>
<tr>
<td>PIP-1</td>
<td>40.0</td>
<td>644</td>
<td>698</td>
<td>8.72</td>
<td>1.02</td>
<td>19.26</td>
</tr>
<tr>
<td>PIP-2</td>
<td>30.0</td>
<td>418</td>
<td>487</td>
<td>8.11</td>
<td>0.79</td>
<td>12.87</td>
</tr>
<tr>
<td>PIP-3(^a)</td>
<td>26.7</td>
<td>526</td>
<td>611</td>
<td>7.64</td>
<td>0.90</td>
<td>10.77</td>
</tr>
</tbody>
</table>

\(^a\)Dog-bone buckle shape.
3.7 Summary

Experimental, analytical, and FE results for propagation buckling of pipe-in-pipe systems were presented in this chapter. Hyperbaric chamber tests were conducted on 1.6-m-long aluminum pipe-in-pipe systems. Confined buckling and flip-flop buckling modes were discovered in the hyperbaric chamber test of PIP-3 which had not been reported in the literature. These buckling modes occurred at propagation pressures close to that observed in a dog-bone buckle shape. Based on observations from the hyperbaric chamber test results of PIP systems, RST and CRST protocols were proposed. Nonlinear FE analyses were conducted and verified against the hyperbaric chamber tests. The FE models provided valuable information about the buckling modes and progress in the carrier and inner pipes.

The following conclusions can be drawn from this chapter:

The modified analytical solution suggested in this chapter accounts for the $D_i/D_o$ ratio and provides more accurate predictions of PIP propagation buckling pressure compared with previous analytical equations;

- The RST and CRST provide lower-bound estimates of PIP propagation pressure; for PIP systems with intermediate $D_i/t_o$ values, the RST pressure is almost half the propagation pressure as measured by the hyperbaric chamber tests; however, the CRST result is very close to the confined buckling propagation pressure obtained from the hyperbaric chamber tests;

- The nonlinear FE results suggest that the new buckling modes observed in PIP-3 are in accordance with different boundary conditions that the carrier pipe may adopt;

- The FE results show that the length of the transition zones of PIP systems are almost twice the corresponding lengths in single pipes; and

- This chapter focused on the propagation pressure of PIP systems with identical inner pipes but different carrier pipes; the FE results presented in Figures 3.13–3.15 showed substantial differences in the collapse pressure of the inner pipes in different PIP systems; further investigation is recommended to fully understand the failure mechanisms of the inner pipe of PIP systems.
NUMERICAL STUDY AND PARAMETRIC ANALYSIS OF THE PROPAGATION BUCKLING BEHAVIOUR OF SUBSEA PIPE-IN-PIPE SYSTEMS

Statement of contribution to co-authored published paper

This chapter includes a co-authored and peer-reviewed paper. The bibliographic details of the co-authored paper, including all authors, are:


The paper has been reformatted to meet the guidelines of the thesis. Minor explanation has been added in the paper for further clarity.

My contribution to this paper involved: literature review, experimental works, numerical modelling, result analysis, discussion of the results, writing, editing and response to reviewers.

(Signed) _________________________________ Date: 23.01.2020
PhD candidate: Mahmoud Alrsai.

(Countersigned) ___________________________ Date: 23.01.2020
Principal Supervisor: Dr Hassan Karampour.

(Countersigned) ___________________________ Date: 23.01.2020
Associate Supervisor: Dr Sanaul Chowdhury.

(Countersigned) ___________________________ Date: 23.01.2020
External Supervisor: Professor Faris Albermani.
Numerical Study and Parametric Analysis of The Propagation Buckling Behaviour of Subsea Pipe-in-pipe Systems

Abstract:

Propagation buckling mechanisms in pipe-in-pipe (PIP) systems with thin and moderately thin carrier pipes with a diameter-to-thickness ($D_o/t_o$) ratio in the range 26-40 are investigated using 2D analytical and 3D nonlinear (material and geometry) finite element (FE) models. The FE models are validated against hyperbaric chamber tests of a PIP system with $D_o/t_o$ of 30. Using the validated FE model, a parametric study is conducted, and two distinct buckle propagation modes in PIPs are observed. Empirical expressions for each mode are proposed and are found to be different from previous expressions suggested for PIPs with ($D_o/t_o$) ratio in the range 15-25.

Keywords:

Offshore pipelines; Pipe-in-pipe; Propagation collapse; External pressure; Buckling.

4.1 Introduction

Subsea pipe-in-pipe (PIP) systems are widely used due to their superior thermal insulation performance. A typical PIP system consists of a concentric inner pipe (also known as the product pipe) and the outer pipe (sometimes called the carrier pipe). Typically, the annulus space between the two pipes is filled with non-structural insulation materials such as polyurethane foam or water. In subsea applications, the outer pipe is designed to provide the protection from external pressure and mechanical damage, while the inner pipe is designed to carry the high temperature and high pressure (HT/HP) of the transporting hydrocarbons inside the pipe. The HP/HT conditions can cause global upheaval (Wang et al., 2015) or lateral buckling (Vaz and Patel, 1999; Wang et al., 2015a) in the PIP system. The external pressure in the vicinity of local dents or ovality in the outer pipe-wall can cause a local collapse, which may catastrophically propagate along with the PIP system, known as propagation buckling.
The lowest pressure required to sustain such a buckle propagation is known as the propagation pressure $P_p$, which is only 15-30% of the initiation pressure $P_I$ of the intact pipe. The propagation buckling (or buckle propagation) has been extensively investigated in single pipelines using analytical (Palmer and Martin, 1975; Mesloh et al., 1973), experimental (Kyriakides and Babcock, 1981; Albermani et al., 2011) and numerical methods (Gong et al., 2012; Pasqualino and Estefen, 2001). The possible interaction between lateral/upheaval buckling and propagation buckling was recently investigated by Karampour et al. (Karampour and Albermani, 2014; Karampour et al., 2013). A novel design for ultra-deep subsea pipelines was proposed and was shown to increase the propagation buckle capacity without increasing the wall thickness (Karampour and Albermani, 2016; Karampour et al., 2015).

Unlike single pipelines, buckle propagation in PIPs has only been marginally addressed (Kyriakides, 2002; Kyriakides and Vogler, 2002; Gong and Li, 2015; Karampour et al., 2017). A thorough experimental study on propagation buckling of steel PIPs with carrier pipes with $D_o/t_o$ values of 24.1, 21.1 and 16.7 and inner pipes of various $D_i/t_i$ ratios ranging between 15 to 37 was conducted by Kyriakides (Kyriakides, 2002). Based on this experimental study, an empirical formula for buckle propagation pressure, $P_{p2}$, of PIPs was proposed by Kyriakides and Vogler (Kyriakides and Vogler, 2002). An extensive finite element study of propagation buckling of PIPs using carrier pipes with $D_o/t_o$ of 25, 20 and 15 and inner pipes having $D_i/t_i$ of 15 and 20 was conducted by Gong and Li (Gong and Li, 2015) and another empirical formula for buckle propagation of PIPs was proposed. Karampour et al. (Karampour et al., 2017) investigated propagation buckling of PIPs with thinner carrier pipes ($26 < D_o/t_o < 40$) using ring squash tests, hyperbaric chamber tests and finite element analyses. They observed confined buckling mode shapes (Stephan et al., 2016) in PIP systems with moderately thick carrier pipes which were not reported before. Their non-linear FE results suggested that the confined buckling modes were in accordance with different boundary conditions that the carrier pipe adopted. The empirical expressions reported in (Kyriakides and Vogler, 2002; Gong and Li, 2015) were derived based on experimental and numerical results of buckle propagation pressures of PIPs with thick carrier pipes ($15 < D_o/t_o < 25$). Increasing the wall thickness of the carrier pipe increases the installation cost of the PIP system. A thicker carrier pipe also amplifies the axial force developed in the PIP system due to high internal pressure.
and temperature which in turn increases the risk of upheaval and lateral buckling in the system (Bokaian, 2004; Karampour et al., 2013). It is rather beneficial to use thinner carrier pipes with buckle arrestors (Olso and Kyriakides, 2003) to mitigate the risk of buckle propagation in the PIP system.

This chapter aims to provide insight on buckle propagation mechanisms of PIPs with moderately thin and thin carrier pipes using simplified analytical solutions and non-linear FE analyses. The FE results are validated against hyperbaric chamber tests of a 1.6 m long aluminium (Al-6060-T5) PIP system. Using the validated FE model, two distinctive buckle propagation modes in PIPs with thin carrier pipes are presented and separate expressions for buckle propagation pressure $P_{p2}$ of each mode are proposed. Through combining the previous experimental (Kyriakides, 2002) and numerical (Gong and Li, 2015) results with current FE results, a more accurate expression of $P_{p2}$ of PIPs with thick and moderately thick carrier pipes is proposed.

### 4.2 Propagation Buckling of Pipe-in pipe Systems

#### 4.2.1 Analytical solution

Previous studies have proposed analytical solutions to the collapse of a single pipeline based on initial and final configurations of the cross-section of the pipe (Palmer and Martin, 1975; Mesloh et al., 1973). In such 2D models, the material is assumed to be rigid-perfectly plastic and deformation is limited to the development of four plastic hinges in the pipe wall shown in Figure 4.1(a-c). The propagation pressure is calculated by equating the external work done by the pressure due to the change in the volume, to the plastic flexural work depleted in the four hinges (Kyriakides, 2002). Albermani et al. (Albermani et al., 2011) accounted for the membrane and flexural effects in the wall of a single pipe during the deformation and arrived at an expression for propagation pressure of a single pipe:

$$
\hat{P}_p = \frac{3}{2.515} \left[ \pi \sigma_y \left( \frac{t}{D} \right)^2 \right] 
$$

(4.1)
The schematic deformation stages in propagation buckling of a pipe-in-pipe system are shown in Figure 4.1(d-f). Kyriakides and Vogler (Kyriakides and Vogler, 2002) used plane strain conditions and strain hardening behaviour and proposed the following expression for the propagation pressure of the PIP system based on the development of four plastic hinges in each of the carrier and the inner pipes (Figure 4.1(d-f)).

\[
\hat{P}_{p2} = \frac{2\pi}{\sqrt{3}} \sigma_{yo} \left( \frac{t_o}{D_o} \right)^2 \left[ 1 + \frac{\sigma_{yi}}{\sigma_{yo}} \left( \frac{t_i}{t_o} \right)^2 \right]
\]

(4.2)

Accounting for the circumferential membrane as well as flexural effects in the outer and inner pipe walls:

\[
W_{ex} = W_{in(f)} + W_{in(m)}
\]

(4.3)

where \(W_{ex}\) is the external work done by the net hydrostatic pressure, and \(W_{in}\) is the internal work due to the circumferential flexure, \(f\), and membrane, \(m\), effects. Based on the experimental observations from the hyperbaric chamber and ring squash tests, the initially circular cross-section of the outer pipe (Figure 4.1(d)) has found to deform into the shape shown in (Figure 4.1(e)). Further increase in the external pressure causes the pipe-in-pipe system to eventually deform into the dog-bone shape (Figure 4.1(f)). Thus Eq. (4.3) can be written as:

\[
\hat{P}_{p2}(\Delta A) = 3\pi \left( m_{po} + m_{pi} \right) + \hat{P}_{p2} \left( r_o \Delta l_o + r_i \Delta l_i \right)
\]

(4.4)

where the subscript “o” denotes the outer pipe, and “i” represents the inner pipe, \(\Delta A\) is the change in the cross-section area, \(\Delta l\) is the change in the circumferential length and \(m_p\) is the plastic moment (Albermani et al., 2011). These are given by:

\[
\Delta A = \pi r_o^2
\]

(4.5a)

\[
\Delta l_o = (2\pi - 4\sqrt{2})r_o; \quad \Delta l_i = (2\pi - 4\sqrt{2})r_i
\]

(4.5b)

\[
m_{po} = \sigma_{yo} \frac{t_o^2}{4}; \quad m_{pi} = \sigma_{yi} \frac{t_i^2}{4}
\]

(4.5c)
Substituting Eqs. (4.5a) to (4.5c) into (4), the propagation pressure, $\tilde{P}_{P2}$, and the propagation pressure normalized by the propagation pressure (Eq. (4.1)) of the single outer pipe, $\tilde{P}_{P2}/\tilde{P}_P$, of the PIP system are obtained as:

$$\tilde{P}_{P2} = \left[ \frac{3\pi \sigma_{yo}}{2.515 \left( \frac{D_o}{D_o} \right)^2} \right] \left[ 1 + \frac{\sigma_{yo}}{t_i} \left( \frac{t_i}{t_o} \right)^2 \right] \left[ \frac{1}{1 - \left( \frac{D_i}{2D_o} \right)^2} \right]$$

(4.6a)

$$\frac{\tilde{P}_{P2}}{\tilde{P}_P} = \left[ 1 + \frac{\sigma_{yo}}{\sigma_{yo}} \left( \frac{t_i}{t_o} \right)^2 \right] \left[ \frac{1}{1 - \left( \frac{D_i}{2D_o} \right)^2} \right]$$

(4.6b)

when $D_i = t_i = 0$, Eq. (4.6a) yields the propagation pressure of a single pipe given in Eq. (4.1). Unlike Eq. (4.2), Eq. (4.6a) accounts for the effect of $D_i/D_o$ as well as that of $t_i/t_o$ and $\sigma_i/\sigma_o$.

When compared with the experimental results, a better prediction of $\tilde{P}_{P2}$ for PIPs was observed in the previous chapter using Eq. (4.6a) rather than Eq. (4.2) and irrespective of $D_o/t_o$ ratio.

**Figure 4-1:** Schematic of deformation stages in propagation buckling: (a–c) single pipe; (d–f) pipe-in-pipe system.
4.2.2  Hyperbaric chamber test

A concentric aluminium (Al-6060-T5) PIP system with parameters represented in Table 4.1 and a length of 1.6 m i.e. \( L/D_o > 20 \), was end-sealed and pressurized inside the hyperbaric chamber shown in Figure 4.2. The chamber has an inner diameter of 173 mm and a length of 4 m and is rated for working pressure of 20 MPa (2000 m water depth). To end-seal the PIP system, thick aluminium discs were glued to the ends, ensuring that the inner and outer pipes were concentric and that the inner pipe was completely sealed from the outer pipe. A set of two valves were connected to the end of each of the outer and inner pipes of the PIP. One valve was used for bleeding the pipe while filling it with water. The second valve was utilized to vent each of the carrier and inner pipes, as well as to separately collect water from the inner pipe and the cavity between the inner and outer pipes during the buckle propagation. The pressure inside the chamber was incremented using a high-pressure pump (shown in Figure 4.2) until the collapse of PIP system occurred under quasi-static steady-state conditions. The total change in volume of the PIP system (\( \Delta V \)) during the test was calculated by adding the weight of water being discharged from the inner pipe and the cavity between the pipes which were measured using digital weighing scales. Control tests using a single pipe (outer pipe) were conducted first.

![Figure 4-2: The hyperbaric chamber used in the test showing; sample, high-pressure pump, pressure gauge and vents.](image-url)
The experimental results of the buckle propagation response of the PIP system of Table 4.1 are shown in Figure 4.3(a). The pressure inside the chamber was increased until collapse of the outer pipe of the PIP system was triggered at the initiation pressure $P_I$. The initiation pressure is very sensitive to imperfections of the pipe cross-section (Kyriakides, 2002) which is defined in terms of ovalization ratio $\Delta_o$:

$$\Delta_o = \frac{D_{\text{max}} - D_{\text{min}}}{D_{\text{max}} + D_{\text{min}}}$$

(4.7)

where $D_{\text{max}}$ and $D_{\text{min}}$ are the maximum and minimum diameters of the outer pipe. The ovalization of the outer pipe cross-section of the PIP system was measured at different points along the length of the PIP and a maximum ratio of $\Delta_o = 0.5\%$ was obtained. Following the initial collapse, the pressure in the system suddenly dropped to a much lower pressure highlighted as $P_{p2}$ in Figure 4.3(a). By maintaining a low rate of pressurization, the chamber pressure was stabilized at propagation pressure, $P_{p2}$, with buckling longitudinally propagating along the PIP system sample accompanied by water flow from the vents. The deformed configuration and dog-bone cross-section of the PIP system after the test are depicted in Figure 4.3(b). The propagation pressures of the control test using a single outer pipe ($P_p$) and the PIP system ($P_{p2}$) are presented in Table 4.2. These results are the average of three PIP tests.

The modulus of elasticity ($E$) of the specimens listed in Table 4.1 was obtained from two compressive stub tests (Figure 4.4(a)) conducted for each $D/t$. The yield stress listed in Table 4.1 was calculated based on the results of two ring squash tests (RST) shown in Figure 4.4(b). The ring squash test (Albermani et al., 2011; Karapour et al., 2017; Kamalarasa and

Table 4.1 Geometric and material parameters of PIP system.

<table>
<thead>
<tr>
<th></th>
<th>$D/t$</th>
<th>$D$ (mm)</th>
<th>$t$ (mm)</th>
<th>$L$ (mm)</th>
<th>$E$ (MPa)</th>
<th>$E'/E$ (%)</th>
<th>$\sigma_y$ (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer pipe</td>
<td>30</td>
<td>60</td>
<td>2</td>
<td>1600</td>
<td>66,680</td>
<td>1.0</td>
<td>139</td>
</tr>
<tr>
<td>Inner pipe</td>
<td>25</td>
<td>40</td>
<td>1.6</td>
<td>1600</td>
<td>66,680</td>
<td>1.0</td>
<td>172</td>
</tr>
</tbody>
</table>
Calladine, 1988) is conducted on a short segment of the pipe specimen compressed between two rigid indenters of the same diameter as the pipe specimen (Figure 4.4(b)). The yield stress, $\sigma_y$, is calculated from

$$\sigma_y = \frac{F_o D}{2L_{RST} t^2}$$

(4.8)

where $F_o$ is the RST load at which the four plastic hinges are shown in Figure 4.1(a-c) are developed in the pipe wall. $L_{RST}$ is the length of the RST sample which is 150mm (Albermani et al., 2011). The material tangent modulus of $E'/E = 1\%$ was adopted for the inner and outer pipes.

Figure 4-3: (a) Pressure versus normalized change in volume for pipe-in-pipe sample in Table 4.1, (b) Deformed configuration of the pipe-in-pipe sample after test, (c) Deformed shape of the FE model.
4.2.3 FE analyses and comparison with experimental results

Finite-element analyses of 1.6-m-long samples of the single carrier pipe and the PIP system described in Table 4.1 and tested in the hyperbaric chamber were carried out using ANSYS 17.0. Thin four-node shell elements (181) were used to model the carrier and the inner pipes. Frictionless contact elements (174 and 170) were used in three pairs to define the non-linear contact between the carrier and inner pipes and the inner surfaces of the inner pipe wall. Because of symmetry, a one-half model of the pipe wall (180°) was discretised using 24 elements in the circumferential direction with seven integration points through the thickness of the carrier and inner pipes. Ovalization ratio of $\Delta o = 0.5\%$ (same as that measured in the test sample) was introduced at mid-length on the carrier pipe in the FE model. The nodes at either end of the PIP system were restrained from translation in all directions. A Von-Mises elastoplastic (bilinear) material definition with isotropic hardening and parameters shown in Table 4.1 was adopted.

The FE response of the PIP system is plotted in Figure 4.3(a), and deformed configurations are shown in Figure 4.3(c) and show good agreement with the experimental results. The propagation pressures of the single pipe and PIP system from the FE models are presented in Table 4.2 together with the experimental results. The FE prediction is nearly 96% of the experimental results for both $P_p$ and $P_{p2}$.
4.3.1 Propagation Buckling of the PIP System with a Solid Inner Pipe

Subsea pipe-in-pipe systems are designed to safely operate at pressures larger than the propagation pressure \( P_{p2} \). The buckle propagation of the system due to the collapse of the outer pipe is controlled by using buckle arrestors at intermittent locations along the length of the PIP system (Olso and Kyriakides, 2003). It is of interest to find the propagation pressure of a PIP system with a solid inner pipe (\( P_{ps} \)) as it provides the abounding value of \( P_{p2} \) in the PIP system. Finite element buckle propagation response of a thin carrier pipe of \( D_o/t_o = 40 \) with a solid insert of \( D_s/D_o = 0.5 \) is shown in Figure 4.5(a). The pressure is normalised to the propagation pressure of a single outer pipe (\( P_p \)) and is plotted against the normalized ovalization of the carrier pipe (\( \Delta D_o/D_o \)). In the FE model, the modulus of elasticity of the solid inner pipe is substantially larger than that of the carrier pipe (\( E_i/E_o =100 \)) to account for its rigid behaviour. Deformed shapes of the pipe at different pressure levels are shown in Figure 4.5(b). Following the initial collapse, the carrier pipe touches the solid insert at configuration (II). Then the pressure is maintained at \( P_{ps}/P_p \) and the buckle propagates along the length of the carrier pipe. At configuration (IV), the buckle is extended to the end bulkheads and the pressure level needs to increase to cause further deformation of the cross-section.

### Table 4.2 Comparison between experimental, numerical and analytical results.

<table>
<thead>
<tr>
<th></th>
<th>Experimental</th>
<th>FEA</th>
<th>Analytical</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_p ) (kPa)</td>
<td>1020</td>
<td>976</td>
<td>578 (Eq. (4.1))</td>
</tr>
<tr>
<td>( P_{p2} ) (kPa)</td>
<td>1570</td>
<td>Test 1 1620 Test 2 1540 Test 3 1550</td>
<td>1504</td>
</tr>
<tr>
<td>( P_{p2}/P_p )</td>
<td>1.539</td>
<td>1.541</td>
<td>2.015 (Eq. (4.6b))</td>
</tr>
</tbody>
</table>
A set of FE analyses are conducted for a range of moderately thick to thin outer pipes ($D_o/t_o = 26.7, 30, 40$) and solid inserts with diameter ratios of $(0.4 < D_s/D_o < 0.75)$ and the results are shown in Figure 4.6(a). Using the Levenberg-Marquardt algorithm of non-linear least squares, the best fit of all data from $D_o/t_o = 30, 40$ and 26.7 of Figure 4.6(a) is obtained with a correlation factor of $R^2=0.9835$ as:

$$\frac{P_{ps}}{P_p} = 1 + 2.097 \left( \frac{D_s}{D_o} \right)^{2.507}$$  \hspace{1cm} (4.9)
The proposed expression (Eq. (4.9)) is plotted against reported experimental (Kyriakides and Vogler, 2002) and numerical (Gong and Li, 2015) empirical equations in Figure 4.6(b). It should be noted that those studies (Kyriakides and Vogler, 2002; Gong and Li, 2015) only considered propagation pressure of thick to moderately thick carrier pipes with solid inner pipes. Current expression (Eq. (4.9)) appears to be in good agreement with that reported in (Kyriakides and Vogler, 2002) which was derived based on extensive experimental results. With respect to Figure 4.6(a), the differences between $P_{ps}$ of moderately thin and moderately thick pipes are constant (about 20%) for various $D_s/D_o$. This suggests that Eq. (4.9) which is derived based on non-linear least squares be independent of $D_o/t_o$ ratio and thus Eq. (4.9) and that proposed by Kyriakides (Kyriakides and Vogler, 2002) yield similar results.

**Figure 4-6:** (a) FE results of propagation pressure $P_{ps}$ of PIPs with solid inner pipes; (b) comparison between current expression and those proposed in previous studies.
4.3.2 Propagation Buckling Modes in Pipe-in-pipe Systems

In section 4.4 of this chapter, a comprehensive parametric study is conducted using the validated FE model to find the buckle propagation pressures of PIP systems with various wall thickness $t_i/t_o$, diameter $D_i/D_o$, and the material yield stress $\sigma_{yi}/\sigma_{yo}$ ratios. Prior to reviewing results of the parametric dependence of propagation buckling of PIPs, it is worth discussing the buckling modes observed in the FE simulations. Based on the results of FE analyses with various parameters adopted for the outer and inner pipes, two dominant modes of failure under external pressure were observed in the PIPs. The buckling response of a thin PIP with $D_o/t_o$ of 40 is shown in Figure 4.7. The thickness ratio is $t_i/t_o = 0.6$ and the properties of the material of outer and inner pipes are identical. By increasing the external pressure, the carrier pipe collapses and gradually deforms from the undeformed shape (I) into the deformed shape (II). At this stage, the outer and inner pipes come into contact. Following the touchdown (II), the pressure is slightly increased, and the collapse is propagated along the lengths of the outer and inner pipes simultaneously as shown in stages (III) to (IV). The buckle propagation mode shown in Figure 4.7 is referred to as Mode $A$ herein. Figure 4.8 shows the pressure response and the deformed shape of a moderately thin PIP with $D_o/t_o$ of 30 and $t_i/t_o$ of 0.8. The mechanical properties of the inner and outer pipes are alike. Following the initiation of collapse in the outer pipe, the pressure in the system is dropped and the buckle is propagated in the carrier pipe as shown in deformed shapes of II and III in Figure 4.8. The buckle propagation in the outer pipe is eventually arrested by the end-caps as shown in the deformed shape (III). In the vicinity of the end-caps, higher pressure is required to perpetuate the collapse in the outer pipe. However, the increase in pressure causes a collapse in the inner pipe at the pressure level (IV) and initiates a buckle which is propagated through the length (V). This buckle propagation mode is referred to as Mode $B$ in this chapter.
Figure 4-7: Finite element results showing pressure against normalized ovality and corresponding deformed shapes of PIP system exhibiting buckle propagation Mode A.
Figure 4-8: Finite element results showing pressure against normalized ovality and corresponding deformed shapes of PIP system exhibiting buckle propagation Mode B.

4.4 Parametric Study on Buckle Propagation Pressure of PIPs

The buckle propagation pressure of the PIP system is related to geometric and material parameters of the outer and inner pipes.
In the parametric study carried out herein, both outer and inner pipes are assumed to be of the same material i.e. aluminum (Al-6060-T5) with the same modulus of elasticity ($E$) and tangent modulus ($E'$), shown in Table 4.1 and Poisson’s ratio of ($\nu = 0.33$). Gong and Li (Gong and Li, 2015) reported that the strain hardening modulus has little effect on the propagation pressure of PIP systems and is thus not considered in this parametric study. Using the dimensional analysis, Eq. (4.10) can be written in terms of non-dimensional geometric and material parameters in the following non-dimensional power-law format:

$$
\frac{P_{p2}}{P_p} = 1 + A_1 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^a \left( \frac{D_i}{D_o} \right)^b \left( \frac{t_i}{t_o} \right)^c
$$

(4.11)

Two significant experimental [15] and numerical [16] studies on buckle propagation of PIP systems with carrier pipes of $D_o/t_o < 25$ were carried out and empirical Eqs. (4.12a) and (4.12b) were derived respectively. Although both studies (Kyriakides and Vogler, 2002; Gong and Li, 2015) covered similar $D_o/t_o$ range of the carrier pipes, the empirical expressions suggested in Eqs. (4.12a) and (4.12b) are different.

$$
P_{p2} = 1 + 1.095 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^{0.4} \left( \frac{D_i}{D_o} \right) \left( \frac{t_i}{t_o} \right)^2
$$

(4.12a)

$$
P_{p2} = 1 + 0.970 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^{0.8} \left( \frac{D_i}{D_o} \right)^{0.3} \left( \frac{t_i}{t_o} \right)^2
$$

(4.12b)

Extensive FE simulations are conducted in the following sections to find best estimates for unknown power coefficients of Eq. (4.11) in PIP systems with thin and moderately thin carrier pipes ($D_o/t_o > 25$). In addition, the raw data from both previous studies (Kyriakides and Vogler, 2002; Gong and Li, 2015) are combined with the current FE results for PIP system with $D_o/t_o = 26.7$ to establish an empirical expression that covers the buckle propagation pressure of PIPs with thick and moderately thick carrier pipes.
4.4.1 Effect of $D_i/D_o$

To investigate the effect of $D_i/D_o$ on the propagation pressure of PIPs, the thickness ratios are varied while the material properties of the two pipes are kept identical. Figure 4.9 shows $P_{p2}/P_p$ versus $D_i/D_o$ for two different $D_o/t_o$ ratios of 40 and 30 in four sets of $t_i/t_o$. The relationship between $P_{p2}/P_p$ and $D_i/D_o$ is linear with a positive slope when $t_i/t_o \leq 0.6$, which corresponds to collapse propagation Mode A (see Figure 4.7). However, by increasing the thickness ratio and at $t_i/t_o \geq 0.8$, the relationship becomes nonlinear with a decreasing negative slope for both $D_o/t_o$ ratios. The decreasing trend at $t_i/t_o \geq 0.8$ is associated with collapse propagation Mode B (see Figure 4.8) and was not reported in the previous studies (Kyriakides and Vogler, 2002; Gong and Li, 2015).

**Figure 4-9:** Propagation pressure as a function of $D_i/D_o$ in (a) thin PIPs and (b) moderately thin PIPs.
A comparison between current FE and analytical results and those predicted by Eqs. (4.12a) and (4.12b) are depicted in Figure 4.10. For $D_o/t_o$ of 40 and $t_i/t_o$ of 0.6 in Figure 4.10(a), the ascending linear trend obtained from the current FE results is similar to the previous predictions. This was expected because all failure modes are of mode A. However, as shown in Figure 4.10(b), for $D_o/t_o$ of 30 and $t_i/t_o$ of 1.0, the previously proposed equations are incapable of predicting the correct propagation pressure of thin to moderately thin PIPs. This is due to the fact that Eqs. (4.12a) and (4.12b) are based on buckling mode A only, whereas the failure mode in Figure 4.10(b) is mode B. The analytical results obtained from Eq. (4.6b) are shown with solid lines and provide upper bounds of $P_{p2}/P_p$. Similar to the FE results of mode A, the analytical normalised pressures in Figure 4.10, increase with the corresponding increase in the diameter ratios. This agrees with the buckle propagation mode (Mode A) assumed in the derivation of the analytical solution shown in Figure 4.1 (d-f).

**Figure 4-10:** Comparison between current FE results and previous studies for (a) thin PIPs and (b) moderately thin PIPs.
For a fixed value of $t_i/t_o$ and assuming identical yield stresses in the outer and inner pipes, Eq. (4.11) is reduced to

$$\frac{P_{p2}}{P_p} = 1 + A_2 \left( \frac{D_i}{D_o} \right)^{b_1} \tag{4.13}$$

Based on the non-linear least-squares fit of eight sets of data in Figure 4.9, the power exponents $b_1 = 0.4$ corresponding to failure mode $A$ and $b_2 = -0.8$ corresponding to failure mode $B$, and the coefficients $A_2 = 0.394$ and $0.168$, respectively are calculated.

### 4.4.2 Effect of $t_i/t_o$

To understand the effect of $t_i/t_o$ on the propagation pressure of PIPs, plots of normalised pressures against thickness ratios for two sets of $D_i/D_o$ are presented in Figure 4.11, and the collapse modes are indicated next to the corresponding numerical data. The thickness ratios cover a practical range used in offshore PIPs. Nonlinear ascending relationships are observed for both $D_i/D_o$ ratios. The results show that in PIPs with $D_i/D_o = 0.4$ and at $t_i/t_o < 0.7$, the collapse propagation mode is Mode (A), and in PIP with $D_i/D_o = 0.7$ a distinction between modes $A$ and $B$ is observed at $t_i/t_o = 0.8$. It is also evident from Figure 4.11 that at larger thickness ratios, previous expressions (Eqs. (4.12a) and (4.12b)) underestimate the propagation pressure of the PIP system significantly. The analytical solution gives the upper bound pressures of buckle propagation mode $A$. Given the material properties of the outer and inner pipes are identical, Eq. (4.11) can be written as

$$\frac{P_{p2}}{P_p} = 1 + A_3 \left( \frac{D_i}{D_o} \right)^{b_1} \left( \frac{t_i}{t_o} \right)^c \tag{4.14}$$

Non-linear least-squares fit of the data in Figure 4.11(a) and (b) yield the same power exponent $c = 2.4$ and coefficient $A_3 = 0.461$ corresponding to failure mode $A$ and $A_3 = 0.137$ corresponding to failure mode $B$. 
Figure 4-11: Comparison between current FE results and previous studies for (a) thin and (b) moderately thin PIPs with different \( t_i/t_o \) ratios. The failure modes are labelled for the corresponding data point.

4.4.3 Effect of \( \sigma_{yi}/\sigma_{yo} \)

To examine the effect of \( \sigma_{yi}/\sigma_{yo} \) ratio on the propagation pressure, two values of \( t_i/t_o \) with two values of \( D_o/t_o \) are assumed and displayed in Figure 4.12. Comparison between current FE results and those of Eqs. (4.12a) and (4.12b) demonstrates similar trends. To identify the power exponent of \( \sigma_{yi}/\sigma_{yo} \), Eq. (4.11) can be represented as
\[ \frac{P_{p2}}{P_p} = 1 + A_4 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^a \] (4.15)

Based on two different values of \( t_i/t_o \) for each value of \( D_o/t_o \), four sets of data are generated. The least squares fit of the data gives the value of power exponent \( a = 0.2 \) and the coefficient \( A_4 = 0.218 \). The linear relationship between normalised pressure and yield ratios predicted in the analytical solution (Eq. 4.6b), provides an upper bound of the FE results.

**Figure 4-12:** Propagation pressure as a function of \( \sigma_{yi}/\sigma_{yo} \) in (a) thin PIPs and (b) moderately thin PIPs the failure modes are labelled for the corresponding data point.
4.5 Empirical Expressions for $P_{P2}$

4.5.1 Empirical expressions for buckle propagation of PIPs with thin and moderately thin carrier pipes.

The parametric study carried out in section 4.4 ascertained the dependency of the propagation pressure of the PIP systems on geometric and material parameters of the outer and inner pipes. Moreover, current FE results proved that the buckle propagation modes of PIPs with large $D_o/t_o$ ratios are not essentially similar to mode $A$ predicted in previous studies (Kyriakides and Vogler, 2002; Gong and Li, 2015). Since proposed Eqs. (2.12a) and (2.12b) are incapable of predicting proper estimates of propagation pressure of PIPs that exhibit buckle propagation mode $B$, it is sensible to propose expressions for buckle propagation modes $A$ and $B$ separately. Based on the results of the parametric study in the previous section and using non-linear square fits of sets of data taken from the FE results, the following expressions are derived for the propagation pressure of PIPs with thin and moderately thin carrier pipes for buckle propagation modes $A$ and $B$

\[
\frac{P_{p2}}{P_p} = 1 + 1.047 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^{0.2} \left( \frac{D_i}{D_o} \right)^{0.4} \left( \frac{t_i}{t_o} \right)^{2.4} \quad \text{Mode (A)} \quad (4.16a)
\]

\[
\frac{P_{p2}}{P_p} = 1 + 0.596 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^{0.2} \left( \frac{D_i}{D_o} \right)^{-0.8} \left( \frac{t_i}{t_o} \right)^{2.4} \quad \text{Mode (B)} \quad (4.16b)
\]

The coefficients in Eqs. (4.16a) and (4.16b) are determined using the Leven-berg-Marquardt algorithm and correspond to correlation factors ($R^2$) of 0.9827 and 0.9860 respectively. Comparison between the FE results and the proposed expressions are shown in Figures 2.13(a) and (b) for buckle propagation modes $A$ and $B$ respectively. The maximum differences between FE results and empirical expressions are less than 6.0%.
Figure 4-13: Comparison between FE results and those predicted by Eqs. (4.16a) and (4.16b) of buckle propagation pressures of PIP with buckle propagation (a) Mode A; and (b) Mode B.
Since Eq. (4.9) is a special case of Eqs (4.16a and 4.16b), it is meaningful to compare buckle propagation pressure predictions obtained from those expressions. Assuming mode $A$ of buckle propagation, Eq (4.16a) can be written as:

$$\frac{P_{p2}}{P_p} = 1 + 1.047 \left( \frac{\sigma_{si}}{\sigma_{yo}} \right)^{0.2} \left( \frac{D_i}{D_o} \right)^{2.8} \left( \frac{D_o}{D_i} \right)^{2.4}$$ \hspace{1cm} (4.16a-1)

Assuming outer and inner pipes of similar $D/t$, and inner pipe with yield stress 100 times larger than the outer pipe ($\sigma_{yi}/\sigma_{yo} = 100$);

$$\frac{P_{ps}}{P_p} = 1 + 1.497 \left( \frac{D_s}{D_o} \right)^{1.6}$$ \hspace{1cm} (4.16a-s)

Using the same analogy, Eq (4.16b) can be written as:

$$\frac{P_{ps}}{P_p} = 1 + 2.617 \left( \frac{D_s}{D_o} \right)^{2.8}$$ \hspace{1cm} (4.16b-s)

For a range of $D_s/D_o$ between 0.4 and 0.8, the difference between results of $P_{ps}/P_p$ from Eqs. (4.9), (4.16a-s) and (4.16b-s) are less than 10%.

Figures 4.14 and 4.15 show plots of normalized propagation pressures of thin and moderately thin PIP systems with different $D_i/D_o$ and $t_i/t_o$ ratios respectively. Buckle propagation modes $A$ and $B$ are distinguished with different data markers on the figures. Buckle propagation pressures of PIP systems with solid inner pipes (Eq. (4.9)) are shown with solid lines in Figure 4.14. It is clear from Figure 4.14 that propagation pressures of PIPs with solid inner pipes yield upper bounds of propagation pressures for mode $A$. In other words, Eq. (4.9) can be used to obtain a maximum of Eq. (4.16a).
Figure 4-14: Buckle propagation pressures of Mode A and B compared against buckle propagation pressures of PIP with solid inner pipe for (a) $D_o/t_o = 30$, and (b) $D_o/t_o = 40$.

Propagation pressures of moderately thick to thin PIP systems with $\sigma_y/\sigma_{yo} = 1.0$ and thickness ratios between 0.3 and 1.2 are presented in Figure 4.15. According to the figure, the buckle propagation modes A and B correspond to $t_i/t_o < 0.7$ and $t_i/t_o > 0.8$ respectively. It can be deduced that in PIPs with outer and inner pipes of similar material properties, the separation between buckle propagation modes A and B occurs at thickness ratios between 0.7 and 0.8. Thus, given $\sigma_y/\sigma_{yo} = 1.0$, Eqs. (4.16a) and (4.16b) can be used to predict the propagation pressures of PIPs with $t_i/t_o < 0.7$ and $t_i/t_o > 0.8$ respectively.
4.5.2 Empirical expressions for buckle propagation of PIPs with thick and moderately thick carrier pipes

The expressions (Eqs. (4.16a) and (4.16b)) derived in section 4.5.1 can be used to predict the propagation pressure of PIP systems with thin and moderately thin carrier pipes. In order to come up with an expression to predict the propagation pressure of PIPs with thick and moderately thick carrier pipes, a total of 254 data points were collected from the raw data reported in (Kyriakides, 2002; Gong and Li, 2015), and the current FE results for PIPs with $D_o/t_o = 26.67$. Using the Levenberg-Marquardt algorithm of non-linear least squares the following expression was derived for the propagation pressure, $P_{p2}$, of PIPs with $D_o/t_o < 27$

$$
\frac{P_{p2}}{P_p} = 1 + 0.803 \left( \frac{\sigma_{yt}}{\sigma_{yo}} \right)^{0.4} \left( \frac{D_l}{D_o} \right)^{0.13} \left( \frac{t_l}{t_o} \right)^{1.8}
$$

(4.17)

with multiple correlation factor ($R^2$) of the fit is 0.9781. To derive Eq. (4.17), same procedure as explained in derivation of Eqs. (4.16a) and (4.16b) is followed and the interaction between non-dimensional variables are incorporated. For the sake of brevity, the procedure is not shown here. Current FE results, the FE results of (Gong and Li, 2015) and experimental
results of (Kyriakides, 2002) are plotted in Figure 14.6 against the proposed expression (Eq. (4.170) and form a nice linear band. It should be noted that all of the data points in Figure 4.16 and thus those derived from Eq. (4.17) correspond to buckle propagation mode \( A \).

\[
\frac{P_{p2}}{P_p} = 1 + 0.803 \left( \frac{\sigma_{yi}}{\sigma_{iy}} \right)^{0.4} \left( \frac{D_i}{D_o} \right)^{0.13} \left( \frac{t_i}{t_o} \right)^{1.8}
\]

Figure 4-16: Comparison between buckle propagation pressures of thick to moderately thick PIP systems from previous studies and current expression. (All results correspond to the buckle propagation mode \( A \))

4.6 Summary

Buckling propagation mechanisms of subsea pipe-in-pipe (PIP) systems under external pressure in quasi-static steady-state conditions were investigated using 2D analytical solutions and 3D FE analyses considering non-linear material and geometric behaviour. The FE results were validated against experimental results of propagation buckling response of a PIP system tested in a hyperbaric chamber. Using the validated FE model, a parametric study was conducted and two major buckle propagation modes in PIPs with thin and moderately thin carrier pipes were observed. In Mode \( A \) the buckle propagated simultaneously in the outer and
inner tubes, and in Mode $B$ the buckle propagated in the outer pipe and the collapse in the inner pipe was delayed.

It was shown that compared to FE results, the analytical solution provides upper bound for normalised buckle propagation pressures ($P_{p2}/P_p$) of the PIP systems. Based on the FE results, a new expression for the propagation buckling pressure of pipe with a solid inner pipe was proposed (Eq. (4.9)). It was found that the proposed expression provides a lower bound for buckling pressure of PIPs in mode $B$. Current FE results of propagation pressures of thin PIPs demonstrated a nonlinearly decreasing trend in the normalised pressure with corresponding increase in $D_i/D_o$ ratios. This behaviour is associated with buckle propagation mode $B$ and was not reported in the previous studies. The FE results showed that in PIPs with outer and inner pipes of the same modulus of elasticity and yield stress, mode $A$ occurs when $t_i/t_o < 0.7$ and mode $B$ happens at $t_i/t_o > 0.7$. Two separate expressions (Eqs. (4.16a) and (4.16b)) for buckle propagation pressures in modes $A$ and $B$ of PIP systems with thin and moderately thin carrier pipes are proposed. Based on the combined data from previous studies and current FE results, a more comprehensive empirical expression (Eq. (4.17)) was proposed to predict the propagation pressure $P_{p2}$ of PIPs with thick and moderately thick carrier pipes.
Statement of contribution to co-authored published paper

This chapter includes a co-authored and peer-reviewed paper. The bibliographic details of the co-authored paper, including all authors, are:


The paper has been reformatted to meet the guidelines of the thesis. Minor explanation has been added in the paper for further clarity.

My contribution to this paper involved: literature review, experimental works, numerical modelling, result analysis, discussion of the results, writing, editing and response to reviewers.

(Signed) _________________________________ Date: 23.01.2020

PhD candidate: Mahmoud Alrsai.

(Countersigned) ___________________________ Date: 23.01.2020

Principal Supervisor: Dr Hassan Karampour.

(Countersigned) ___________________________ Date: 23.01.2020

Associate Supervisor: Dr Sanaul Chowdhury.

(Countersigned) ___________________________ Date: 23.01.2020

External Supervisor: Professor Faris Albermani.
On Collapse of The Inner Pipe of a Pipe-in-pipe System Under External Pressure

Abstract:

Collapse of the inner pipe of a pipe-in-pipe (PIP) system under external pressure is studied experimentally and numerically herein. Hyperbaric chamber test results of three PIP systems with identical inner pipes and different outer pipes are presented. It is observed that the geometric and material properties of the outer pipe affect the collapse pressure of the inner pipe. Using validated finite element analyses (FEA), a parametric study is conducted and collapse mechanisms of PIPs with various combinations of outer and inner pipes with a practical range of diameter-to-thickness ratios \(D/t\) between 15 to 40 are discussed. Empirical expressions are proposed for the collapse pressure of the inner pipe \(P_{ci}\), and its upper and lower bounds. The proposed empirical equation for \(P_{ci}\), is shown to agree well with the experimental results of the tested PIPs. Moreover, two distinctive modes of collapse in the inner pipe are identified and discussed.

5.1 Introduction

Subsea pipe-in-pipe systems are preferred to conventional single-walled pipelines due to their superior thermal insulation performance. The PIP system consists of a concentric inner pipe (also known as the product pipe) and the outer pipe (sometimes called the carrier pipe) (Bai and Bai, 2005; Bokaian, 2004). The inner pipe is designed to carry the high temperature and high pressure (HT/HP) of the hydrocarbons inside the pipe. The outer pipe protects the system from external pressure and mechanical damage. The annulus (the space between the tubes) is either empty or filled with non-structural insulation material such as foam or water. Pipe-in-pipe systems are exploited in subsea developments, where the carrier pipe is designed to resist high hydrostatic pressures (water depths up to 3,000 m) and the inner pipe is designed to transmit hydrocarbons at temperatures as high as 180°C and internal pressure up to 10 MPa (Jukes et al., 2009). The HP/HT flow can cause global upheaval (Wang et al., 2015;
Karampour et al., 2013) or lateral (Vaz and Patel, 1999; Wang et al., 2015a; Karampour, 2018) buckling of the system.

In a single pipeline under external pressure, a local dent or ovalization in the pipe wall can cause a local collapse. The collapse pressure of a single pipeline \( (P_{cr}) \), with perfectly circular cross-section, can be approximated by the classical expression for buckling of elastic tubes under uniform external pressure. The classical expression (Eq. (5.1)) was first derived by Bresse, (1866) to predict the collapse pressure for a ring under external pressure and then extended by Bryan, (1888) to include long cylinder, furthermore in 1961, Timoshenko and Gere, (1961) investigated the collapse pressure of elastic rings in the presence of initial ovality:

\[
P_{cr} = \frac{2E}{1-\nu^2} \left( \frac{t}{D} \right)^3
\]

In offshore applications, the pipelines typically have diameter-to-thickness ratios \( (D/t) \) ranging from 15 to 40. It should be noted that, in thick pipes \( (15 < D/t < 20) \), the collapse mechanism is inelastic, and thus Eq. (5.1) may not yield accurate results (Ju and Kyriakides, 1991; Fraldi and Guarracino, 2011; Fraldi et al., 2011; Yeh and Kyriakides, 1986). In single pipelines, once the buckle is triggered in the pipe, the pipe cross-section is rapidly transformed into a dog-bone shape. The buckle then travels along the pipeline as long as the external pressure is high enough to sustain propagation. The lowest pressure required to perpetuate the buckle is termed propagation pressure, \( P_p \), which is only a fraction of the collapse pressure. The collapse and propagation of buckling in single pipelines have been extensively investigated using analytical, experimental, and numerical methods. Most notable are the analytical studies by Mesloh et al. (Mesloh et al., 1973) and Palmer and Martin (Palmer and Martin, 1975), the experimental and numerical investigations by Kyriakides and Babcock (Kyriakides et al., 1984) and Albermani et al. (Albermani et al., 2011), the study of collapse pressure under confined buckling (Stephan et al., 2016), and investigations of interaction between global buckling and propagation buckling of submarine pipelines (Karampour et al., 2013; Karampour and Albermani, 2016; Karampour and Albermani, 2014; Karampour et al., 2015).
Unlike single pipelines, collapse mechanisms of PIPs have only been marginally addressed (Kyriakides, 2002; Kyriakides and Vogler, 2002; Gong and Li, 2015; Karampour et al., 2017; Li, 2018; Alrsai et al., 2018). Moreover, these studies have been purely focused on the buckle propagation pressure ($P_{p2}$) of the PIP systems. The existing knowledge on buckling of single-walled pipelines under external pressure can be used to predict the collapse pressure of the outer pipe of a PIP system. However, as will be discussed later in this chapter, the buckling mechanisms of the inner pipe and its collapse pressure (referred to as $P_{ci}$ in this chapter) are different from those of a single pipeline. To the authors’ knowledge, there is no existing study on collapse of the inner pipe of a PIP system under external pressure.

This chapter aims to provide insight on buckling mechanisms and capacity of a non-pressurised inner pipe within a PIP system, following the collapse of the outer pipe under external pressure. In section 5.2 experimental results from hyperbaric chamber tests of three PIPs with different outer pipes and identical inner pipes are presented. In section 5.3, a parametric study on the collapse pressure of the inner pipe ($P_{ci}$) is conducted using validated FE analyses, and an empirical expression for $P_{ci}$ is provided. The buckle mechanisms and accuracy of the proposed empirical equation in comparison with the experimental results are discussed in section 5.4. The chapter is concluded with a brief outline of the significant outcomes of the study.

5.2 Collapse of Pipe-in-pipe Systems under External Pressure: Experimental Observations and Validation of the Finite Element Analysis

5.2.1 Mechanical properties of the PIPs

Three sets of concentric aluminium (Al-6060-T5) PIP systems with parameters given in Table 5.1 were selected for the experimental study. To compare the collapse pressures of the inner pipes ($P_{ci}$) from the three PIPs, identical inner pipes were adopted. The diameter to thickness ratio of outer and inner pipes ($D_o/t_o$ and $D_i/t_i$), designated by subscript “o” and “i” for outer and inner pipe respectively, are between 25-40 which is the practical range in offshore pipeline application. The stress-strain history of the aluminium tubes was obtained from tensile tests conducted on coupon samples (transverse strips), cut from the tubes and having the full thickness of the wall tube according to AS1391-2007 (AS 1391, 2017). The stress-
strain curve of the 80x2 mm aluminium tube is depicted in Figure 5.1(a). Since the coupon strips cut from the tube are not straightened, the modulus of elasticity obtained from such tensile test may not always be accurate. Thus, the modulus of elasticity of the samples was obtained from compressive tests of stub columns with length equal to the tube diameter ($D$), as shown in Figure 5.1(b). According to AS1391-2007 (AS 1391, 2017), the length of the stub column should be at least equal to $D/4$. The modulus of elasticity ($E$) of the samples listed in Table 5.1 was obtained from two compressive stub tests (Figure 5.1(b)) conducted for each $D/t$. The material tangent modulus of $E' = 1\%$ was adopted for the inner and outer pipes. Previous studies (Albermani et al., 2011; Karampour et al., 2017; Kamalarasa and Calladine, 1988) have shown that the ring squash test is a reliable method to calculate the yield stress in metallic tubes. Therefore, the ring squash test was utilised herein to obtain the yield stress of the samples. The yield stresses were calculated based on results of two ring squash tests (RST) shown in Figure 5.2. The ring squash test (RST) (Albermani et al., 2011; Karampour et al., 2017; Kamalarasa and Calladine, 1988) is conducted on a short segment of the pipe specimen compressed between two rigid indenters of the same diameter as the pipe specimen (Figure 5.2). The yield stress, $\sigma_y$, is calculated from

$$\sigma_y = \frac{F_o D}{2L_{RST} t^2}$$

where $F_o$ is the RST load shown in Figure 5.2 at which the four plastic hinges are developed in the pipe wall. $L_{RST}$ is the length of the RST sample which is 150mm.

Table 5.1 Geometric and material parameters of PIP systems tested in the hyperbaric chamber.

<table>
<thead>
<tr>
<th>ID</th>
<th>Material</th>
<th>$D$ (mm)</th>
<th>$t$ (mm)</th>
<th>$D_o/t_o$</th>
<th>$D_i/t_i$</th>
<th>$D_o/D_o$</th>
<th>$t_o/t_o$</th>
<th>$E$ (MPa)</th>
<th>$E'/E$ (%)</th>
<th>$\sigma_y$ (MPa)</th>
<th>$\sigma_{y_i}/\sigma_{y_o}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>PIP-1 Outer pipe</td>
<td>60</td>
<td>2.0</td>
<td>30.0</td>
<td>25.0</td>
<td>1.20</td>
<td>0.67</td>
<td>0.80</td>
<td>66,680</td>
<td>1.0</td>
<td>139</td>
<td>1.12</td>
</tr>
<tr>
<td>PIP-1 Inner pipe</td>
<td>40</td>
<td>1.6</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PIP-2 Outer pipe</td>
<td>80</td>
<td>2.0</td>
<td>40.0</td>
<td>25.0</td>
<td>1.60</td>
<td>0.50</td>
<td>0.80</td>
<td>66,680</td>
<td>1.0</td>
<td>169</td>
<td>0.93</td>
</tr>
<tr>
<td>PIP-2 Inner pipe</td>
<td>40</td>
<td>1.6</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PIP-3 Outer pipe</td>
<td>80</td>
<td>3.0</td>
<td>26.7</td>
<td>25.0</td>
<td>1.07</td>
<td>0.50</td>
<td>0.53</td>
<td>66,680</td>
<td>1.0</td>
<td>209</td>
<td>0.75</td>
</tr>
<tr>
<td>PIP-3 Inner pipe</td>
<td>40</td>
<td>1.6</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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</tbody>
</table>
5.2.2 Hyperbaric chamber tests

The experimental study on collapse of PIPs under external pressure is carried out in a specially designed and fabricated hyperbaric chamber shown in Figure 5.3(a). The chamber has an inner diameter of 173 mm and a length of 4 m and is rated for working pressure of 20 MPa (2000 m water depth). Three sets of concentric aluminium (Al-6060-T5) PIP systems with parameters given in Table 5.1 and length of 1.6m i.e. $L/D_o > 20$, were end-sealed and pressurized inside the hyperbaric chamber. To end-seal the PIP system, thick aluminium discs were glued to the ends, ensuring that the inner and outer pipes were concentric and that the inner pipe was completely sealed from the outer pipe. To measure the collapse pressures in
the outer \((P_{co})\) and inner pipes \((P_{ci})\), a set of two valves were connected to the end of each of the outer and inner pipes of the PIP system. For each pipe, one valve was used for bleeding the pipe while filling it with water before the test commences. The second valve was utilized to vent each of the carrier and inner pipes, as well as to separately collect water from the inner pipe and the cavity between the inner and outer pipes during the buckle propagation (through the red and black hoses shown in Figure 5.3(b)). Volume-Controlled pressurization with a high-pressure pump (shown in Figure 5.3(a)) was used. The change in volume of the system \((\Delta V)\) during the test was calculated by measuring the weight of water being discharged from the inner pipe and the cavity between the pipes separately using digital weighing scales shown in Figure 5.3(a).

![Figure 5-2](image)

**Figure 5-2**: Ring squash test results, single pipe (60×2 mm), Initial state 1 (top left), Collapse state 2 (top right) and load-deflection response.
Figure 5.4 shows the experimental pressure-volume change response for the outer and inner pipes of PIP-1, 2 and 3, separately. The pressure levels inside the chamber are plotted against the normalized change in volume of the outer pipe, (change in the volume of the annulus and the inner pipe), and that of the inner pipe, separately. Each test was conducted three times, and the curves in Figure 5.4 show the average values of three tests. As shown in Figure 5.4, the pressure initially rises sharply until the collapse pressure $P_{co}$, is reached at which a section
of the outer pipe collapses. Following the collapse of the outer pipe the pressure inside the chamber drops drastically until the outer and the inner pipes come into contact at a pressure level $P_{ini}$, not shown in the figure. This is followed by the collapse of the inner pipe at collapse pressure $P_{ci}$. The collapse progresses in the outer and inner pipes and quickly spreads throughout the length of the outer and inner pipes at a slightly lower pressure $P_{p2}$, known as the propagation pressure of the PIP system (Kyriakides, 2002; Gong and Li, 2015; Karampour et al., 2017; Alrsai et al., 2018) as explained in the previous chapters. Longitudinal and cross-sectional views of the deformed shape of PIP-2 are shown in Figure 5.4(b). Other tested PIPs showed similar deformed patterns.

![Figure 5-4](image)

**Figure 5-4:** Hyperbaric chamber test results of PIP-1, 2 and 3 in Table 5.1, showing (a) pressure against change in volume normalised to initial volume of the PIP system and (b) the deformed shape of PIP-2.
The collapse pressure in the outer pipe $P_{co}$, is very sensitive to imperfections in the cross-section of the outer pipe (Fraldi and Guarracino, 2011; Fraldi, et al., 2011; Yeh and Kyriakides, 1986) which is defined in terms of ovalization ratio $\Delta_o$ (DNV-OS-F101, 2017):

$$\Delta_o = \frac{D_{max} - D_{min}}{D}$$

(5.3)

where $D_{max}$ and $D_{min}$ are the maximum and minimum diameters of the outer pipe, and $D$ is the nominal outside diameter represented in Table 5.1. The initial ovality of outer pipes of the tested PIPs were measured as $\Delta_o = 0.5\%$, 0.2% and 0.6% for PIP-1, 2 and 3, respectively. The collapse pressures ($P_{co}$) of PIP-1, 2 and 3 (averages of three tests) are 13%, 1% and 16% lower than the corresponding elastic collapse pressures predicted from Eq. (5.1) for the outer pipes of each PIP system respectively.

The collapse pressures of the inner pipes ($P_{ci}$) measured in the tests are 1645, 844 and 2050 kPa for PIP-1, 2 and 3 respectively. These pressures are not the same and are different from the collapse pressure 9578 kPa, predicted from Eq. (5.1). It is observed that although the inner pipes are identical in the three PIP systems used here, their collapse pressures ($P_{ci}$) are considerably different. It suggests that the pressure at which the inner pipe collapses is related to the geometric and material properties of the outer pipe of the PIP system. The relation between the collapse pressure of the inner pipe and the parameters of the outer pipe of the PIP system will be discussed in sections 5.3 and 5.4 of this chapter.

### 5.2.3 Finite element model and validation against experimental results of PIP-1

Numerical simulation of the collapse of PIP-1 tested in the hyperbaric chamber with parameters represented in Table 5.1 and length of 1.6m was conducted in ANSYS (ANSYS 17.0). Thin four-node shell elements (181) were used to model the carrier and the inner pipes. Frictionless contact elements (174 and 170) were used in three pairs to define the non-linear contact between the carrier and inner pipes and the inner surfaces of the inner pipe wall. These contact pairs are shown in Figure 5.5(b). The first and second pairs are defined between the inner surfaces of the outer pipe and the outer surface of the inner pipe. The third pair considers the contact between the inner top and bottom surfaces of the inner pipe. Because of
symmetry, a one-half model of the pipe wall (180°) was discretised using 24 elements in the circumferential direction with seven integration points through the thickness of the carrier and inner pipes. In order to validate the numerical model, the initial ovality of $\Delta_o = 0.5\%$ (same as that measured in the test sample and defined in Eq. (5.3)) was introduced at mid-length on the carrier pipe of PIP-1 in the FE model. The nodes at either end of the PIP system were restrained from translation in all directions. A Von-Mises elastoplastic (bilinear) material definition depicted in Figure 4.1(a) and with parameters are shown in Table 5.1 was adopted. To overcome convergence issues, the Arc-Length method (Riks, 1979) which is a strong tool for solving non-linear snap-through buckling problems was used.

The FE response of PIP-1 is plotted in Figure 5.5(a), corresponding deformed configurations are shown in Figure 5.5(b), and show good agreement with the experimental results. The collapse pressure of the outer pipe ($P_{co}$), of the inner pipe ($P_{ci}$) and the propagation pressure ($P_{p2}$) of PIP-1 from the FE models are presented in Table 5.2 together with the experimental results. These experimental results are the average of three repeats of hyperbaric chamber tests of PIP-1. Discrepancies between the experimental pressures and FE predictions are negligible.

<table>
<thead>
<tr>
<th></th>
<th>Experimental</th>
<th>FEA</th>
<th>$FE/Exp$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{co}$ (kPa)</td>
<td>4820</td>
<td>4795</td>
<td>0.99</td>
</tr>
<tr>
<td>$P_{ci}$ (kPa)</td>
<td>1645</td>
<td>1539</td>
<td>0.94</td>
</tr>
<tr>
<td>$P_{p2}$ (kPa)</td>
<td>1610</td>
<td>1511</td>
<td>0.94</td>
</tr>
</tbody>
</table>

**Table 5.2** Comparison between experimental and numerical results for PIP-1.
5.2.4 Influence of ovality on collapse pressure of the inner pipe

The dependency of the collapse pressure of the inner pipe ($P_{ci}$) to parameters of the outer pipe will be discussed in the next section using a comprehensive parametric study. Prior to the parametric study, the effect of initial imperfection on collapse of a single pipe and that of an inner pipe of a PIP system is investigated using the validated FE analysis. According to recommendations of DNV-OS-F101 (DNV-OS-F101, 2017), ovalities between 0.5% and 3% are adopted in the FE models. Collapse pressures of single pipelines with $D/t = 15$ and 40 and material properties of Table 5.1, are plotted against ovality ($\Delta o$) and are shown with dashed lines in Figure 5.6. The collapse pressures are normalised to the elastic critical pressure (Eq. (5.1)). As expected from previous studies (Karampour et al., 2013; Gong et al., 2013; He et al., 2014) the collapse pressure of the pipeline decreases with a corresponding increase in the cross-section ovality. Also, it’s worth noting that the expression for elastic buckling pressure (Eq. (5.1)) does not represent the buckle pressure of the thick single pipe ($D/t = 15$) accurately. Normalised collapse pressures of the inner pipe ($P_{ci}$) of PIP systems with the same

Figure 5-5: (a) Pressure versus normalized change in volume for PIP-1 in Table 5.1, (b) Deformed shape of the FE model and the frictionless contact pairs.
inner pipe (same as single pipes studied in Figure 5.6), and outer pipes with parameters shown, are depicted in Figure 5.6 with solid lines. It is evident from Figure 5.6 that unlike a single pipeline, the collapse pressure of the inner pipe \( (P_{ci}) \) within the PIP system is not sensitive to imperfections.

![Figure 5-6: Normalized collapse pressure vs. ovality curves for single pipe and inner pipe of PIP systems of different \( D_i/t_i \).](image)

5.3 Parametric Study on Collapse Pressure of the Inner Pipe \( (P_{ci}) \) of PIPs Using Finite Element Analysis

The hyperbaric chamber results disused in the previous section suggest that the collapse pressure of the inner pipe of the PIP system, \( (P_{ci}) \), is a function of geometric and material parameters of both inner and outer pipes:

\[
P_{ci} = F \left( D_o, t_o, \sigma_{yo}, E_o, E_{o}', D_i, t_i, \sigma_{yi}, E_i, E_{i}' \right).
\]  

(5.4)
In the parametric study carried out herein, both outer and inner pipes are assumed to be of the same material i.e. Aluminium (Al-6060-T5). Therefore, the same modulus of elasticity \( E \) represented in Table 5.1 and the Poisson’s ratio \( \nu = 0.33 \) are adopted for all PIPs. Using dimensional analysis, Eq. (5.4) can be written in terms of three non-dimensional geometric groups and two non-dimensional material groups:

\[
\frac{P_{ci}^{*}}{P_{cr}^{*}} = f \left( \frac{D_i}{D_o}, \frac{t_i}{t_o}, \frac{E_i}{E_o}, \frac{\sigma_{yi}}{\sigma_{yo}} \right) \tag{5.5}
\]

Obviously, \( D_i/t_i \) can be presented in terms of the other two geometric groups in Eq. (5.5). However, since the collapse pressure of single pipeline is significantly related to the diameter to thickness ratio (Eq. (5.1)), this term needs to be accounted for in the proposed empirical expression. Knowing that \( P_{ci}/P_{cr} < 1 \), the following constitutional power-law is considered:

\[
\frac{P_{ci}}{P_{cr}} = A_l \left( \frac{D_i}{D_o} \right)^a \left( \frac{t_i}{t_o} \right)^b \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^c \left( \frac{E_i}{E_o} \right)^d \tag{5.6}
\]

Using the validated FE model, a parametric study is conducted herein to establish the best estimates for the unknown power coefficients of Eq. (5.6).

### 5.3.1 Effect of \( D_i/D_o \)

To explore the influence of the annulus on the collapse pressure of the inner pipe of PIPs, the thickness ratios are varied while the material properties of the two pipes are kept identical. Figure 5.7 shows normalised collapse pressures of the inner pipes versus \( D_i/D_o \) for four different \( D_i/t_i \) ratios of 15, 25, 30 and 40 and in two sets of \( t_i/t_o \) i.e. 0.8 and 1. The parametric study covers a practical range of \( D_i/D_o \) between 0.4 and 0.9. A relatively cubic nonlinear relationship is observed between normalised pressure and diameter ratio for both \( t_i/t_o \) values.

By decreasing the annulus, the collapse pressure of the inner pipe increases considerably, and this is more evident in the PIPs with thin inner pipes (larger \( D_i/t_i \)).

For a fixed value of \( t_i/t_o \) and assuming identical yield stress and tangent modulus in the outer and inner pipes, Eq. (5.6) is reduced to
Using non-linear least-squares fitting for eight sets of data in Figure 5.7, the coefficients $a = 3.2$ and $A_2 = 0.595$ are obtained.

\[
\frac{P_{ci}}{P_{cr}} = A_2 \left( \frac{D_i}{D_o} \right)^a
\]  

**Figure 5-7:** Collapse pressure of the inner pipe as a function of $D_i/D_o$ with (a) $t_i/t_o = 0.8$ and (b) $t_i/t_o = 1.0$. 

(a) $t_i/t_o = 0.8; \sigma_T/\sigma_{T_o} = 1.0; E'/E'_o = 1.0$

(b) $t_i/t_o = 1.0; \sigma_T/\sigma_{T_o} = 1.0; E'/E'_o = 1.0$
5.3.2 Effect of $t/t_o$

Effect of wall thickness on the collapse pressure of the inner pipe of PIPs is studied for a range of $t/t_o$ between 0.5 to 1.2 and two sets of $D_i/D_o$. Normalised pressures are plotted against normalised thicknesses in Figure 5.8. To generate data-points for each $D_i/t_i$, the diameter of the outer pipe of the PIP system is kept constant (to retain the $D_i/D_o$), and its thickness is changed. In all PIPs, reduction in the thickness of the outer pipe results in a decrease in the collapse pressure of the inner pipe. The decreasing trend in collapse pressure $P_{ci}$, becomes less pronounced as $t/t_o$ approaches 1. Given the material properties of the outer and inner pipes are identical, Eq. (5.6) can be written as

$$
\frac{P_{ci}}{P_{cr}} = A_3 \left( \frac{D_i}{D_o} \right)^a \left( \frac{t_i}{t_o} \right)^b
$$

(5.8)

Using the non-linear least square fitting of the data in Figure 5.8, the coefficients $b = -1.88$ and $A_3 = 0.472$ are obtained.
5.3.3 Effect of $D_i/t_i$

Figure 5.9 shows the normalised pressure against $D_i/t_i$ ratio, for three different $D_i/D_o$ and in two sets of $t_i/t_o$. The normalised collapse pressure of the inner pipe increases as the inner pipe gets thinner. This is more pronounced at larger $D_i/D_o$. Given the material properties of the outer and inner pipes are identical, Eq. (5.6) can be written as

$$\frac{P_{ci}}{P_{cr}} = A_4 \left( \frac{D_i}{D_o} \right)^a \left( \frac{D_i}{t_i} \right)^c$$

(5.9)

Based on the non-linear least-squares fitting of the data, the coefficients $c = 0.64$ and $A_4 = 0.093$ are obtained.
Figure 5-9: Collapse pressure of the inner pipe as a function of $D_i/t_i$ with (a) $t_i/t_o = 1.0$ and (b) $t_i/t_o = 0.8$. 

$D_i/D_o = 0.4$  
$D_i/D_o = 0.7$  
$D_i/D_o = 0.9$
5.3.4 Effect of $\sigma_{yi}/\sigma_{yo}$

PIPs with $\sigma_{yi}/\sigma_{yo}$ between 0.6 and 1.4, $t_i/t_o$ and $E'/E'_o$ equal to 1 and in two sets of $D_i/D_o$ were modelled and the results are shown in Figure 5.10. In all PIPs shown in Figure 5.10, the collapse pressure of the inner pipe plummets when the yield stress of the outer pipe decreases. To identify the power exponent of $\sigma_{yi}/\sigma_{yo}$, Eq. (5.6) can be represented as

$$\frac{P_{ci}}{P_{cr}} = A_5 \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^d$$

(5.10)

Based on the total eight sets of data presented in Figure 5.10, and using non-linear regression, $d = -0.6$ and $A_5 = 0.474$ are calculated.

![Graph showing the relationship between $\sigma_{yi}/\sigma_{yo}$ and $P_{ci}/P_{cr}$ for different values of $D_i/D_o$, $t_i/t_o$, and $E'/E'_o$.]
Figure 5-10: Collapse pressure of the inner pipe as a function of $\sigma_{yi}/\sigma_{yo}$ with (a) $D_i/D_o = 0.5$ and (b) $D_i/D_o = 0.8$.

5.3.5 Effect of $E'_{i}/E'_{o}$

Previous studies (Fraldi and Guarracino, 2011; Fraldi, et al., 2011; Yeh and Kyriakides, 1986) have shown that the pipeline may experience collapse under external pressure, at stress levels beyond the proportional limit. Thus, Eq. (5.1) is normally modified to account for material nonlinearities by incorporating the tangent modulus ($E'$) (Fraldi and Guarracino, 2011; Fraldi, et al., 2011). In order to investigate the effect of the tangent modulus on the collapse of the inner pipe, the geometric and yield stress parameters of outer and inner pipes are kept identical and the $E'_{i}/E'_{o}$ is altered. Results are plotted in Figure 5.11 for various $D_i/t_i$ and with $D_i/D_o = 0.5$. As evident, compared to the thickness, diameter and yield ratios, the tangent modulus has little impact on the collapse pressure $P_{ci}$. Collapse of PIPs with $D_i/D_o=0.8$ was modelled (not shown here) and no significant changes in results were observed. Knowing that except the tangent modulus, other parameters are unchanged, Eq. (5.6) can be re-written as

\[
\frac{P_{ci}}{P_{cr}} = A_6 \left( \frac{E_i}{E_o} \right)^e
\]  

(5.11)

where $e = -0.3$ and $A_6 = 0.591$ are obtained from least square nonlinear regression.
5.3.6 Empirical expression for collapse pressure $P_{ci}$ of PIPs

The parametric study carried out in previous sections ascertained the dependency of the collapse pressure $P_{ci}$ of the PIP systems on geometric and material parameters of the outer and inner pipes. Based on the results of the parametric study in the previous section and using non-linear square fits of sets of data taken from the FE results, the following normalised expression is derived for the collapse pressure of the inner pipe of PIPs

$$
\frac{P_{ci}}{P_{cr}} = 0.05 \left( \frac{D_i}{D_o} \right)^{3.2} \left( \frac{t_i}{t_o} \right)^{-1.88} \left( \frac{D_i}{t_i} \right)^{0.68} \left( \frac{\sigma_{yi}}{\sigma_{yo}} \right)^{-0.6} \left( \frac{E_i}{E_o} \right)^{-0.3}
$$

(5.12)

The coefficient (0.05) in Eq. (5.12) is determined using the Levenberg-Marquardt algorithm with a correlation factor ($R^2$) of 0.9882. Comparison between the FE results and the proposed expression (Eq. (5.12)) is depicted in Figure 5.12 for the studied range of $D_i/t_i$. The maximum difference between FE results and empirical expression (Eq. (5.12)) is less than 6.0%. The normalised collapse pressures obtained from the proposed empirical expression (Eq. (5.12)) and those acquired from the hyperbaric chamber for the tested PIPs are represented in Table

Figure 5.11: Collapse pressure of the inner pipe as a function of $E'/E'_o$ with $D_i/D_o = 0.5$. 

[Graph showing the relationship between collapse pressure and $E'/E'_o$ for different values of $D_i/D_o$.]
5.3. The differences are less than 5%. As represented in the last column of Table 5.3, the empirical expression predicts the experimental results with good accuracy.

![Figure 5-12](image.png)

**Figure 5-12:** Comparison between FE results and those predicted by Eq. (5.12).

<table>
<thead>
<tr>
<th></th>
<th>( P_{ci}/P_{cr} ) (Eq. (5.12))</th>
<th>( P_{ci}/P_{cr} ) (Exp.)</th>
<th>Difference (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PIP-1</td>
<td>0.173</td>
<td>0.166</td>
<td>4.05</td>
</tr>
<tr>
<td>PIP-2</td>
<td>0.077</td>
<td>0.077</td>
<td>0.00</td>
</tr>
<tr>
<td>PIP-3</td>
<td>0.188</td>
<td>0.184</td>
<td>2.13</td>
</tr>
</tbody>
</table>
5.4 Discussion of Results

In Section 5.3, a comprehensive parametric study was conducted using the validated FE model to predict the collapse pressures \( P_{ci} \) of PIP systems with various wall thickness \( t_i/t_o \), diameter \( D_i/D_o \), diameter to thickness \( D_i/t_i \), material yield stress \( \sigma_Y/\sigma_{Yo} \), and tangent modulus \( E'/E'_{o} \) ratios. After reviewing the results of the parametric dependence of collapse pressures \( P_{ci} \) of PIPs, it is worth discussing the collapse modes of the inner pipe observed in the FE simulations. Based on results of FE analyses with various parameters adopted for the outer and inner pipes, two dominant modes of failure under external pressure were observed in the PIPs.

The collapse pressure and deformed shapes of PIPs with \( D_i/t_i \) of 25 and 30 are shown in Figures 5.13(a) and (b) respectively. In Figure 5.13(a), the change in the pressure of the system is plotted against the ovality of the inner pipe of the PIP. The collapse mode \( A \) is shown for a PIP system with \( D_i/t_i \) of 30, thickness ratio of \( t_i/t_o=0.8 \) and with identical materials properties in outer and inner pipes. In Mode \( A \), by increasing the external pressure, the outer pipe collapses and gradually deforms into the deformed shape (I), at a pressure level called initial pressure \( P_{ini} \). At this stage, the outer and inner pipes come into contact. Following the touchdown, the pressure stays at the same level (stage II), and the inner pipe collapses at \( P_{ci} \) almost equal to \( P_{ini} \). Then, the collapse is propagated along the lengths of the outer and inner pipes simultaneously as shown in stages (III) to (IV) in Figure 5.13(b) for Mode \( A \).

Mode \( B \) is depicted for the PIP system with \( D_i/t_i \) of 25 and \( t_i/t_o \) of 1.0. The mechanical properties of the inner and outer pipes are alike. In mode \( B \), following the initiation of collapse in the outer pipe, the pressure in the system plunges. Since the corresponding change in ovality of inner pipe only is analysed herein, the associated drop in pressure due to collapse of the outer pipe is not shown in Figure 5.13(a). At pressure level \( P_{ini} \), (stages (I) of Figure 5.13(c)) the outer and the inner pipes touch, and the collapse is propagated along the PIP but only in the outer pipe. In the vicinity of the end-caps, higher pressure is required to perpetuate the collapse in the outer pipe. However, the increase in pressure causes a collapse in the inner pipe as shown in stage (II), at a higher pressure \( P_{ci} \), shown in Figure 5.13(a). The collapse is
then propagated through the length of the inner pipe (stages (III and IV)). As suggested in Figure 5.13(a), the collapse pressure $P_{ci}$ is substantially larger than the initial pressure $P_{ini}$, in failure mode $B$, compared to mode $A$.

**Figure 5-13:** Finite element results showing (a) pressure against normalized ovality, (b) corresponding deformed shapes of the inner pipe of PIP system exhibiting collapse mode $A$ and; (c) mode $B$. 

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To understand the distinction between modes $A$ and $B$, collapse mechanisms of PIPs with outer and inner pipes of similar mechanical properties are investigated. Collapse pressures normalized by the initial pressures of the inner pipe of PIP systems with $\sigma_{yi}/\sigma_{yo} = 1.0$, $E'_i/E'_o = 1.0$ and thickness ratios between 0.3 and 1.2 are presented in Figure 5.14. The results correspond to four sets of $D_i/t_i$ and two sets of $D_i/D_o$. With respect to Figure 5.13, the collapse modes $A$ and $B$ are associated with $P_{ci}/P_{ini} \approx 1$ and $P_{ci}/P_{ini} > 1$ and correspond to $t_i/t_o \leq 0.7$ and $t_i/t_o > 0.7$ respectively as shown in Figure 5.14. It can be inferred from Figure 5.14, that in PIPs with outer and inner pipes of similar material properties, the separation between collapse modes $A$ and $B$ occurs at thickness ratio $t_i/t_o = 0.7$. Regardless of $D_i/t_i$ ratio, the upsurge in $P_{ci}/P_{ini}$ at $t_i/t_o > 0.7$, is more pronounced in PIPs with smaller $D_i/D_o$ (i.e. larger annulus).

**Figure 5.14:** Collapse pressures normalized by the initiation pressures of the inner pipe, showing effect of $t_i/t_o$.

Figure 5.15 shows the normalized collapse pressures of the inner pipe against the normalized diameter-to-thickness ratios of outer and inner pipes, for 150 PIPs with $0.3 \leq t_i/t_o \leq 1.2$ and $0.4 \leq D_i/D_o \leq 0.9$. As suggested by the trend in Figure 5.15, when the outer pipe of the PIP is
thinner than the inner pipe, \( (D_o/t_o > D_i/t_i) \), failure mode \( B \) occurs in the PIP. The distinction between failure modes \( A \) and \( B \) corresponds to \( D_o/t_o = 1.25 \ D_i/t_i \) in Figure 5.15.

The upper and lower bounds of data points in Figure 5.15 can be estimated, assuming the material properties of the outer and inner pipe are identical. Thus, Eq. (5.12) can be written as

\[
\frac{P_{ci}}{P_{cr}} = 0.05 \left( \frac{D_i}{D_o} \right)^{1.32} \left( \frac{D_i}{t_i} \right)^{0.68} \left( \frac{D_o/t_o}{D_i/t_i} \right)^{-1.88} \tag{5.13}
\]

In the current study and in practical PIPs, \( D_i/D_o \) and \( D_i/t_i \) range between 0.4-0.9 and 15-40, respectively. Inserting \( D_i/D_o = 0.9 \) and \( D_i/t_i = 40 \) in Eq. (5.13), the upper bound can be predicted by Eq. (5.14):

\[
\frac{P_{ci}}{P_{cr}} = 0.534 \left( \frac{D_o/t_o}{D_i/t_i} \right)^{-1.88} \quad ; \quad 0.5 < \frac{(D_i / t_o)}{(D_i / t_i)} < 2.0 \tag{5.14}
\]

The lower bound is obtained by adopting \( D_i/D_o = 0.4 \) and \( D_i/t_i = 15 \) in Eq. (5.13), and can be shown with the following expression:

\[
\frac{P_{ci}}{P_{cr}} = 0.095 \left( \frac{D_o/t_o}{D_i/t_i} \right)^{-1.88} \quad ; \quad 0.5 < \frac{(D_i / t_o)}{(D_i / t_i)} < 2.0 \tag{5.15}
\]

Eqs. (5.14) and (5.15) are shown with solid lines in Figure 5.15 and provide upper and lower bounds of the FE results.
5.5 Summary

The buckling mechanisms and capacity of the inner pipe of the PIP system, under external pressure, were investigated experimentally and numerically herein. The hyperbaric chamber test results showed that in PIPs with identical inner pipes and different outer pipes, the inner pipe may collapse at different pressures ($P_{ci}$). To find an expression for $P_{ci}$, a comprehensive parametric study was conducted using validated nonlinear geometric and material FE analyses. In the parametric study, the effect of the wall thickness $t_i/t_o$, diameter $D_i/D_o$, diameter to thickness $D_i/t_i$, material yield stress $\sigma_{yi}/\sigma_{yo}$, and tangent modulus $E'_{yi}/E'_{yo}$ ratios, on collapse pressure of the inner pipes were investigated. It was understood that unlike single pipelines, the collapse pressure of the inner pipe is not significantly affected by the ovality in the cross-section of the inner pipe. Results showed that the collapse pressure of the inner pipe ($P_{ci}$), drastically drops with the corresponding decrease in $D_i/D_o$ (PIP with larger annulus), or

![Figure 5-15: Collapse pressures of the inner pipe against the normalized diameter-to-thickness ratios of outer and inner pipes of PIPs.](image_url)
with corresponding increase in $t_i/t_o$ (PIP with thick outer pipe). These decreasing trends were shown to be more pronounced in PIPs with thin inner pipes ($30 < D_i/t_i < 40$). The effects of material properties such as $\sigma_{Yi}/\sigma_{Yo}$ and $E'/E'_o$, on the collapse pressure of the inner pipe, were shown to be less significant. Two buckling modes were observed in the FE results. In mode $A$, the inner pipe collapses straight after the outer and inner pipes make contact. However, in mode $B$, after the two pipes touch, the buckle propagates in the outer pipe first and collapse of the inner pipe occurs at a greater pressure. It was shown that in PIPs with outer and inner pipes of similar material properties, the separation between collapse modes $A$ and $B$ occurs at thickness ratio $t_i/t_o = 0.7$. Also, a thorough examination of all buckle modes showed that the distinction between failure modes $A$ and $B$ corresponds to $D_o/t_o = 1.25 D_i/t_i$. This is confirmed through observation of failure mode $B$ in hyperbaric chamber response of PIP-2 (Figure 5.4), with $t_i/t_o = 0.8$, $D_o/t_o = 1.60 D_i/t_i$, and $\sigma_{Yi}/\sigma_{Yo}$ almost equal to one. An empirical expression for $P_{ci}$ was proposed and was shown to be in good agreement with hyperbaric chamber test results. Through combining all FE results, expressions for upper and lower bounds of the collapse pressure of the inner pipes were advised.
Statement of contribution to co-authored published paper

This chapter includes a co-authored and peer-reviewed paper. The bibliographic details of the co-authored paper, including all authors, are:


The paper has been reformatted to meet the guidelines of the thesis. Minor explanation has been added in the paper for further clarity.

My contribution to this paper involved: literature review, experimental works, result analysis, discussion of the results, writing, editing.

(Signed) _________________________________ Date: 23.01.2020
PhD candidate: Mahmoud Alrsai.

(Countersigned) ___________________________ Date: 23.01.2020
Principal Supervisor: Dr Hassan Karampour.

(Countersigned) ___________________________ Date: 23.01.2020
Associate Supervisor: Dr Sanaul Chowdhury.

(Countersigned) ___________________________ Date: 23.01.2020
External Supervisor: Professor Faris Albermani.
Carbon Fibre Buckle Arrestors for Offshore Pipelines

Abstract:
Feasibility and efficiency of using carbon fibre reinforced polymer (CFRP) buckle arrestors in steel offshore pipelines with $D/t$ of 28 and 40 are investigated using hyperbaric chamber tests. CFRP arrestors are manufactured using Prepreg (PP), Wet-Layup (WL) and Vacuum Bagging (VB) curing methods, with coarse and fine sand surface preparations. A parametric study is performed that outlines the performance of CFRP arrestors in various geometric configurations. Efficiency of CFRP arrestors using different manufacturing methods and various geometric configurations are calculated and compared with those of conventional steel buckle arrestors. It is shown that the efficiencies of CFRP arrestors vary between 0.74 and 1.0 for different manufacturing methods. Optimum efficiencies are obtained in the WL technique, using fine sanding, with CFRP arrestor of thickness twice the steel pipe-wall thickness, and fibres oriented in the hoop direction. Results show that at similar efficiencies, the CFRP arrestors can be much thinner than conventional slip-on or integral arrestors.

Keywords
Offshore pipelines; buckle propagation; buckle arrestor; collapse under external pressure; carbon fibre; composites

6.1 Introduction
A major concern in design of subsea pipelines in deep and ultra-deep waters is the collapse under external hydrostatic pressure. In the presence of local damage in the pipe-wall (in the form of out-of-roundness, dents or corrosion) local collapse can be initiated in the pipeline. Once initiated, the circular cross-section of the pipe transforms into a dog-bone shape, and eventually a flat shape (causing shutdown of the pipeline), as the buckle rapidly propagates along the pipeline (Xue and Fatt, 2001; Albermani, et al., 2011; Karampour et al., 2017; Alrsai et al., 2018; He et al., 2014; Alrsai et al., 2018). The corresponding local collapse due
to external pressure may be coupled with other loadings in the pipeline, such as bending and axial force, resulting in buckle interaction (Karampour et al., 2013; Karampour and Albermani, 2016; Karampour and Albermani, 2014; Karampour et al., 2015; Karampour, 2018). The lowest pressure required to perpetuate the local buckle is termed propagation pressure ($P_P$), which is typically only 15% of the collapse pressure ($P_c$). In case the external pressure exceeds the propagation buckle criterion (DNV G., 2017), buckle arrestors are installed at certain intervals along the pipeline based on cost and spare pipe philosophy. Such arrestors are snugly fitted around the pipeline to limit the damage and safeguard the downstream section of the pipeline (Netto and Estefen, 1996; Kyriakides et al., 1998; Lee and Kyriakides, 2004).

Existing buckle arrestors are made of stiff metal rings which locally augment the circumferential stiffness of the pipeline, and thus hinder the buckle propagation. A buckle arrestor can halt the buckle completely or may allow the buckle to cross-over at a higher pressure. The buckle arrestor pressure capacity ($P_X$) is closely related to the length $L$, thickness $h$ and yield stress $\sigma_{ya}$ of the arrestor as well as diameter $D$, wall thickness $t$, and yield stress $\sigma_y$ of the pipe. Efficiency of a buckle arrestor ($\eta$), is defined as (Kyriakides and Babcock, 1979)

$$\eta = \frac{P_X - P_P}{P_C - P_P}$$

(6.1)

where $P_P$ and $P_c$ are the propagation and collapse pressures of the adjacent pipe, respectively. An efficiency of 1.0 ($P_X = P_{CO}$) is achievable, if the buckle can be arrested in the upstream section of the pipeline. Most common types of buckle arrestors are: (1) slip-on arrestors (Lee and Kyriakides, 2004), where the arrestor is slipped over the pipe, (2) integral arrestors (Kyriakides et al., 1998), where the arrestor is welded to the pipe, (3) spiral arrestors (Kyriakides and Babcock, 1982), in which the arrestor is wound onto the pipe, and (4) clamped arrestor (Kyriakides, 2002). Amongst those, slip-on arrestors and integral arrestors are more prevalent. Slip-on arrestors are normally preferred to integral arrestors, since no welding is required. However, previous research has shown that their efficiency is normally lower than the integral arrestors (Kyriakides, 2002). The integral arrestor is a thick ring that is
welded onto the pipeline. The weld should be robust enough to resist large deformations during the buckle propagation (Kyriakides et al., 1998; Li, 2016). Therefore, the costs of installation of integral arrestors are significantly higher than other options.

Due to its excellent properties, such as high specific strength and stiffness, performance to weight ratio, thermal stability and corrosion resistance (Wonderly et al., 2005; Keller et al., 2013; Goertzen and Mullins, 2007; Sen and Mullins, 2007), carbon fibre reinforced polymer (CFRP) wraps are used to repair corroded and mechanically damaged offshore pipelines (Shamsuddoha et al., 2013; Duell et al., 2008; Seica and Packer, 2007). Moreover, experimental and numerical investigations have proved that application of CFRP in pipeline repair improves the capacity of the damaged pipeline in carrying bending, compression, tension and torsional loads, in both quasi-static and cyclic loading scenarios (Lukács et al., 2010; Lukács et al., 2011).

The common buckle arrestors may hinder the pipe laying operation. For instance, in reel-lay method the pre-installed devices (such as buckle arrestors) interfere with the reeling and unreeling process (Smith and Clough, 2010). Moreover, current buckle arrestors cannot be used in the inner components of pipe-in-pipe systems and pipeline bundles (Karampour et al., 2017; Alrsai et al., 2018; Olso and Kyriakides, 2003; Alrsai and Karampour, 2016). The current study proposes a CFRP buckle arrestor and investigates its feasibility, efficiency and appropriateness in offshore pipelines. The current work complements a previous study by the current authors (Karampour et al., 2019) which proved the feasibility of the CFRP buckle arrestor concept. To do so, experiments are conducted on stainless steel pipelines with diameter-to-thickness ratio, \( D/t \), of 28 and 40, without arrestors (bare samples) and with CFRP arrestors, in a hyperbaric chamber. A parametric study is conducted to find the optimum thickness \( (h) \), length \( (L) \), and orientation \( (\theta^\circ) \) of the CFRP arrestor. Moreover, different CFRP arrestor manufacturing methods are tested. Using the experimental results, efficiency of CFRP arrestor are calculated and compared against those of existing buckle arrestors.
6.2 Materials and Methods

6.2.1 Material properties

6.2.1.1 The pipeline

The hyperbaric chamber tests were conducted on small-scale seamless, SS-304 pipelines with $D/t$ ratios of 28 and 40. Mechanical properties of each pipe used in the experiments were measured using uni-axial tensile tests conducted on coupon samples cut from each batch (6 meters long) of the pipes, and along their longitudinal direction according to the recommendations of AS 1391-2007 (Australian Standard, 2007). The hyperbaric chamber tests showed different buckle propagation speeds in the pipes with different $D/t$ ratio. Therefore, the coupon tests were conducted at two different strain-rates; slow rate (such that rupture in the sample occurs in 4-5 minutes) and fast rate (8 times the slow rate). The tests were paused at the vicinity of the yield and ultimate points, to capture the upper and lower yield and ultimate stresses, respectively. The forces were calculated directly from the INSTRON 5900 universal testing system, and the strains were measured using a clip-on extensometer with a gauge length of 50.8 mm, attached to the middle section of the coupon sample. The ends of the coupon samples were flattened before being clamped into the machine. In order to account for the curved coupon surface, the correction factor recommended in AS 1391-2007 (Australian Standard, 2007) was used. Detail of the coupon samples and strain rates are given in Table 6.1.
Table 6.1. Coupon of the pipeline and results.

<table>
<thead>
<tr>
<th>Coupon ID</th>
<th>Strain rate (s⁻¹)×10⁻³</th>
<th>D (mm)</th>
<th>t (mm)</th>
<th>E (GPa)</th>
<th>( \sigma_{0.2%} ) (MPa)</th>
<th>( \sigma'_{y} ) (MPa)</th>
<th>( \sigma''_{y} ) (MPa)</th>
<th>( \sigma'_u ) (MPa)</th>
<th>( \sigma''_u ) (MPa)</th>
<th>Elongation at Rupture (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>28A1</td>
<td>2.667</td>
<td>25.4</td>
<td>0.9</td>
<td>207.6</td>
<td>348.3</td>
<td>437.0</td>
<td>444.5</td>
<td>679.7</td>
<td>692.0</td>
<td>64.2%</td>
</tr>
<tr>
<td>28A2</td>
<td>2.667</td>
<td>25.4</td>
<td>0.9</td>
<td>209.2</td>
<td>360.0</td>
<td>426.0</td>
<td>430.7</td>
<td>681.7</td>
<td>695.3</td>
<td>64.2%</td>
</tr>
<tr>
<td>28B1</td>
<td>2.667</td>
<td>25.4</td>
<td>0.9</td>
<td>199.9</td>
<td>357.3</td>
<td>402.5</td>
<td>407.4</td>
<td>655.2</td>
<td>672.2</td>
<td>64.2%</td>
</tr>
<tr>
<td>28B2</td>
<td>2.667</td>
<td>25.4</td>
<td>0.9</td>
<td>202.2</td>
<td>346.4</td>
<td>394.1</td>
<td>396.6</td>
<td>639.0</td>
<td>653.5</td>
<td>65.0%</td>
</tr>
<tr>
<td>28C1</td>
<td>0.333</td>
<td>25.4</td>
<td>0.9</td>
<td>208.9</td>
<td>337.4</td>
<td>382.3</td>
<td>386.2</td>
<td>642.9</td>
<td>655.0</td>
<td>65.6%</td>
</tr>
<tr>
<td>28C2</td>
<td>0.333</td>
<td>25.4</td>
<td>0.9</td>
<td>206.4</td>
<td>332.8</td>
<td>373.0</td>
<td>380.0</td>
<td>629.8</td>
<td>643.0</td>
<td>67.5%</td>
</tr>
<tr>
<td>28D1</td>
<td>0.333</td>
<td>25.4</td>
<td>0.9</td>
<td>196.2</td>
<td>341.2</td>
<td>387.7</td>
<td>388.0</td>
<td>662.2</td>
<td>672.2</td>
<td>66.5%</td>
</tr>
<tr>
<td>28D2</td>
<td>0.333</td>
<td>25.4</td>
<td>0.9</td>
<td>198.3</td>
<td>338.4</td>
<td>386.9</td>
<td>391.6</td>
<td>658.4</td>
<td>678.3</td>
<td>67.2%</td>
</tr>
<tr>
<td>28E1</td>
<td>0.333</td>
<td>25.4</td>
<td>0.9</td>
<td>209.6</td>
<td>340.1</td>
<td>385.3</td>
<td>389.0</td>
<td>654.2</td>
<td>666.9</td>
<td>68.1%</td>
</tr>
<tr>
<td>28E2</td>
<td>2.667</td>
<td>25.4</td>
<td>0.9</td>
<td>203.5</td>
<td>361.2</td>
<td>441.7</td>
<td>449.5</td>
<td>690.7</td>
<td>701.4</td>
<td>65.2%</td>
</tr>
<tr>
<td>40A1</td>
<td>0.733</td>
<td>63.5</td>
<td>1.6</td>
<td>201.0</td>
<td>311.2</td>
<td>337.7</td>
<td>340.0</td>
<td>674.2</td>
<td>685.3</td>
<td>88.3%</td>
</tr>
<tr>
<td>40A2</td>
<td>0.733</td>
<td>63.5</td>
<td>1.6</td>
<td>202.7</td>
<td>317.6</td>
<td>341.9</td>
<td>347.6</td>
<td>677.5</td>
<td>690.0</td>
<td>87.0%</td>
</tr>
<tr>
<td>40B1</td>
<td>0.733</td>
<td>63.5</td>
<td>1.6</td>
<td>198.9</td>
<td>316.5</td>
<td>351.0</td>
<td>354.3</td>
<td>687.0</td>
<td>697.6</td>
<td>88.5%</td>
</tr>
<tr>
<td>40B2</td>
<td>0.733</td>
<td>63.5</td>
<td>1.6</td>
<td>203.0</td>
<td>313.2</td>
<td>350.4</td>
<td>352.5</td>
<td>685.7</td>
<td>699.5</td>
<td>85.0%</td>
</tr>
<tr>
<td>40C1</td>
<td>0.093</td>
<td>63.5</td>
<td>1.6</td>
<td>204.6</td>
<td>301.8</td>
<td>322.3</td>
<td>326.7</td>
<td>684.4</td>
<td>694.3</td>
<td>91.7%</td>
</tr>
<tr>
<td>40C2</td>
<td>0.093</td>
<td>63.5</td>
<td>1.6</td>
<td>199.7</td>
<td>309.6</td>
<td>331.6</td>
<td>337.2</td>
<td>673.0</td>
<td>682.0</td>
<td>88.7%</td>
</tr>
</tbody>
</table>

6.2.1.2 CFRP arrestors

Coupon samples according to ASTM d3039/D3039M-08 (ASTM, 2018) were cut from the CFRP sheets. All the coupons were 25.0 mm wide by 250 mm long unidirectional beam laminates. The coupon samples had thickness of 0.80-0.92 mm in the vacuum bagging samples (VB), 1.25-1.31 mm in wet lay-up samples (WL), and 0.86-0.95 mm in prepreg samples (PP). The samples were tested in the longitudinal (0°) and transverse (90°) directions. Average results of the coupon tests for different manufacturing methods and corresponding volume fractions \( (V_f) \) are given in Table 6.2.
### Table 6.2: Coupon of the CFRP and average results.

<table>
<thead>
<tr>
<th>Manufacturing Technique</th>
<th>Fiber Orientation</th>
<th>No. of Samples</th>
<th>$E$ (GPa)</th>
<th>$\sigma_u$ (MPa)</th>
<th>$V_f$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prepreg (PP)</td>
<td>Longitudinal (0°)</td>
<td>3</td>
<td>132.7</td>
<td>1967.47</td>
<td>0.547</td>
</tr>
<tr>
<td></td>
<td>Transverse (90°)</td>
<td>1</td>
<td>7.41</td>
<td>13.58</td>
<td></td>
</tr>
<tr>
<td>Vacuum Bagging (VB)</td>
<td>Longitudinal (0°)</td>
<td>5</td>
<td>120.9</td>
<td>1854.97</td>
<td>0.513</td>
</tr>
<tr>
<td></td>
<td>Transverse (90°)</td>
<td>3</td>
<td>7.62</td>
<td>25.27</td>
<td></td>
</tr>
<tr>
<td>Wet Lay-up (WL)</td>
<td>Longitudinal (0°)</td>
<td>5</td>
<td>85.79</td>
<td>1310.44</td>
<td>0.350</td>
</tr>
<tr>
<td></td>
<td>Transverse (90°)</td>
<td>3</td>
<td>5.47</td>
<td>27.70</td>
<td></td>
</tr>
</tbody>
</table>

### 6.2.2 Manufacturing

#### 6.2.2.1 The methods

The CFRP buckle arrestors were fabricated and installed using three manufacturing techniques: (1) the vacuum bagging (VB), (2) the wet lay-up (WL), and (3) the prepreg (PP).

In the VB technique, a weight ratio of 50% carbon fibre (UD, 200 g/m²) to 50% resin was used to calculate the required amount of epoxy resin, taking into account a resin loss factor of 25%. The epoxy resin was made up of resin (R 180) and hardener (H 180) with a ratio of 5:1 of resin to hardener. The fibre layers were initially cut into the required length ($L$) and orientation ($\theta$). A metal roller was then used for consolidation, and the resin was spread evenly and gradually (to avoid any voids). Then, the wet fibre layer was tightly wound around the pipe at the required location. The procedure was repeated until the required thickness ($h$) was achieved. Then, the vacuum bag was installed, and the CFRP arrestor was consolidated under 1 bar vacuum pressure and cured for 24 hours at room temperature. Figure 6.1 shows a summary of the manufacturing process for the vacuum bagging technique.

In the WL technique, the same manufacturing process as in the VB was adopted. However, the samples were left to cure in the room temperature for 24 hours (Figure 6.2).
In the PP method, the CFRP buckle arrestors were manufactured from the VTM62 HS200 (unidirectional prepreg, 200 g/m² high strength) epoxy resin products obtained from Advanced Composites group (ACG, 2018). The VTM62 products were chosen due to their flexible curing capability in single prepreg system at low temperature and their free-standing posture capability. The prepreg CFRP arrestors were manufactured by stacking the self-sticking VTM62 layers to achieve the required thickness ($h$) and in the desired orientation ($\theta$). Each layer of the prepreg had a thickness slightly over 0.2 mm. A shrink tube was wrapped over the arrestor and the samples were oven cured for 1 hour at a temperature of 120°C. The PP manufacturing method procedure is shown in Figure 6.3.

In order to study the effect of the bond between the CFRP arrestor and the pipeline, two different surface preparation methods were implemented; (a) fine sanding (F) using a 180 grit sandpaper, and (b) coarse sanding (C) with a 40 grit sandpaper.
Figure 6-2: Curing process in wet lay-up (WL) method.

1- Sample preparations; includes Surface sanding and Sample marking.

2- Cut CFRP layers in required dimension and orientation.

3- Wrap the required number of CFRP layers.

6- Ready for test.

5- Insert the sample into the oven for curing (2 hours @ 120°C).

4- Slide the shrink tube over the CFRP to provide consolidation pressure.

Figure 6-3: The manufacturing process in the prepreg (PP) method.
6.2.3 Samples and labelling

6.2.3.1 General

The tested pipelines are selected based on the $D/t$ (slenderness ratio). The thicker pipeline has $D/t \approx 28$ ($D = 25.4$ mm and $t = 0.9$ mm) and the thinner pipeline has $D/t \approx 40$ ($D = 63.5$ mm and $t = 1.6$ mm). From each $D/t$, three batches (A, B and C) in length of 6 meters were purchased. From each batch, two or three pipelines were cut and were used to manufacture bare samples (without arrestors) or samples with CFRP arrestors. The samples are identified by their $D/t$ (28 or 40), pipe batch label (A to E), coupon test ID (1 or 2), CFRP manufacturing method (VB, WL or PP) and surface preparation process (F or C). For example, sample 28A2WLF represents a pipeline with $D/t \approx 28$ ($D = 25.4$ mm and $t = 0.9$ mm), cut from batch A, coupon test 2, with CFRP manufactured using the wet layup method (WL), and surface prepared via fine sanding (F). The bare samples are identified with “BS”. So, 40C1BS refers to a pipeline with $D/t \approx 40$ ($D = 63.5$ mm and $t = 1.6$ mm), cut from batch C, coupon test number 1, without any arrestors.

6.2.3.2 Parametric study

In the parametric study, 6 samples were prepared from the thick pipe ($D/t \approx 28$), all with CFRP arrestors manufactured using the PP method and fine sanding surface preparation. Different batches of steel (batch D and E) were used to manufacture the pipelines in the parametric study. In this case, a 28D1PPF1 represents a thick pipe ($D/t \approx 28$), from batch D, coupon test 1, manufactured with PP method and fine sanding, in CFRP configuration 1 (see Figure 6.5).

6.2.4 Hyperbaric chamber tests

6.2.4.1 Test setup

The samples were tested inside a hyperbaric chamber with an inner diameter of 176 mm and clear length of 4 m, with a nominal capacity of 30 MPa (water depth of 3,000 m) shown in Figure 6.4. The pipeline samples were sealed at both ends by welding on a thick disc, placed inside the hyperbaric chamber and then filled with water. One end of the pipe was connected to an outlet nozzle through the chamber’s wall and vented to the atmosphere. The change in
the volume of the pipe during the test was monitored using a digital scale to measure the weight of the water exiting the pipe via the outlet in the chamber’s wall as shown in Figure 6.4. A buckle monitoring system comprised of DVU unit (DVU400-17) (Figure 6.4(b)), and a high-pressure camera (Titan-3000) and LED light (C-DragonHP) were placed inside the chamber, to monitor the collapse and its propagation. In order to maintain the position and angle of the camera and LED light during the test, a custom-made acrylic support was manufactured using a laser cutter. The chamber was then sealed and filled with water. The pressure inside the chamber was increased in a control-volume method using a water-pump. The hyperbaric pressure time-history of the chamber was monitored using a pressure gauge shown in Figure 6.4.

In order to control the location of the collapse in the pipeline, an initial imperfection in the shape of a dent was induced to one end of the pipe. To impose the dent, the pipeline was clamped in the jaw of a universal testing machine, and a rigid semi-circular rod of the same diameter was gradually pressed against it. The imperfection was quantified by measuring the maximum and minimum diameters \(D_{\text{max}}\) and \(D_{\text{min}}\) of the dented pipe and represented in terms of ovality \(\Delta_0\)

\[
\Delta_0 = \frac{D_{\text{max}} - D_{\text{min}}}{D_{\text{max}} + D_{\text{min}}}
\]  

(6.2)
6.2.4.2 Test plans

In total 18 hyperbaric chamber tests were conducted comprising three testing series; (1) Bare samples were tested to calculate the buckling behaviour and the propagation pressure ($P_P$) of the pipeline without arrestors. The results were used as benchmarks for the tests with CFRP arrestors. Two samples from each $D/t$ with length of 2.5 meters each were tested, and their properties and obtained results are given in Table 6.3.
Table 6.3. Bare samples and the hyperbaric chamber results (Test plan 1, Length of the tested pipeline is 2.5 meters).

<table>
<thead>
<tr>
<th>Exp. ID</th>
<th>$D$ (mm)</th>
<th>$t$ (mm)</th>
<th>$D/t$</th>
<th>$P_p$ (MPa)</th>
<th>$P_I$ (MPa)</th>
<th>$P_{ico}$ (Eq.6.3) (MPa)</th>
<th>$P_P$ (Eq.6.7) (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>28C1BS</td>
<td>25.4</td>
<td>0.9</td>
<td>28.22</td>
<td>2.60</td>
<td>8.60</td>
<td>9.91</td>
<td>2.63</td>
</tr>
<tr>
<td>28C2BS</td>
<td>25.4</td>
<td>0.9</td>
<td>28.22</td>
<td>2.53</td>
<td>7.60</td>
<td>9.11</td>
<td>2.52</td>
</tr>
<tr>
<td>40C1BS</td>
<td>63.5</td>
<td>1.6</td>
<td>39.69</td>
<td>1.00</td>
<td>4.55</td>
<td>4.75</td>
<td>1.05</td>
</tr>
<tr>
<td>40C2BS</td>
<td>63.5</td>
<td>1.6</td>
<td>39.69</td>
<td>1.00</td>
<td>4.60</td>
<td>4.77</td>
<td>1.08</td>
</tr>
</tbody>
</table>

(2) A parametric study was conducted to find the optimum thickness ($h$), length ($L$), and orientation ($\theta^\circ$) of the CFRP arrestor. A total of six tests were conducted on the pipeline with $D/t \approx 28$, using PP method with fine sanding. The CFRP buckle arrestor configurations are shown in Figure 6.5 and their parametric properties are represented in Table 6.4. In configurations 1 and 2 (Figure 6.5 from top), effect of the thickness of CFRP arrestors were studied. In configurations 3 and 4, two CFRP arrestors were oriented in $\theta = -35^\circ/35^\circ$ and $\theta = -55^\circ/55^\circ$ biaxial (with respect to the longitudinal pipeline axis), respectively, in order to investigate fibre orientation effect. The effect of the length of the CFRP arrestor ($L$) was studied by comparing configurations 1-2 and 5-6. All pipe samples had a length of 1.6 meters each.

(3) A total of eight hyperbaric chamber tests were conducted on samples with CFRP arrestors, to investigate the effect of the manufacturing technique and diameter to thickness ratio. Properties of the samples are outlined in Table 6.5. The pipe samples had lengths of 2.5 meters each.
Figure 6-5: The Stainless-steel tube samples \((D = 24.5\text{mm} \text{ and } t = 0.9\text{mm})\) with different CFRP arrestsor configurations (Test plan 2).
Table 6.4. Parametric study of CFRP and hyperbaric chamber results (Test plan 2, Length of the tested pipeline is 1.6 meters).

<table>
<thead>
<tr>
<th>Exp. ID</th>
<th>CFRP Arrestor ID</th>
<th>$h/t$</th>
<th>$L/D$</th>
<th>Fibre Orientation (FO)</th>
<th>$P_{xi}$ (MPa)</th>
<th>$\left(\frac{P_{xi}}{P_p}\right)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>28D1PPF1</td>
<td>1_A1</td>
<td>2.0</td>
<td>2.0</td>
<td>90°</td>
<td>5.3</td>
<td>2.08</td>
</tr>
<tr>
<td></td>
<td>1_A2</td>
<td>2.0</td>
<td>2.0</td>
<td>90°</td>
<td>7.0</td>
<td>2.74</td>
</tr>
<tr>
<td></td>
<td>1_A3</td>
<td>2.0</td>
<td>2.0</td>
<td>90°</td>
<td>6.5</td>
<td>2.55</td>
</tr>
<tr>
<td>28D2PPF2</td>
<td>2_A1</td>
<td>1.0</td>
<td>2.0</td>
<td>90°</td>
<td>6.2</td>
<td>2.43</td>
</tr>
<tr>
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<td>2_A2</td>
<td>1.0</td>
<td>2.0</td>
<td>90°</td>
<td>7.0</td>
<td>2.74</td>
</tr>
<tr>
<td>28D2PPF3</td>
<td>3_A1</td>
<td>1.0</td>
<td>2.0</td>
<td>35°</td>
<td>3.3</td>
<td>1.29</td>
</tr>
<tr>
<td></td>
<td>3_A2</td>
<td>1.0</td>
<td>2.0</td>
<td>35°</td>
<td>2.5</td>
<td>1.00</td>
</tr>
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<td>28E1PPF4</td>
<td>4_A1</td>
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<td>2.0</td>
<td>55°</td>
<td>2.7</td>
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</tr>
<tr>
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<td>4_A2</td>
<td>1.0</td>
<td>2.0</td>
<td>55°</td>
<td>3.2</td>
<td>1.25</td>
</tr>
<tr>
<td>28E1PPF5</td>
<td>5_A1</td>
<td>1.0</td>
<td>1.0</td>
<td>90°</td>
<td>5.5</td>
<td>2.16</td>
</tr>
<tr>
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<td>5_A2</td>
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<td>1.0</td>
<td>90°</td>
<td>4.5</td>
<td>1.76</td>
</tr>
<tr>
<td>28E2PPF6</td>
<td>6_A1</td>
<td>2.0</td>
<td>1.0</td>
<td>90°</td>
<td>8.0</td>
<td>3.14</td>
</tr>
<tr>
<td></td>
<td>6_A2</td>
<td>2.0</td>
<td>1.0</td>
<td>90°</td>
<td>7.65</td>
<td>3.00</td>
</tr>
</tbody>
</table>
Table 6.5. Pipelines with CFRP arrestors and results (Test plan 3, Length of the tested pipeline is 2.5 meters).

<table>
<thead>
<tr>
<th>Exp. ID</th>
<th>$D$ (mm)</th>
<th>$t$ (mm)</th>
<th>$D/t$</th>
<th>$\Delta_0$ (%)</th>
<th>$h/t$</th>
<th>$L/D$</th>
<th>$P_P$ (MPa)</th>
<th>$P_X$ (MPa)</th>
<th>$\frac{P_X}{P_P}$</th>
<th>$\frac{P_X}{P_L}$</th>
<th>$\Delta t_d$ (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>28A1WLF</td>
<td>25.4</td>
<td>0.9</td>
<td>28.22</td>
<td>0.994</td>
<td>2.0</td>
<td>2.0</td>
<td>2.70</td>
<td>9.40</td>
<td>3.48</td>
<td>1.07</td>
<td>56</td>
</tr>
<tr>
<td>28A2WLC</td>
<td>25.4</td>
<td>0.9</td>
<td>28.22</td>
<td>0.994</td>
<td>2.0</td>
<td>2.0</td>
<td>2.80</td>
<td>9.10</td>
<td>3.25</td>
<td>1.18</td>
<td>51</td>
</tr>
<tr>
<td>28B1VBF</td>
<td>25.4</td>
<td>0.9</td>
<td>28.22</td>
<td>0.994</td>
<td>2.0</td>
<td>2.0</td>
<td>2.60</td>
<td>8.85</td>
<td>3.40</td>
<td>1.00</td>
<td>50</td>
</tr>
<tr>
<td>28B2VBC</td>
<td>25.4</td>
<td>0.9</td>
<td>28.22</td>
<td>0.994</td>
<td>2.0</td>
<td>2.0</td>
<td>2.80</td>
<td>7.90</td>
<td>2.82</td>
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6.3 Results and Discussion

6.3.1 Material properties

The yield stresses ($\sigma_y$), modulus of elasticity ($E$), ultimate stresses ($\sigma_u$), and elongation at rupture of the SS-304 stainless steel coupon samples are given in Table 6.1. The stress-strain curves at two different strain rates are shown in Figure 6.6. The elongation at rupture is calculated by dividing the gauge length of the coupon sample before the test and at the rupture (drop in the load shown in Figure 6.6) and is represented in percentage in Table 6.1. The high elongations at rupture show that the stainless steel has a large ductility. The thinner pipe ($D/t \approx 40$) is more ductile than the thick pipe ($D/t \approx 28$). The average differences between the upper and lower stress (shown with superscripts $U$ and $L$ in Table 6.1, respectively) at the yield and at the ultimate stress are 1.2% and 1.9%, respectively.

The moduli of elasticity of the samples are independent of the strain rate. However, the characteristic stresses upsurge with an increase in the strain rate (increase in the speed of the tensile test). Particularly, the lower yield stress is more sensitive to the strain rate compared to
other characteristic stress values. Elongations at rupture (ductility) of the samples seem to decrease with an increase in the strain rate.

The modulus of elasticity \((E)\) and ultimate stress \((\sigma_u)\) and corresponding volume fraction \((V_f)\) of the CFRP coupon samples are given in Table 6.2 for three manufacturing techniques and in two fibre orientations. The average moduli of elasticity of the CFRP along the direction of the fibre \((0^\circ)\) from the PP manufacturing method is 10% and 54% greater than those of the VB and WL methods, respectively. The ultimate stresses of the PP along the fibre are 6% and 50% larger than VB and WL, respectively. The moduli of elasticity perpendicular to the direction of the fibre are between 5-6% of those parallel to the fibre direction from different manufacturing methods.

![Stress-strain curves](image)

**Figure 6-6:** Typical coupon results, showing stress-strain curves at strain rates of \(0.333 \times 10^{-3}\) \((s^{-1})\) and \(2.667 \times 10^{-3}\) \((s^{-1})\) for samples 28C1 and 28A2, respectively (refer to Table 6.1).

### 6.3.2 Hyperbaric chamber test results of the bare samples (Test plan 1)

The pressure response of the bare samples is plotted against the normalised change in the volume (volume of the water discharged from the pipe divided by the initial volume of the
(1) The initiation pressure ($P_I$) at which the collapse is initiated in the pipeline and is sensitive to the imperfection $\Delta_0$ (shown on the figure), and (2) the propagation pressure ($P_P$) which is much smaller than the initiation pressure and is not imperfection sensitive. Previous researches (Albermani et al., 2011; Chater and Hutchinson, 1984; Kyriakieds et al., 1984) have shown that the initiation pressure is a combined elastic-plastic failure. The in-elastic expression ($P_{ico}$) suggested by Timoshenko (Timoshenko and Gere, 1961) can be used to predict the collapse pressure of a pipe under uniform external pressure, based on its yield pressure ($P_o$), elastic critical buckling pressure ($P_{cr}$), and imperfection parameter ($\phi$):

$$P_{ico} = \frac{1}{2} \{(P_o + \phi P_{cr}) - [(P_o + \phi P_{cr})^2 - 4 P_o P_{cr}]^{0.5}\} \quad (6.3)$$

$$P_0 = \frac{2t\sigma_y}{D} \quad (6.4)$$

$$P_{cr} = \frac{2E}{(1-v^2)} \left(\frac{t}{D_o}\right)^3 \quad (6.5)$$

$$\phi = 1 + 3\Delta_0 \frac{D}{t} \quad (6.6)$$

The propagation pressure ($P_P$) can be predicted by the empirical expression suggested by Kyriakides and Lee (Kyriakides and Lee, 2005) for stainless steel (SS304)

$$P_P = 20.69\sigma_y \left(\frac{t}{D}\right)^{2.362} \quad (6.7)$$

As can be seen in Figure 6.7, the propagation pressures are very similar between different tests of each $D/t$. Using Eqs. (6.3) and (6.7), by substituting material properties from Table 6.1, and imperfections $\Delta_0$, shown in Figure 6.7, the collapse pressures and propagation pressures obtained from the experiments and the suggested equations are compared in Table 6.3. It can be seen from Table 6.3 that the propagation pressures predicted from Eq. (6.7) are
similar to those obtained from the experiments with a maximum difference of 8%. However, the collapse pressures from Eq. (6.3) are 18% and 4% higher than those from the experiments for $D/t$ of 28 and 40, respectively. Same conclusion was reported in (Kyriakides and Corona, 2007) because, the Timoshenko’s formula (Eq. (6.3)) is known to give more accurate results in thin pipes (large $D/t$).

Images of the collapsed pipelines after the test, and during the test (using the buckle monitoring system) are also shown in Figure 6.7. The failed pipelines removed from the pressure chamber showed a flat mode, whereas the pictures from inside the chamber and during the test, depict a dog-bone buckle mode. The flat buckle mode happens at higher pressures and when the buckle reaches the vicinity of the bulkhead. This is observed in the upsurge of the pressure towards the end of the pressure-volume responses in Figure 6.7.
Figure 6-7: Pressure-volume response from the hyperbaric chamber tests of the bare samples
(a) $D/t \approx 28$, showing a snapshot from high-pressure camera of the buckled tube, (b) $D/t \approx 40$,
showing a snapshot from high-pressure camera showing of buckled tube. Failed samples are
removed from the chamber also shown.

6.3.3 Parametric study on geometric parameters of CFRP buckle arrestors
(Test plan 2)

A parametric study on effect of CFRP arrestor thickness ($h$), Length ($L$), and fibre orientation
($\theta^\circ$) using the PP manufacturing method and fine sanding surface finish, on the buckle
propagation response of the pipeline with $D/t = 28.22$ is given here. The test configurations
are shown in Figure 6.5 and represented in Table 6.4. Pressure responses of the samples with
CFRP arrestors (solid line) and corresponding bare samples (dotted line) are plotted in
Figures. 6.8-6.10, in groups of 2 in each figure. The collapse starts at the location of the
imposed imperfection (a distance $6D$ away from the bulkhead shown in Figure 6.5). The
pressure then drops, and the buckle propagates towards the CFRP arrestors ($A_i$). The
maximum pressure measured at the vicinity of each CFRP arrestor (the cross-over pressure) is
denoted by $P_{xi}$. The increase in the pressure capacity due to the presence of the CFRP arrestor
is shown as the ratio of the cross-over pressure ($P_{xi}$) divided by the propagation pressure of
the bare sample \( (P_P) \). These ratios are represented in Table 6.4 for all tested configurations. As shown in Figure 6.8(a), a more congested positioning of the buckle arrestors (arrestor spacing smaller than 20\( D \)), does not increase the cross-over pressure. This was confirmed in the experimental tests of steel slip-on buckle arrestors (Lee and Kyriakides, 2004), which suggest that a spacing of 20\( D \) between the buckle arrestors is sufficient. Comparison between results of Figures. 6.8(a) and 6.8(b) shows that, with \( \theta = 90^\circ \) and \( h/t = 1.0 \), the cross-over ratio \( P_{Xo}/P_P \) is always greater than 2.0, if \( L/D \) of 2.0 is maintained.
Maintaining $h/t = 1.0$ and $L/D = 2.0$, the CFRP arrestors were stacked in $\pm 35^\circ$ and $\pm 55^\circ$ biaxial orientation with respect to the longitudinal pipeline axis. The $35^\circ$ angle mimics the steel armour orientation used in flexible subsea pipelines and risers (Sævik and Li, 2013). As shown in Figure 6.9, fractures were observed in all CFRP arrestors. No significant increase in the cross-over ratios compared to the propagation pressures of the bare samples was observed. This suggests that the optimum fibre orientation is ($\theta = 90^\circ$), which provides the maximum strength in the hoop direction.
Figure 6-9: Pressure responses of; (a) sample 28D2PPF3, and (b) sample 28E1PPF4.
Effect of \( h/t \), maintaining \( L/D = 1.0 \) and \( \theta = 90^\circ \), on the cross-over pressure is shown in Figure 6.10. A significant increase in \( P_{xi} \) is observed with \( h/t = 2.0 \), compared to \( h/t = 1.0 \). The close-up view of the CFRP arrestors shows that the largest cross-over pressure corresponds to the U-shape buckle mode observed in arrestors \( A_1 \) and \( A_2 \) in Figure 6.10(b). The same conclusion was reported in the confined buckle propagation modes investigated by Stepehn et al. (Stepehn et al., 2016) in single pipelines, Lee and Kyriakides (Lee and Kyriakides, 2004) in slip-on buckle arrestors, and Karampour et al. (Karampour et al., 2017) in pipe-in-pipe systems.

![Pressure graph with graphs showing CFRP arrestors](image-url)
Time-history of pressure responses of all samples from the parametric study are shown in Figure 6.11. The elapsed time from the initiation of the collapse, until the buckle reaches the arrestor $A_i$ is marked on the curves. The time gap ($\Delta t_d$) between the onset of the buckle initiation ($t = 0$) and the time at which the buckle crosses the respective arrestor, is denoted as the delay in buckle propagation. The largest delay was observed in the sample with $\theta = 90^\circ$ and $h/t = 2.0$ (28E2PPF6, Figure 6.10(b)). It took 31 seconds for the buckle to reach arrestor A1 and 51 seconds to reach arrestor A2. The second best in terms of delaying the buckle was the sample with $\theta = 90^\circ$ and $L/D = 2.0$ (28D2PPF2, Figure 6.8(b)), in which the buckle reached arrestor A1 in 11 seconds and arrestor A2 in 33 seconds. In the bare sample (dashed line in Figure 6.11) the buckle reached the opposite bulkhead in just 15 seconds.

A video (Video 6.1) of the buckle propagation of sample 28E2PPF6 inside the hyperbaric chamber is provided in the online version of the article. The buckle is initiated at $t = 5s$ at the location of the imposed dent, in the far end of the chamber. The buckle propagates to A1 and
is arrested until \( t = 36 \text{s} \). The buckle then crosses-over A1 and reaches A2 in about 0.24 seconds (equivalent to a speed of propagation equal to 0.22 \( m/s \)). The buckle is delayed between A1 and A2 (a clear length of 20\( D \)) for 20 seconds, at \( P_X/P_P = 3 \).

![Figure 6-11: Pressure-time history of all samples from the parametric study (Test plan 2).](image)

**6.3.4 Effect of the manufacturing method (Test plan 3)**

The parametric study (in test plan 2) revealed the superior performance of CFRP arrestors (manufactured with PP method) with \( h/t = 2 \) and \( \theta = 90^\circ \). Therefore, in the manufacturing study (test plan 3), those parameters were adopted. Conservatively, a \( L/D=2 \) was adopted to ensure maximum cross-over pressure capacities are obtained. In test plan 3, only one single CFRP arrestor using VB and WL methods was installed at mid-length of the pipeline, to make certain that the capacity of the arrestor was not influenced by the end caps or adjacent arrestors. A schematic pressure vs. change of volume result from the hyperbaric chamber, showing the initiation, propagation and cross-over pressures is depicted in Figure 6.12. The pressure ratios are presented in Table 6.5. Result of the pipeline with the optimum CFRP arrestor configuration manufactured using the PP method (from test plan 2) is also included.
Compared to $P_X/P_P$, the $P_X/P_I$ ratio is a less reliable factor for comparison of performance of CFRPs. Because, although similar initial imperfections of almost identical magnitude were introduced to all pipes (Table 6.5), the initiation pressures depict some differences. The reason is associated with the anisotropy in yielding introduced during the manufacturing process of seamless tubes, which in turn can affect the initiation pressure (Kyriakides and Yeh, 1988).

Hyperbaric chamber results showed that all tested pipelines deformed in a flatten buckle mode, the buckle penetrated through the CFRP arrestor in a U-shape buckle mode. No fractures were observed in the CFRP arrestors. In all samples, regardless of the manufacturing method and surface preparation, $P_X/P_P$ larger than 3 were observed at either $D/t$. However, the results in Table 6.5 clearly show that the cross-over pressure ($P_X$) is affected by the manufacturing technique and the surface preparation method. Despite lower moduli of elasticity of WL samples compared to others (Table 6.2), no significant change in $P_X/P_P$ from different manufacturing methods are observed. On the other hand, the fine sanding consistently enhances the cross-over ratio $P_X/P_P$, with an average of 20% increase compared to the coarse sanding.

Figure 6-12: Schematic of pressure versus change of volume inside the hyperbaric chamber.
Time-history of pressure responses for each $D/t$ is shown in Figure 6.13. The figure shows the elapsed time from the initiation of the collapse until the buckle reaches the CFRP arrestor. The buckle arrestor delay times ($\Delta t_d$) are given in Table 6.5. The largest $\Delta t_d$ corresponds to the WL manufacturing method with fine sanding. The VB method comes second in terms of delaying the buckle.

**Figure 6-13:** Hyperbaric chamber pressure-time histories of test plan 3: (a) $D/t = 28$, (b) $D/t = 40$. 

160
6.3.5 Efficiency of CFRP buckle arrestors (comparison with slip-on and integral arrestors)

Gong and Li (Gong and Li, 2017) performed a parametric study to measure the cross-over pressure \( P_x/P_P \) in integral buckle arrestor based on the geometric parameters, \( D/t \) and \( h/t \). They proposed the following empirical equation for pipelines with \( 15 < D/t < 35 \)

\[
\frac{P_x}{P_P} = 1 + 1.086 \left( \frac{t}{D} \right)^{0.46} \left( \frac{h}{t} \right)^{3.05}
\]  

(6.8)

By using Eq. (6.8) and the parameters in Table 6.5, the cross-over ratio \( P_x/P_P \) of 2.94 and 2.65 are calculated for pipes with \( D/t \approx 28 \) and 40, respectively. All of the experimental \( P_x/P_P \) in Table 6.5 for current CFRP arrestors are larger than those predicted by Eq. (6.8). Moreover, Gong and Li (Gong and Li, 2017) found that in integral arrestors, the cross-over ratio \( P_x/P_P \) decreases with the increase in \( D/t \). However, current results show that in CFRP arrestors the \( P_x/P_P \) is not affected by the \( D/t \) of the pipeline.

The arresting efficiency \( (\eta_{CFRP}) \) of the CFRP arrestors are listed in Table 6.6. The propagation pressures \( (P_P) \) and cross-over pressures \( (P_x) \), are taken from the experimental results. The collapse pressure \( (P_{ico}) \) of each sample is calculated using the Timoshenko’s formula (Eq. 6.3). Then, these pressures are used to find the CFRP arrestor efficiency \( (\eta_{CFRP}) \) in accordance with Eq. 6.1.

It can be seen from Table 6.6 that the pipeline with \( D/t \) of 28 yields higher efficiencies compared to the pipeline with \( D/t \) of 40, regardless of the manufacturing and surface preparation methods. Efficiencies almost equal to 1.0 are observed in the pipeline with \( D/t \) of 28 and using the WL method with fine sanding. Lowest efficiencies are observed in samples manufactured with the VB method and coarse sanding. Using mechanical properties obtained from coupon tests performed at faster strain rates, lower efficiencies are calculated. In average, differences between efficiencies at fast and slow rates for \( D/t \approx 28 \) and 40, are 4.7% and 1.2%, respectively.
Kyriakides and Lee (Lee and Kyriakides, 2004; Kyriakides, 2002) performed hyperbaric tests on 22 stainless steel pipelines with $D/t$ between 14 and 94 and slip-on buckle arrestors of various geometric configurations. Based on the experimental results, they proposed empirical expressions and upper/lower bounds for efficiency of slip-on arrestors at different $D/t$. Figure 6.14 shows efficiencies of the CFRP arrestors from the current study compared with the lower and upper bounds for slip-on arrestors from (Lee and Kyriakides, 2004). As shown in Figure 6.14, at $D/t \approx 28$ the slip-on arrestor efficiencies range between 0.68 to 0.81. However, except 2 test results, all measured CFRP efficiencies are larger than the predicted upper bound for

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slip-on arrestors. At $D/t \approx 40$, the efficiencies of CFRP arrestors drop and are closer to the predicted lower bound for slip-on arrestors. The CFRP efficiencies with coarse sanding in WL and VB methods are below the lower bound of slip-on arrestors at $D/t \approx 40$. It should be noted that to obtain the lower bound efficiency in slip-on arrestors in Figure 6.14, $h \geq 2.5t$ should be maintained (Lee and Kyriakides, 2004); however, the CFRP arrestors use $h=2t$.

![Graph showing efficiency comparison](image)

**Figure 6-14:** Comparison between efficiency of the current CFRP arrestors (dot points) and slip-on arrestors (Lee and Kyriakides, 2004) (lines).

Kyriakides and Park (Kyriakides et al., 1998; Park and Kyriakides, 1997) performed comprehensive experimental and numerical studies on the design of the integral buckle arrestors. They found a minimum required thickness ($h_{cm}$) for the integral buckle arrestors, at which the cross-over pressure ($P_X$) becomes equal to the collapse pressure of the pipe ($P_{CO}$),
i.e. efficiency of 1. By comparing the results from this chapter to the predictions in (Kyriakides et al., 1998; Park and Kyriakides, 1997), $h_{cm} = 3.5t$ and $4.6t$ are required (for $D/t \approx 28$ and 40, respectively) using integral arrestors to achieve similar efficiencies to CFRP arrestors with $h=2t$ as listed in Table 6.6.

### 6.4 Summary

Buckling responses of steel pipelines with CFRP buckle arrestors were experimentally investigated using hyperbaric chamber tests. The CFRP buckle arrestors were fabricated and installed on steel pipelines with $D/t \approx 28$ and $D/t \approx 40$ using three manufacturing techniques: (1) vacuum bagging (VB), (2) wet lay-up (WL), and (3) prepreg (PP) methods. Two different surface preparation methods were tested, (a) fine sanding, and (b) coarse sanding. Mechanical properties of the steel pipes and CFRP sheets were obtained from standard coupon tests at different strain rates. The hyperbaric chamber tests were performed in 3 test plans; (1) bare samples, (2) parametric study on effect of thickness ($h$), length ($L$), and orientation ($\theta^\circ$) of the CFRP arrestor, on its cross-over capacity (only PP method and with $D/t \approx 28$), and (3) the effect of the manufacturing technique and $D/t$ ratio.

The following significant outcomes were found from the current work:

- The CFRP arrestors build large cross-over pressures and significantly delay the buckle propagation in the pipeline.
- The optimum fibre orientation of the CFRP arrestor is $\theta = 90^\circ$, which provides the maximum stiffness in the hoop direction.
- The optimum manufacturing technique is the WL method with fine sanding, which provides the maximum buckle capacity and longest delay of the buckle propagation.
- The cross-over pressure ($P_X$) is affected by the manufacturing technique and the surface preparation method. However, by adopting $\theta = 90^\circ$, $h/t = 1.0$, and $L/D = 2.0$ a minimum cross-over ratio ($P_X/P_P$) equal to 3 can be achieved, regardless of the pipe $D/t$ ratio or the manufacturing method.
• The efficiency of a CFRP arrestor installed on a pipeline with \( D/t = 28 \) (used in deep waters) is higher than the predicted upper bound efficiency of a slip-on arrestor (Lee and Kyriakides, 2004) with similar parameters.

• To achieve similar efficiencies, the integral buckle arrestors need to be 1.75 and 2.3 times thicker than the proposed CFRP arrestors, for \( D/t = 28 \) and 40, respectively.

• Unlike slip-on and integral buckle arrestors, the normalised cross-over capacity \( (P_{xi}/P_P) \) of a CFRP arrestor is independent of the \( D/t \) of the pipeline.

Current results showed that the proposed CFRP arrestors can be an appropriate alternative to the existing slip-on and integral buckle arrestors. The superior results obtained from the WL method are promising, since this method can easily be used to manufacture CFRP arrestors in offshore pipeline with large diameter.
7.1 Conclusions

A detailed investigation into the propagation buckling of subsea pipe-in-pipe systems has been documented in this thesis. This research program considered the collapse mechanisms of the inner pipe of the pipe-in-pipe systems, also proposed a new buckle arrestor concept. The methodology for carrying out this research was based on comprehensive experimental and numerical studies of the propagation buckle phenomenon of subsea pipe-in-pipe systems.

The first stage of this thesis (Chapter 2) focused on conducting a comprehensive literature review of theoretical, experimental and numerical studies performed in the past to investigate the propagation buckling and collapse mechanisms of single pipelines and PIP systems. It was found that the previous studies were conducted mainly on single pipe systems, and only marginal research was undertaken on propagation buckling failure of pipe-in-pipe systems under external pressure. Further, previous studies have focused only on propagation buckling of PIP systems having carrier pipes with low $D/t$ values. Numerous analytical solutions have also been suggested to estimate the propagation buckle capacity of a single pipe. However, in the existing research, only one analytical solution was proposed for PIP propagation pressure ($P_{P2}$), which underestimated the experimental observed capacity.

This chapter concluded that collapse mechanisms of PIP systems have only been marginally addressed and there is no existing study on the collapse of the inner pipe of a PIP system under external pressure. Also, the existing buckle arrestors have shown certain limitations in practice.

Chapter 3 presented the details of the experimental studies, analytical and numerical analyses conducted on pipe-in-pipe systems consisting of carrier pipes with a diameter-to-thickness...
\((D_i/D_o)\) ratio in the range 26-40. The modified analytical solution was suggested and provided more accurate predictions of PIP propagation buckling pressure compared with previous analytical equations. Details of the test set-up, including hyperbaric chamber and ring squash tests and tests results, were described and compared with the numerical results. Three different buckling modes were discovered in the hyperbaric chamber tests.

Chapters 4 and 5 provided insight on buckling and collapse mechanisms and capacity of PIP systems and inner pipe of the PIP systems, respectively. Comprehensive parametric studies using validated FE model were conducted to predict the PIP propagation buckling pressure and collapse pressure of the inner pipe of the PIP system. The proposed expressions were shown to be in good agreement with experimental and numerical results.

In chapter 6, a new buckle arrestor concept was proposed to overcome the existing buckle arrestors limitations and to provide higher arresting efficiency. The concept consisted of wrapping CFRP layers around the pipeline in certain configurations, including thickness, length and orientation.

### 7.2 Significant Outcomes of This Research Program

The significant contributions of this thesis are summarised below:

- This research program has significantly improved the understanding and filled the knowledge gap of the propagation buckling failure of the subsea pipe-in-pipe systems. The effects of various parameters including geometric and material properties on the propagation buckling response of PIP system were investigated. Several comprehensive and extensive parametric studies were conducted.

- Modified analytical solution to predict \(P_{p2}\) was suggested in this research and accounts for the \(D_i/D_o\) ratio (Eq. 3.9) and was shown to provide more accurate predictions of PIP propagation buckling pressure compared with previous analytical equations.

- Novel RST and CRST protocols were proposed to provide estimates of propagation pressure in PIP systems in chapter 3 (section 3.4) and were shown to provide lower-
bound estimations of the propagation pressure of PIP system with respect to buckling modes observed in experimental works.

- The development of reliable and efficient FE models to replicate the experimental tests has been described in detail in this research. The modelling procedures employed herein have been appropriately validated and compared by an experimental investigation and can be used as a basis for future numerical investigation in the relevant field of research.

- Using the validated FE model, a comprehensive parametric study was conducted in chapter 4 on propagation buckling mechanisms of subsea pipe-in-pipe systems under external pressure. Two major buckle propagation modes in PIP system were observed and identified, also the separation between them was shown in this chapter. Empirical expressions for propagation buckling of PIPs with thin and moderately thin carrier pipes were proposed for each buckle modes. Furthermore, a separate empirical expression for propagation buckling of PIPs with thick and moderately thick carrier pipes was proposed. All proposed expressions were found to be different from previous expressions suggested for PIPs.

- The buckling mechanisms and capacity of the inner pipe of the PIP system, under external pressure, were investigated experimentally and numerically in chapter 5. The hyperbaric chamber test results showed that in PIPs with identical inner pipes and different outer pipes, the inner pipe may collapse at different pressures ($P_{ci}$). Also, the inner pipe is not significantly affected by the ovality in the cross-section of the inner pipe (see Figure 5.6). Results from this chapter showed that $P_{ci}$ drastically drops with the corresponding decrease in $D_i/D_o$ or with the corresponding increase $t_i/t_o$.

- A comprehensive parametric study was conducted to study the effect of geometric and material parameters of the PIP system on collapse pressure of the inner pipes including the wall thickness $t_i/t_o$, diameter $D_i/D_o$, diameter to thickness $D_i/t_i$, material yield stress $\sigma_Y/\sigma_{Yo}$, and tangent modulus $E'/E'_o$ ratios. Empirical expression for $P_{ci}$ was proposed and was shown to be in good agreement with hyperbaric chamber test results.
• Through combining all FE results in chapter 5, expressions for upper and lower bounds of the collapse pressure of the inner pipes were advised (Eqs. 5.14 and 5.15). Moreover, two distinctive modes of collapse in the inner pipe were identified and discussed. The separation between two collapse modes was found to occur at thickness ratio $t_i/t_o = 0.7$.

• A new buckle arrestor concept was proposed in chapter 6 for subsea pipelines using CFRP. The efficiency and feasibility of CFRP buckle arrestors were investigated using hyperbaric chamber tests in this research. It was found that CFRP buckle arrestors can reach higher levels of arresting efficiency compared to slip-on and integral buckle arrestors, therefore, efficiency of 1 is achievable with CFRP arrestor.

• The CFRP arrestors build large cross-over pressures and significantly delay the buckle propagation in the pipeline. Also, the optimum fibre orientation of the CFRP arrestor was found at $\theta = 90^\circ$, which provides the maximum stiffness in the hoop direction. Moreover, the optimum manufacturing technique was the wet-layup method with fine sanding.

Also, detailed conclusions and summary have been added at the end of each chapter, to show the significant outcomes from this research program.

7.3 Recommendations for Further Research

Based on the work presented in this thesis, the following research tasks can be recommended to further extend the knowledge base of structural instabilities of subsea PIP systems:

• Further investigation is suggested to understand the interaction between propagation buckling and upheaval/lateral buckling of the PIP system, in order to propose more comprehensive design rules.
• The speed of propagation buckle has not been thoroughly investigated for the single pipeline and PIP systems. Further research should be conducted to determine the parameters affecting this speed.

• The results of this research added insight into the applicability and efficiency of the CFRP buckle arrestor for a single pipeline system. However, an investigation on the efficiency of the CFRP buckle arrestor for PIP system should be considered for further research.
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