Wet Collapse behavior of Flexible UDW Risers

Literature study into the factors influencing the critical wet collapse pressure and mode shape

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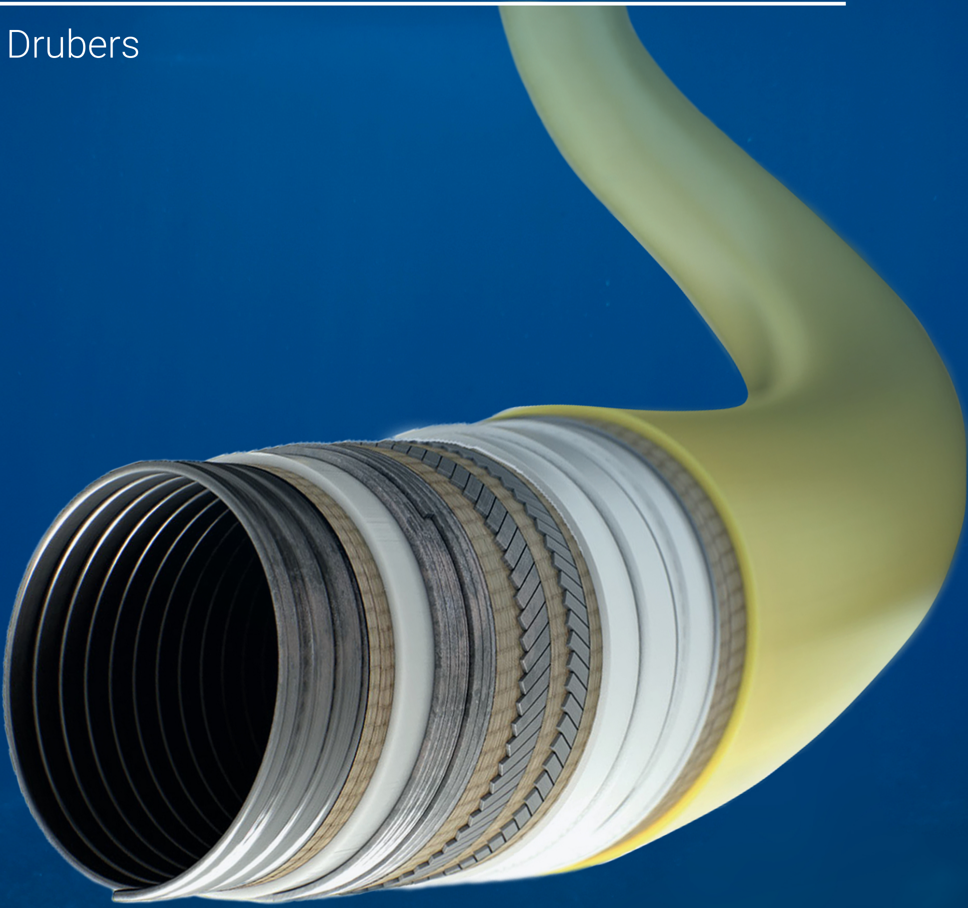

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Summary

Risers are pipes used for carrying oil and gas from wellheads and manifolds on the seabed to (floating) platforms on the water-surface or to shore. The current demand for oil and gas required the industry to exploit more difficult fields in ultra-deep water (UDW) environments. The extreme conditions in this UDW environment are dominated by the increased hydrostatic pressure and increased axial loads.

This requires the industry te develop risers capable of operating reliably in these conditions. In general there are two types of risers: rigid and flexible risers. Rigid risers are composed of steel and are susceptible to fatigue, something that can significantly reduce the lifespan of these structures. Flexible risers are designed to be compliant and do not suffer from fatigue during their lifespan. The main cause of failure for unbonded flexible risers is leakage into the annulus which can cause wet collapse.

The design of UDW risers is predominated by incorporating one or more pressure resisting layers that increase the collapse capacity, also known as the collapse pressure, of the rises such that it sufficiently large compared to the hydrostatic pressure present. When the hydrostatic pressure exceeds the risers' collapse capacity it will result in the radial buckling of the riser. This phenomenon is referred to as collapse failure. This failure mode of tubular structures had been a object of study since the late 1960s. Two common collapse modes of risers are known as ”heart mode” and ”eight mode”.

There are no mandatory regulation regarding risers, but manufacturers can acquire a certificate from a well established authority, like the American Petroleum Institute. This certification ensures the operator of the flexible riser that the quality and testing done is up to the standards of this authority which ensures a certain quality of the riser. The data required for certification is acquired by using either analytical, numerical or experimental methods that can analyze the collapse behavior of flexible risers.

The complex concentric multi-layered structure of unbonded flexible risers results in complex nonlinear interactions and behavior. Furthermore, the collapse is a plastic deformation which induces even more nonlinearities. This makes analytical methods unsuitable for this type of riser.

Riser collapse experiments yields the most reliable results but is a very costly method, especially as most flexible risers are custom build to suit the application and location. This would mean that each purpose build riser would require extensive testing before being approved by the customer.

The most feasible, and often used, approach is numerical analysis using Finite Element (FE) methods. FE models have become a very reliable tool in predicting the collapse behavior of flexible risers. FE models are also used to calculate the limitations of the riser, and prescribe the safe working conditions.

Even with these certification authorities failure of flexible risers occurs. The two most common of which are leakage and reduction of cross section better known as collapse. Failure is caused by operating outside specified design limits, errors in design or fabrication or internal or external damaging of the riser. The latter of which is most interesting in the examining the collapse of unbonded flexible risers. When the external sheath is damaged, it allows water to enter the annulus and the external pressure to pass along the pressure armor acting directly on the carcass. This is the lowest and most critical collapse pressure for flexible risers and this phenomenon is called wet collapse.

Using data from other research, mostly FE simulations, it was found that the collapse mode of flexible risers is influenced by several factors including geometric imperfections, material properties and the layer composition. The most influential factors found are:

- Curvature
- Gap width
- Ovalization (and geometric imperfections)
- Polymeric layer thickness
- D/t ratio
- Material properties
- Axial load
- Pressure armor strength (thickness)
- Carcass profile
- Pitch and lay angle of the carcass and pressure armor
- Friction between layers
- Initial interference between polymeric layer and pressure armor
- Mass flow rate

Pipes naturally tend to collapse in eight mode shape, and of these thirteen factors only three were found to significantly influence the collapse mode shape. These include the curvature, ovalization and pressure armor strength. The way these influence the collapse mode shape can be separated into two principles.

The first is by creating an amount of symmetry either in the loading or in the carcass shape. Curvature induces both ovalization and singly asymmetric loading and can significantly influence the collapse mode shape. The asymmetry in the carcass shape is mostly singly or doubly symmetric ovalization, where singly symmetric ovalization complements the heart shape mode and therefore increases the probability of heart mode collapse. Doubly ovalization has a large resemblance with the eight mode shape, and can induce eight mode collapse.

The second principle is by changing the radial stiffness of the layers surrounding the carcass, most significantly the pressure armor as this layer has the highest radial stiffness where its main purpose is to withstand external radial pressure. The radial stiffness is closely related to the thickness of the pressure armor where a small thickness results in a low radial stiffness and a hight thickness in a large radial stiffness. When the pressure armor has a high radial stiffness compared to the carcass it restricts the outward expansion of the carcass present with eight mode collapse, forcing the carcass into heart mode collapse. With a relatively small radial stiffness the pressure armor can ovalize with the carcass and allow it to collapse in eight mode shape.

Preface

Deepwater risers are used for carrying oil and gas from wellheads and manifolds to surface platforms. The increasing demand for oil and gas, and the scarcity of easily accessible fields in the deep sea environment, forces the industry to exploit more difficult fields in ultra-deep waters. This requires te industry to develop risers capable of operating in these ultra-deep water environments predominated by extreme external hydrostatic pressure.

Flexible risers are a rapidly developing and popularity gaining technology over the more conventional steel risers due to the higher resistance against fatigue. Flexible risers consist of multiple concentric layers. These layers can be unbonded, meaning they are able to move relative to each other. These layers work together to resist the axial and radial forces that arise when operating in an ultra-deep water environment.

This unbonded structure helps in the fatigue resistance but also makes the riser susceptible to ”wet collapse”, a phenomenon where the surrounding water enters the structural layers of the riser rendering some of the structural layers useless against external pressure. This reduces the collapse capacity of the riser and can result in radial buckling.

The radial buckling or radial collapse of tubular structures has been studied since the late 1960's. Radial collapse of flexible risers generally occurs in two modes known as ”eight mode” and ”heart mode”. Which collapse mode occurs is known to influenced by many different factors including geometric imperfections, material properties and the surroundings. How these factors influence the critical collapse pressure as well as the collapse mode shape is still unclear.

This literature study aims to give a general introduction into the state of the art of risers and the regulatory standards present, which ensure the quality of the flexible riser is sufficient to withstand the external pressure without collapse. Failure modes of flexible risers and techniques used for analyzing the collapse pressure will be discussed in order to finally shed some light on how these factors influence the collapse capacity and shape of flexible risers.

Project description

The subject of this literature project is the collapse mode of flexible ultra-deep water (UDW) risers under external hydrostatic pressure.

In nature, the collapse modes of risers under water are influenced by several factors, including geometric imperfections, material properties and their surroundings. However, how those factors influence the collapse modes as well as critical pressure of risers are still unclear.

This literature assignment aims to make an overview of the development of theories and approaches deployed for the research on the collapse modes of deep sea pipeline subjected to external pressure. The following aspects will be illustrated in the report:

- Development of deep sea riser industry, including riser types, structural configuration, material, relevant rules /standards etc.
- Main factors that affect the critical pressure and collapse modes of the risers.
- Available theories and approaches (analytical, numerical, experimental methods etc.) deployed to identify the reasons of collapse modes of risers.
- Possible collapse mechanisms of the risers when influenced by the those factors.
- Possible inter-relationship among the main factors that influence the collapse modes.

Contents

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State of the art of (ultra-)deep water riser development

Risers are pipes used for transporting oil and gas from wellheads and manifolds on the seabed to (floating) platforms on the water surface or to shore. The current demand for oil and gas requires the industry to exploit more difficult fields in ultra-deep water (UDW) environments. The extreme conditions in this UDW environment are dominated by the large hydrostatic pressure. This requires the industry te develop risers capable of operating reliably in these conditions.

The design of UDW risers is predominated by incorporating (multiple) pressure resisting layers that increase the collapse capacity, also known as the collapse pressure, of the rises such that it sufficiently large compared to the hydrostatic pressure present. When the hydrostatic pressure exceeds the risers' collapse capacity it will result in the radial buckling of the riser. This phenomenon is referred to as collapse failure.

A riser system forms the connection between the seabed and a interface on the surface of the sea in the form of a floater, ship or fixed rig. The dynamic loads and extreme service conditions imposed on the risers system makes it a critical component in the offshore pipeline development. According to the American Petroleum Institute (API) a riser system typically consists of [6]:

- Top interface
- Riser (Metal- or flexible pipe)
- Bottom interface

1.1. Riser systems

There are numerous types of risers, including Steel Catenary Risers (SCRs), Top Tensioned Risers (TTRs), Composite Risers, Attached Risers, Pull Tube Risers and Hybrid Risers. The choice of riser type is dependent on many factors. For example; Composite Risers are known to be expensive per unit length compared to SCRs, but are less expensive to install and are more forgiving to dynamic loads. Furthermore, Composite Risers can be designed with better insulation properties if flow assurance is a concern. [6]

Risers can generally be divided up into three types, based on the material being used. These are:

- Steel risers
- Composite risers
- Combined risers (steel and composite sections)

1.1.1. Steel Risers

Steel risers, as the name already suggest, are composed of steel. There are however different configurations for supporting the riser, including the Steel Catenary Riser, Top Tensioned Riser and risers fixed to the support structure of a platform such as the Attached and Pull Tube Risers.

Steel Catenary Riser

A Steel Catenary Riser (SCR, shown in figure 1.1) is a steel conductor pipe that serves as the prolongation of a sub-sea pipeline to a surface interface, where its function is to convey fluids from the sub-sea pipeline to the surface interface. They are suspended from the surface interface to the seabed and make an arced shape in between. The simplicity of the structure (which in essence only consists of a steel pipe) make SCRs a cheap configuration in current-day riser design, and a very popular choice for deep as well as UDW applications. [6, 14, 31]

The disadvantage of using SCRs is that steel is susceptible to fatigue and earlier research shows that SCRs have a limited dynamic performance. In the (ultra-)deep water environment loads like platform movements, Vortex Induced Vibrations (VIV) and sea current cause excessive bending in the riser structure, especially in the Touch Down Point (TDP,) which can result in fatigue.[14, 69]

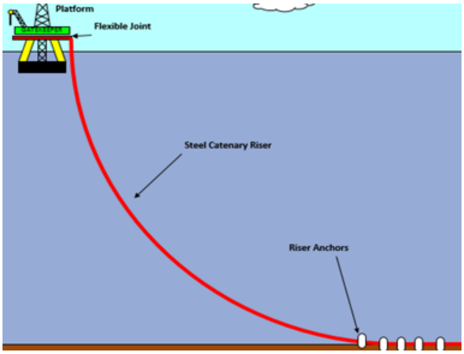

Figure 1.1: Steel Catenary Riser [33]

Top Tensioned Riser

Top tensioned risers (TTRs) are most often used for ”Dry Tree” applications in connecting a floating installation (tension leg platform) to the seabed. This type of riser is only suitable for floaters with very limited lateral movement like TLPs and Spars. TTRs are long cylinders that are provided with tensioners at the top to keep them stable along with their apparent weight. These tensioners are often a hydraulic heave compensator system or individual buoyancy tanks. These tensioners also allow the riser to move axially or stroke relative to the platform. TTRs were originally designed for shallow water but can now also be seen in deep-sea environments. [6, 44]

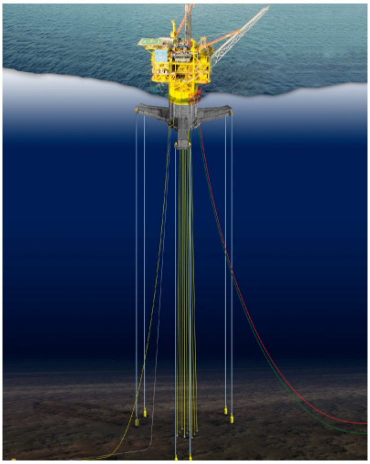

Figure 1.2: Top Tensioned Riser [57]

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Attached and Pull Tube Risers

The Attached Riser (AR) and Pull Tube Riser (PTR) differ from the SCR and TTR as they require a platform that is attached to the seabed. This can be fixed platforms, compliant towers or concrete gravity structures to which the risers are attached. This is also why these types of risers are not used for UDW applications as these depths require a floating platform.

The difference between AR and PTR is that the AR is used for transporting fluids or gases, whereas the PTR is used as a sleeve through which a flow-line is pulled by a winch. This can also be seen in figure 1.3. [6, 33]

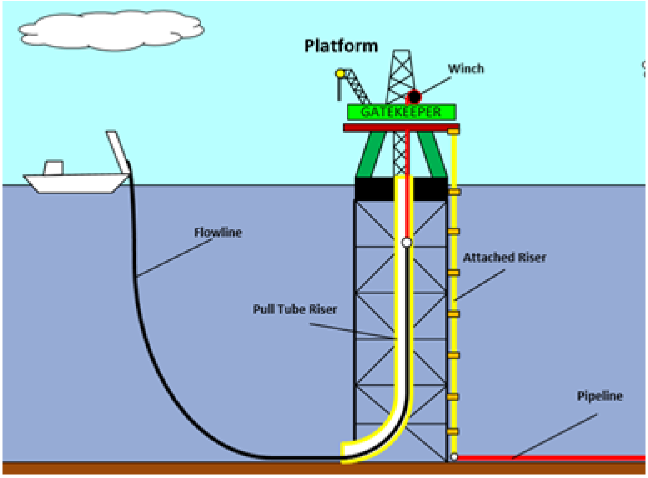


Figure 1.3: Attached and Pull Tube Risers[33]

1.1.2. Composite Risers

Composite risers are made up of different concentric layers. These layers are composed of polymers or helically wounded steel, allowing the riser to bend. These composite risers are therefore better known as flexible risers.

Flexible riser technology is a rapidly developing and popularity gaining technology. At first, flexible risers were used exclusively in fair weather environments but are currently also being used in various fields in the North Sea and the Gulf of Mexico where they have to withstand large (vessel and current) motions. Current developments allow the use of (unbonded) flexible risers in water depths down to 8,000 feet (2,438 meter) while withstanding high pressures and temperatures up to 10,000 psi (689 bar) and 150°F (65°C). [32]

Flexible risers have a low bending to axial stiffness ratio which causes them to be flexible while having little axial elongation due to axial stresses like gravitational force and movements of the floating installation. [5]

Risers are subjected to many different and extreme loads induced by the surface floater (surface waves and wind) and the direct environment (currents), on top of the functional loads (pressure, temperature, corrosiveness of fluids, etc.). The compliance of the riser allows it to cope with these dynamic motions without additional equipment like motion- or heave compensators. The compliance is increased or decreased accordingly by the use of buoyancy modules to form different riser loops which uncouples the motion of the top section, influenced by the movement of the surface floater, and bottom section of the riser. The configuration of buoyancy modules allows for different loop shapes such as the “Lazy S”, “Lazy Wave”, “Pliant Wave”, “Steep-S”, “Steep Wave”, etc. With increasing depths, and corresponding riser length, the simple free-hanging configuration also became feasible as the additional length provides enough flexibility to uncouple the motions of the seabed and the floater. [14, 27]

The flexible risers were originally designed and used for shallow water applications, but have currently been qualified for water depths up to 3000m. This is achieved by the use of composite materials (e.g. carbon fibre) resistive to corrosive fluids and are capable of withstanding high temperatures (e.g. 150°C) and pressures. Flexible risers can be used to make up the entire length of the riser, but can also be used in only a small section. Examples of this are jumpers and hybrid risers, the latter will be further discussed in section 1.1.3.

There are two types of flexible risers, bonded and unbonded risers. Both can have many different layer compositions. API RP 17B classifies flexible pipes into three categories, as shown in table 1.1. Deep-sea and ultra deep-sea flexible risers are either in family I of III, as they always require a pressure armor.

Table 1.1: Classification of standard, unbonded flexible pipes based on the layers that make up the riser according to API RP 17B [36]

Main Structural Layer	**Product Family I (Smooth Bore)**	**Product Family II (Rough Bore)**	**Product Family III (Rough Bore)**
Internal carcass		x	x
Inner liner	x	x	x
Pressure armor	x		x
Intermediate sheath	x[1]		
Tensile armor	x	x[2]	x
Outer sheath	x	x	x

[1] *The use of an intermediate sheath is optional.*
[2] *The cross-wound tensile armor may be applied with a lay angle close to 55°to balance radial and axial loads*

Bonded risers

In bonded pipes, different layers of fabric, polymeric plastics, elastomers and steel are bonded together in a process called vulcanization bonding. In general bonded pipes are only used in short shallow water sections, drag chain hoses for FPSO turrets and jumpers, but there are a few examples where they are used as long length riser in relatively deep water for both dynamic production risers and large bore oil export. The bonded risers considered are the types of pipe covered by API Spec 17K, which is the standardization of bonded flexible pipes according to the API. [3, 5, 37, 42]

Bonded flexible risers can often be used in the same applications as the unbonded flexible risers. However, some properties specific to bonded flexible risers are to be taken into account. First of all, bonded flexible risers are produced in limited lengths, and the maximum length differs with the diameter. For large bore risers (16 to 24 inch) the standard length of a section is 12 meter. For smaller diameters (4 to 10 inch) the length is typically less than 100 meter. It is possible to connect multiple of these sections to form a longer riser, where they are connected with steel joints, but are therefore less suited for (ultra) deep-sea applications than unbonded flexible risers, which can be produced in sections of several kilometers.

Secondly, bonded risers consist of multiple concentric layers of metal and polymeric thermoplastics like unbonded risers. The difference with unbonded risers is that the layers are bonded together, which results in larger shear deformations under bending in the thermoplastics. The only type of material that is able to subject large shear deformations are elastomers bonded to the steel armoring through a process called vulcanization bonding. [63] The properties of the riser are highly dependent on the various types of elastomers where there is also a wide spread of additives that further influence the mechanical, thermal and chemical properties. A small selection of possible elastomers and their properties are shown in table 1.2. The use of elastomers results in a smaller bending radius as it is capable of withstanding large shear deformations. Another construction that is frequently used for offshore applications is the bonded fiber-reinforced flexible hose. [29, 42]

Table 1.2: General properties of the most commonly used elastomers of bonded risers.[42]

Elastomer	**General Properties**
Butyl rubber	Excellent weather resistance, low air and gas permeability, good acid and caustic resistance, good physical properties, good heat and cold resistance, no resistance to mineral-oil-derived liquids
Chlorbutyl rubber	Variant of butyl rubber
Chlorinated polyethylene(CPE)	Excellent resistance to ozone and weather, medium resistance to oil and aromatic compounds, excellent flame resistance
Ethylene propylene rubber (EPDM)	Excellent ozone, chemical, and ageing properties, low resistance to oil-derived liquids, very good steam resistance, good cold and heat resistance (-40°C to +175°C), good resistance to brake fluid based on glycol
Hydrogenated nitrile rubber (HNBR)	Good resistance to mineral oil-based fluids, vegetable and animal fats, aliphatic hydrocarbons, diesel fuels, ozone, acid gas, diluted acids and caustics, suitable for high temperatures
Chlorosulfonated polyethylene	Excellent weather, ozone, and acid resistance, limited resistance to mineral-oil-derived liquids
Natural rubber	Excellent physical properties, high elasticity, flexibility, very good abrasion resistance, limited resistance to acids, not resistant to oil
Polychloroprene (Neoprene)	Excellent weather resistance, flame-retardant, medium oil resistance, good physical properties, good abrasion resistance
Acrylo-nitrile rubber (Nitril, NBR)	Excellent oil resistance, limited resistance to aromatic compounds, the resistance to fuel and flexibility to cold depends on ACN content
NVC (NBR/PVC)	Excellent oil and weather resistance for both lining and cover, not particularly resistant to cold
Acrylate rubber	Excellent oil and tar resistance at high temperatures

Bonded flexible risers differ greatly from unbonded flexible risers with respect to manufacturing method and overall design. Bonded flexible risers are primarily designed for applications where short lengths and a small bending radius is required, but can also be found as shallow sea risers. They have several advantages over unbonded flexible risers [42]:

- Produced in short sections (typically 10-12m for large diameters of 16-24”, or up to 100m for small diameters of 4-10”), therefore requiring less space in storing and installing. These smaller sections can be connected with reliable couplings to form a longer riser.
- Small bending radius due to the flexibility of the rubber.
- The production in short lengths means that special requirements can be incorporated into a small part (individual section) of the entire riser.

The disadvantages include:

- Limited length per section, a longer length requires joints.
- Much lower crash resistance, of critical pressure, than unbonded risers.
- Much lower resistance to axial forces.

These disadvantages make bonded flexible risers unsuitable for (ultra) deep-sea riser applications. They can sometimes be used for interconnecting shallow seabed installations over short distances, and to transport high pressure hydrocarbon. In the latter blistering is one of the possible failure modes in case of rapid decompression, two alternative designs have been used to avoid this blistering effect [3, 42]:

- Addition of a corrugated metallic inner liner that is diffusion tight, and therefore still flexible.
- Addition of an internal steel carcass that compresses the elastomer layer together with the reinforcement layers. This has proven to improve the blistering resistance.

Another type of bonded flexible hose often used in offshore applications is the fiber-reinforced flexible hose. It is also possible to have a steel fiber reinforcement, but fiber reinforced flexible hoses are not suited for (ultra-)deep water riser applications as they are not suited for operating with high external pressures as they lack a pressure sheath and will therefore collapse easily under pressure. Table 1.2 shows some typically used elastomers, and their respective properties, used in the production of bonded hoses.

Unbonded risers

Unbonded flexible risers are composed of different concentric layers that are not bonded together, meaning the layers are free to move relative to each other, only being restrained by the friction between them. This also means that the behavior of the riser depends on the interaction between these layers. They are typically operating with internal pressures of 70 to 700 bar with fluid temperatures up to 130°C. The exact composition of the different layers differs depending on the specific design requirements. These flexible risers are designed specifically for the intended application, there are no standard off-the-shelf products. This also means that the analysis and verification/certification has to be done for each newly developed riser as there is no generalized standard. More information on the certification and analysis will be given in chapters 3 and 4 respectively.[13, 74]

The use of flexible risers enabled the exploitation of a large number of fields that would otherwise (financially) be infeasible. Flexible unbonded risers are significantly more complex than steel rigid risers, being composed of multiple different layers each assigned a specific function. All of these layers have to be intact for the riser to function properly and safely. Unbonded flexible risers therefore have more vulnerabilities as each of the layers could fail and can all be influenced by different factors. This is also shown by the number of failures, where flexible risers have a higher failure rate than steel risers. One study of the PSA-Norway showed that the average life span of the flexible risers in the Norwegian offshore sector, where there had been close to 200 flexible risers in service, was only about 50% of their intended service life (typically 20 to 25 years).

The production of flexible risers is governed by the API, which prescribes the fabrication of all flexible pipe layers, quality control and mandatory documentation. All flexible riser producers comply with these API requirements and have to undergo a set of tests, or FAT (Factory Acceptance Tests), designed to reveal a large range of fabrication defects. However, failures related to the dynamics of the operating conditions or aging are not covered by the FAT, but should be covered in the design process. [1]

Unbonded flexible risers are produced in a continuous process, meaning that the end product is an flexible riser with a length of up to several kilometers. Transportation of these risers is therefore done by winding them on large reels. This process of winding on reels as well as the production and general handling has to be monitored closely as only a small ovalisation can already cause the collapse when the riser is put under tension or pressure.

Unlike bonded risers, unbonded risers are capable of being used in UDW environments which is a result of developments in the design and production in the last 25 years. Water depths greater than 2,000 meters are getting close to the limitations of conventional free hanging riser configurations, but the increasing demands for oil and gas urge the producers of flexible risers to develop flexible risers able to operate in water depths of 3,000 meter to explore new fields. One of the developments is the use of Carbon Fibre Composite

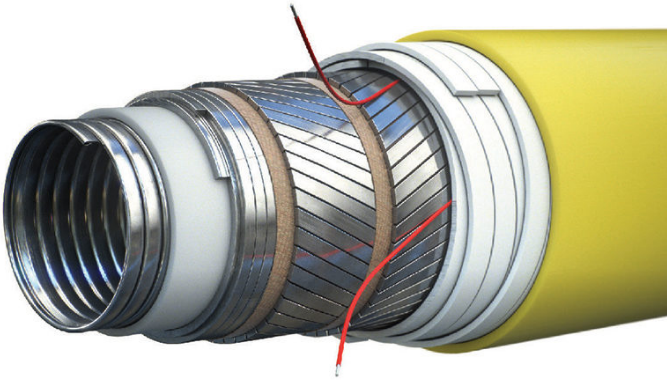


Figure 1.4: Unbonded Flexible Riser. [73]

(CFC) material.

The increase in market penetration of advanced composite materials like carbon fibre allowed for the development and the commercialization of composite materials for use in the automotive industry, aviation and wind energy applications. Although it may not seem very obvious to use such an advanced material that is famous for its large specific strength (ultimate tensile strength to density ratio), it may help solve one of the main difficulties limiting the maximum operating depth of flexible risers. In water depths of more than 2,000 meter the weight of a flexible riser becomes critical, not only for the tensile armor of the riser itself, but also for the production floater. [24]

1.1.3. Hybrid Risers

A hybrid riser is basically a combination between a TTR and a Flexible Riser. A rigid metal TTR rises up from the seabed to a buoyancy tank functioning as a distribution station or connector, also known as goosenecks, located 30 to 50 meter below the water surface. From this gooseneck a flexible riser connects to a ship or floating platform allowing a certain amount of relative motion between the two. Alternatively, there is not a single sub-surface buoyancy tank but multiple buoyancy modules attached to the riser, often made from synthetic foam. [6, 21, 33]

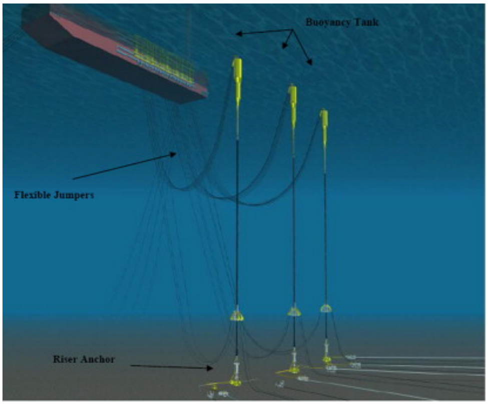


Figure 1.5: [33]

1.2. Riser materials

As mentioned before, in general two types of risers exist with respect to the materials being used, namely flexible and rigid risers. Some riser system configurations use a combination of flexible and rigid riser sections to accomplish a compliant but cost effective system. Pipelines that are transporting corrosive fluids may be fabricated from solid corrosion resistant alloys (CRAs), carbon steel clad with CRA lining or from flexible pipes with a corrosive resistant inner layer.[53]

1.2.1. Rigid risers

Rigid risers in general consist of a steel pipe or outer shell that can be lined on the inside for the reduction of corrosive and abrasive effects on the riser. Many different liners are used, mostly either made from an alloy (e.g. alloy 625) or a polymer. Some steel risers are metallurgically cladded (metclad) with an alloy to achieve the same result without the need for a separate liner.

The corrosive resistance of the steel itself can also be increased with the addition of nickel and/or chromium, which is especially effective against the corrosion of fluids containing carbon dioxide is suppressed. This corrosive effect reaches a minimum when the chromium content equals or exceeds 12% or nickel content equals or exceeds 9%. These solid CRA pipes are often made of stainless steel and have to comply with API specification 5LC.

Internally clad pips are made from a carbon-manganese steel lined with a thin layer of corrosion resistant material (typically 2-3mm). This lining can either be metallurgically bonded or added as a separate tight fitting layer. Some CRAs that are commonly used are stainless steel (319L) and high-nickel alloys (type 825 and 625).[53]

1.2.2. Flexible risers

Flexible risers are made up of different concentric layers of metals and polymeric thermoplastic materials, all with a specific task. Depending on whether the construction of the riser is bonded or unbonded the materials used can differ. In a unbonded structure the layers are able to move relative to each other whereas with a bonded structure the layers are bonded together through a vulcanization process.[70]

The main layers consist of extruded polymer sheaths functioning as fluid barriers and steel armour windings that provide the strength while allowing compliance. The exact design of the layers and material selection is based on the operational environment.

The production of these multilayer composites is sequentially done from the inside outward. In general, the layers (from the inside to the outside) consist of the carcass, the liner, layers responsible for handling the loads from the longitudinal and radial stresses and finally an outer sheath, also shown in figure 1.6.

Carcass

The carcass is the innermost layer through which the carried fluid flows. The carcass is not gas- or fluid-tight, but is surrounded by an liner which is. The function of the carcass is to prevent the collapse of the liner due to external hydrostatic pressure, ensuring that the fluid being transported has a free path to flow through. When operating at low pressures it also helps to retain this low pressure within the transporting fluid or gas by counteracting external pressure. [50]

The carcass is often made of interlocking helical sheet metal. Typical materials being used are carbon steels (AISI 4130), austenitic stainless steel (AISI 304, 304L ,316, 316L) and duplex stainless steel(UNS S31803). In some applications there is no carcass, also known as smooth bore pipelines. These are used mainly for transporting stable crude oil and as water injection.[53]

The external pressure that acts on outer layer of the riser is due to the water surrounding the riser. This pressure increases linearly with the water depth with about 0.1 bar per meter. This means that at 3000m below the sea-surface the external pressure on the riser is near 300 bar. The carcass on itself is not capable of handling this extreme pressure and is therefore supported by a pressure armor, as well as all the other layers that provide a (small) amount of resistance against this external force.

Liner

The liner, sometimes referred to as pressure sheath, is often made from high-density polyethylene, nylon and fluorinated polymers. The liner is in direct contact with the fluid being transported, and the main factor determining the service life of the liner is therefore the degradation it endures due to this contact. Which liner is being used is dependent on the fluid being transported as well as the temperature at which it has to operate. [56]

Pressure armor

The pressure armor is located between the inner liner and the tensile armors, providing pressure resistance to both internal and external pressures. As it surrounds the inner liner it prevents this liner to expand due to a positive pressure difference between the inner transport fluid and the seawater surrounding the riser. And as it is located within the external sheath it also resists external pressures acting upon the outside of the riser. This layer mostly consists of helically wounded steel, such that it can cope with the hoop stresses that result from the pressure difference between the fluid being transported and the surroundings. [50]

Tensile armor

The tensile armor layers often form the outermost metallic layers of the riser. Their purpose is to provide axial rigidity while minimally compromising the flexibility. This layer also contributes to the capability to withstand large internal pressures of the riser. Is is however of little use in resisting external pressure as it is made from helically wound strips that do not interlock. Pressure from the inside can therefore be resisted but pressure from the outside will make the strips to buckle and deform rather easily.

Outer sheath

The outer sheath, applied as a hot melted plastic, serves as an protection layer against corrosion of the steel armouring. This sheath is most often made from high density polyethylene (HDPE), because it has some properties that make it very suitable for the extreme underwater environment of risers. These properties include good adhesion, extensibility, abrasion resistance, electrical properties and low water absorption.

The space between the internal and external sheaths is known as the annulus and contains the pressure and tensile armour layers. A key role of the external sheaths is to protect the annulus agains external environment. [19]

Additional layers may be used for reducing the gas permeability, increase flexibility by allowing movement between the different load holding layers (often made from steel), reduce wear or to provide additional thermal insulation. [53]

This multi-layered structure makes the production and material costs much higher than that of rigid risers. However, the process of laying and installing the pipe is much cheaper, simpler and faster.

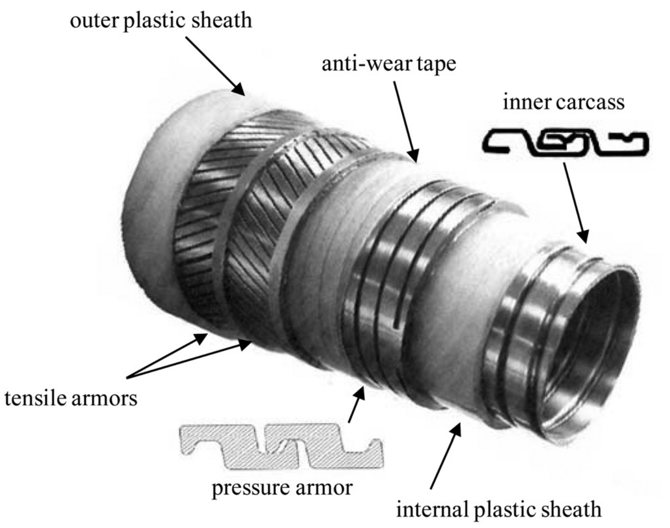


Figure 1.6: Multi-layer composition of flexible composite risers.[65]

1.3. Conclusion

Riser can differ greatly in terms of complexity, ranging from the simple SCR to the complex flexible multi-layered composite riser. In general three types can be distinguished: steel, composite and hybrid risers, where the last is a combination between steel and composite risers. The configuration in which these risers can be used is very diverse, and which is best is dependent on many factors. With respect to the material being used a division can be made between rigid and flexible risers. Rigid risers generally consist of a steel pipe with an optional inner liner, whereas flexible risers consist of multiple concentric layers. Flexible risers come in two configurations: bonded and unbonded. In a bonded flexible riser the layers, consisting of fabric, polymeric plastics, steel and elastomers, are fused together in a process known as vulcanization. Bonded pipes are primarily used for short sections or in shallow water applications and are therefore not in the scope of this research. Unbonded flexible risers are however capable of being used in UDW environments, and the layers generally consist of at least: a carcass, liner, pressure armor, tensile armor and outer sheath. These layers are free to move relative to each other which results in the flexibility. The danger of the unbonded structure is the susceptibility to collapse when water enters the annulus.

2

Failure of unbonded flexible risers

The use of flexible risers in UDW applications comes with increased loads on the riser, these loads mainly consist of:

1. Increased axial loads due to the top tensioning and the associated fatigue, a result of the increased length and thus weight of the riser.
2. Increased hydrostatic pressure, especially when the riser is empty for instance during installation and production stops.

These problems are addressed by some design changes. A commonly used method is to design a riser in two different sections. The philosophy behind this is that the two before mentioned loads each have the highest influence in different sections of the riser.

In the top section the most significant load is the axial load due to top tensioning, as almost the entire weight of the riser is pulling on this section. This means that there is more stress in the tensile armor of top section than there is in the bottom section. By using stronger tensile armor in the top and a weaker and more lightweight tensile armor in the bottom section, a more efficient use of material is established while also reducing the weight and thus the stresses.

In the bottom section the hydrostatic pressure is predominant. A similar method can therefore be used with respect to the pressure armor and carcass, by using a stronger pressure armor/carcass in the bottom section while reducing the weight of these components in the top section. [11]

2.1. Failure types

The two most common types of failure in the operation of flexible riser are leakage and reduction of cross-section, better known as (radial) collapse.[77] Beside these failure modes there are some other less common failure types that are also described by API Recommended Practices 17B.

2.1.1. Leakage

Leakage is related to the failure of one or more of the polymer liners responsible for forming fluid or gas barriers. In total three types of leakage can occur considering a flexible riser with the layers as described in the previous chapter. The first is leakage of the inner liner, allowing the fluid or gas being transported to enter the "annulus", this is the space between the outer sheath and the inner liner. This type of leakage often does not imminently lead to a total failure of the riser as external pressure on the outer sheath prevents the riser from expanding any further. However, it could lead to bird-caging and failure of the outer sheath.

This second type of leakage results in total failure of separating the transported gas or fluid from the external environment as all the fluid barriers have failed.

The final type of leakage is failure of only the external sheath. This means that the water surrounding the riser can enter the annulus. If the water enters the annulus it passes the tensile and pressure armor and pressure acts directly on the internal liner. This can cause the carcass and liner to collapse.

2.1.2. Collapse

Reduction of cross section can have many different causes. One of the most significant is the failure of the carcass, the component responsible for preventing the liner from collapsing, which results in a partial or total reduction of the cross section.

Carcass Collapse

In general there are two ways the carcass and pressure sheaths can collapse under external pressure, these are known as *dry collapse* and *wet collapse*. Ordinarily, an intact flexible riser is subjected to external pressure acting on the outer sheath. This way all the layers, especially the carcass and pressure armor, work together to sustain this external pressure. When all these layers fail together in resisting this external pressure and the riser collapses it is known as dry collapse.[9]

Wet collapse however is a result of the structural failure of one or more of the polymer layers, excluding the inner liner, such that external fluid is able to enter the annulus. This means that the external pressure is directly applied to the carcass, as the liner is incapable of resisting this pressure. This situation is often critical in the design of the riser and therefore the main focus for this study. [50]

Collapse mode

Collapse of the carcass generally occurs in two shapes or modes: "Eight" mode and "Heart" mode. An unconfined and not significantly deformed tube will naturally collapse in an eight mode under external pressure [26]. However the collapse mode can be influenced by different factors, as explained in chapter 5, such that heart mode collapse is induced. Eight mode collapse is also known as four hinge collapse. This is because four points on the circumference deform plastically and effectively form hinges, also shown in figure 2.1. Therefore, it is only these four points that have to deform where the remaining sections can stay intact not requiring any deformation energy.

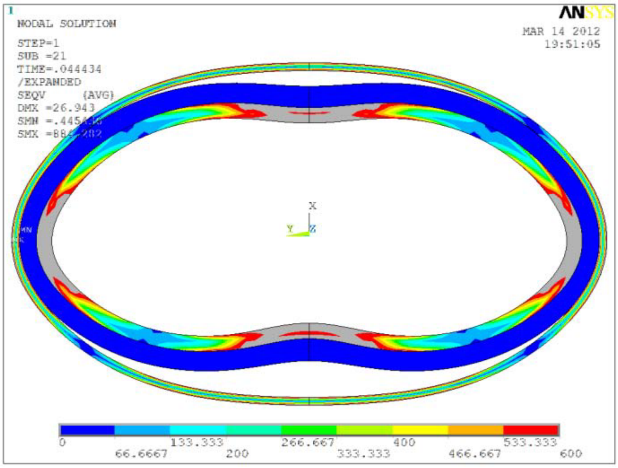


Figure 2.1: Von Mises stress for the "eight" mode collapse. The gray parts have plastically deformed and act as hinges. [45]

Collapse capacity

The pressure at which collapse occurs is called the collapse capacity, critical capacity, collapse pressure or critical (collapse) pressure. Two of the most significant influencing factors on this critical pressure of risers, and pipes in general, are geometric and material imperfections. The geometric imperfection is often referred to as the ovality or elliptical imperfection. Flexible risers are allowed to have a small imperfection in the ovality in the production, see chapter 3, but can also be damaged during storage, transport or installation [25]. Geometric imperfections also include indentations.

Material imperfections can be related to shortcomings in the production or handling of the material. All materials have some imperfections which influences their properties, most significantly the (local) yield strength and Youngs modulus. This effects the stress-strain response of the material. Corrosion, errosion and fatigue are the most famous and often occurring material imperfection. Material imperfections can be accounted for in the design of the riser using Finite Element (FE) analysis where a part of the cross-section has a smaller thickness, simulating a grove or eroded section. [61]

2.1.3. Other types of failure

The increase in operating depths creates an increase of external pressure on the riser, which can result in an increase in failures. This compressive force has to be sustained by the carcass and pressure armor layers. API Recommended Practice 17B describes the most common failure modes for unbonded flexible risers and the potential failure mechanisms as shown in Table 2.1. Of the failure modes listed, two can be more likely to happen in UDW environments, besides collapse, due to the extreme external pressure and tension; tensile and compressive failure.

Tensile failure is the failing of the tensile layer due to the weight of the riser, or by external factors that can cause excessive tension in the riser such as over-bending or snagging by fishing trawl board or an anchor. This tensile stress due to excessive tension is normally handled by the tensile armor, but the stresses can also be transferred to the carcass or pressure armor, the effect of this is further discussed in section 5.1.7. Excessive tension in the tensile armor also increases the external pressure on the pressure armor and carcass due to the helically wound structure of the tensile armor, which makes it tend to move inward under axial tension.

The helically wound layers are designed to be under tensional loads, when they are under compression they can fail due to this compression. This improper loading can lead to bird-caging. The buckling under axial compression of the tensile armor is also referred to as lateral buckling. This causes the individual strips to expand outwards resulting in the rupture of the outer sheath. This differs from failure of the carcass or pressure armor as it is a result of an improper load, where the carcass or pressure armor fail under their intended loading direction. The name bird-caging comes from the shape, which can be seen in figure 2.2 The simple way to prevent bird-caging is to ensure the riser is never under improper loading, and is therefore not a subject of interest in the failure of the flexible riser in UDW applications. [10, 65]

Tensile failure is also out of the scope of this investigation as the focus is on the collapse under hydrostatic pressure, not due to excessive axial or lateral forces. However, it is important to be aware of that the carcass and pressure armor can be under axial tension which can lead to failure. The effect of axial tension on the critical collapse pressure is also described in section 5.1.7.

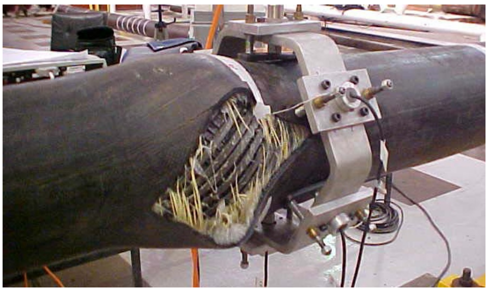


Figure 2.2: Bird caging effect. [65]

Table 2.1: Pipe failure modes and their potential failure mechanisms [36].

Pipe failure mode	**Potential failure mechanism**
Collapse	1) Collapse of carcass and/or pressure armor due to excess external pressure or tension. 2) Collapse of carcass and/or pressure armor due to installation loads or ovalization due to installation loads. 3) Collapse of internal pressure sheath in smooth-bore pipe.
Burst	1) Rupture of pressure armors because of excess internal pressure 2) Rupture of tensile armors due to excess internal pressure
Tensile failure	1) Rupture of tensile armors due to excess tension. 2) Collapse of carcass and/or pressure armor and/or internal pressure sheath due to excess tension 3) Snagging by fishing trawl board or anchor, causing overbendin or tensile failure
Compressive failure	1) Bird-caging of tensile amor wires. 2) Compression leading to upheaval and excess bending.
Overbending	1) Collapse of carcass and/or pressure amor or internal pressure sheath. 2) Rupture of internal pressure sheath. 3) Unlocking of interlocked pressure or tensile-armor layer. 4) Crack in outer sheath.
Torsional failure	1) Failure of tensile armor wires. 2) Collapse of carcass and/or internal pressure sheath. 3) Bird-caging of tensile armor wires.
Fatigue failure	1) Tensile armor wire fatigue. 2) Pressure armor wire fatigue.
Erosion	Of internal carcass
Corrosion	1) Of internal carcass 2) Of pressure- or tensile armor exposed to seawater. 3) Of pressure- or tensile armor exposed to diffused water.

2.2. Causes of failure

Risers are designed to last for a certain period, also know as the expected service life. Failure to fulfill this expected service life, and the cause of the failures as described in table 2.1, is due to either of three reasons:

1. Operating outside specified design limits
2. Errors in design or fabrication
3. Internal or external damaging of the riser

The first is often a result of slowly changing operational conditions, where the operators are not fully aware of this change or the effect it has on the riser. These changing conditions can result in the operating conditions exceeding the design limits of the riser, resulting in failure. Often there are control systems present that are aware of the limitations of the riser to prevent this type of failure. Errors in design and fabrication have to mitigated as much as possible, this is also why API Spec 17J and 17B been developed to prescribe standards and standard practices for the design, fabrication, installation and operation of flexible pipes.[47]

The greatest cause of failure however is due to internal or external damage. In particularly the damaging of the outer sheath and flooding of the annulus are the prime mechanics for failure and collapse of the riser, as shown in figure 2.3. Although this data is from 2002 and only covers the UK and Norwegian sectors of the North Sea it clearly shows the relative frequency of these types of failure. The report also mentioned 26 incidents during the installation and commissioning of which 58% of the incidents included damage to the external sheath and 19% had a flooded annulus. [38]

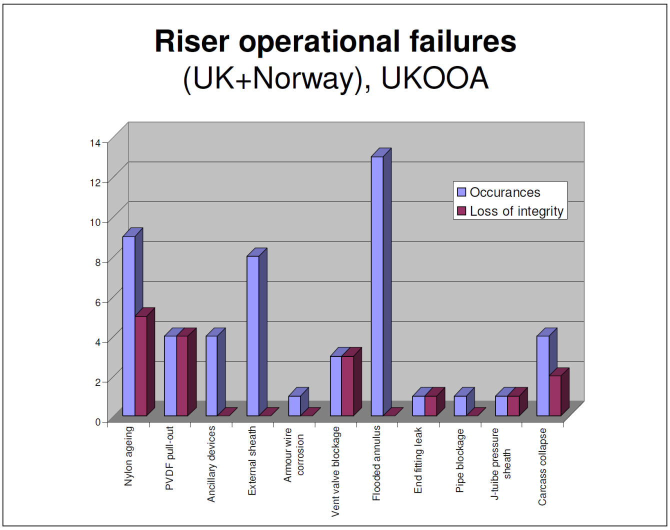


Figure 3-1: UKOOA statistics on flexible pipe failures

Figure 2.3: Statistics on types of failure of flexible pipes in the UK and Norwegian regions of 2002.[47]

2.3. Conclusion

UDW environments increase the load imposed on the riser in two ways. The first is higher hydrostatic pressure in the bottom section of the riser. The second is larger axial tension in the top section due to the increased length and consequently weight of the riser. With these increased loads the riser is more susceptible to failure, where especially the hydrostatic pressure causes many of the failures of flexible risers. The failure of risers can be divided into two types: leakage and radial collapse. The most occurring phenomenon is flooding of the annulus, which could lead to the radial collapse of the carcass especially in the UDW environment due to the extreme external pressure. This type of collapse is known as wet collapse. Failure in general is caused by one of three reasons: operation outside specified design limits, design or fabrication errors or (internal or external) damaging of the riser. Collapse of the riser generally occurs in one of two shapes or modes: heart mode or eight mode. An unconfined pipe has a natural tendency to collapse in eight mode, where four parts of the cross-section that deform plastically effectively form hinges. This collapse mode is because of this phenomenon also known as four hinge collapse.

3

Rules, standards and certification

Initially manufacturers' experience of in house testing of risers set the specifications for flexible risers. In the late 1980s both the API and DNV developed guidelines generally accepted industry; API RP 17B: recommended practice for flexible pipe (1988) and DNV Guidelines for flexible pipes (1987). In 1994 "Rules for certification of flexible risers and pipes" was published by the DNV, followed in 1996 by the API with the industry standard API Spec 17J named "Specification for unbonded flexible pipes", which included standards for the design, materials, manufacturing, documentation and testing of unbonded risers. API RP 17B was updated in 1998 and 2002, with information on bonded and unbonded flexible pipes for onshore, subsea and marine applications. Supplementary equipment of flexible pipes is covered by API Spec 17L, bonded flexible pipe have a separate section called API Spec 17K. [13]

3.1. Authorities and standards

Many different authorities and classification societies have developed codes with respect to design of risers, these include ISO, API, NPD, HSE, NS, BS, CSA, DNV and ABS. The most interesting and commonly used ones with respect to flexible riser design are the ISO, API and DNV standards. More specifically the following [5, 23, 35]:

- API
  - API Spec 17J, Specification for Unbonded Flexible Pipe
  - API Spec 17K, Specification for Bonded Flexible Pipe
  - API-RP 17B, Recommended practive for flexible pipe
  - API RP 2RD, Design of Risers for Floating Production Systems (FPSs) and Tension-Leg Platforms (TLPs)

- DNV
  - DNV-OSS-301 Verification and Certification of Submarine Pipelines
  - DNV-OSS-302 OFFSHORE RISER SYSTEMS
  - DNV-OS-F201 Dynamic Risers
  - DNV-OS-F101 Submarine Pipeline Systems
  - DNV-RP-F202 Composite Risers
- ISO
  - ISO 13628-2 Subsea flexible pipe systems
  - ISO 13628-11 Flexible pipe systems for subsea and marine applications

These standards regarding the design of flexible risers, and risers in general, are not mandatory but provide a certification, if the standard is used and followed properly, that is well established in the offshore sector. Following a standard does not inhibit anyone from using any other practices, and is not binding or gives any guarantees with respect to the quality of the product. It creates a form of reassurance for the riser operator that the product is designed and tested appropriately and helps proving the quality of the product being produced. Manufacturers apply a marking of the standard followed during the design and testing, in conformance with the marking requirements, on the riser if this is correctly used. This manufacturer is responsible for complying with all the applicable requirements of that standard. [23, 35]

3.2. Design requirements

In general, two types of design requirements are prescribed [5]:

- mandatory requirements
- Recommendations to satisfy the mandatory requirements

The mandatory requirements in essence form a checklist of the essential properties, parameters and guidelines that need to be met by the pipe manufacturer and verified by the pipe operator ordering the flexible riser. It also prescribes the information the pipe operator has to provide to the manufacturer responsible for designing the flexible riser. The operator purchasing a flexible riser must provide the system requirements of the project to the manufacturer as prescribed in section 5.6 of API Spec 17J, and this shall be defined in the design premise including design load cases. I.e. the purchaser should specify flow line parameters shown in figure 3.1 and shall specify riser parameters as shown in 3.2 to the manufacturer. These parameters influence much of the pipe design, such as material selection and layer thickness.

This also shows the way the standards for flexible risers are to be read, where the verbal forms defines how the requirement is to be interpreted. Whenever “shall” is being used the requirement has to be strictly followed, “should” indicates that among several possibilities one is recommended as particularly suitable, without excluding or mentioning the others, or to indicate that one course of action is preferred but not necessarily required. Sometimes “may” is used to indicate a course of action permissible within the limit of the document. [23]

Parameter	Details
Flowline routing	Route drawings, topography, seabed/soil conditions, obstacles, and installed equipment and pipelines
Guides and supports	Proposed geometry of guides, I-tubes, J-tubes, and bellmouths through which flowline is to be installed
Protection requirements	Trenching, rock dumping, mattresses, and extent of protection requirements over length of pipe. Design impact loads, including those from trawl boards, dropped objects, and anchors
On-bottom stability	Allowable displacements
Upheaval buckling	Specification of design cases to be considered by manufacturer
Crossover requirements	Crossing of pipes (flexible and rigid), including already installed pipes and gas lines
Pipe attachments	Bend restrictors, clamps, and attachment methods
Load cases	Definition of yearly probability for installation and normal and abnormal operation. Specification of accidental load cases and yearly probabilities

Figure 3.1: Table to be provided by the operator to the manufacturer specifying the required flowline parameters. [35]

Parameter	Details
Riser configuration	Specification of any requirements for the configuration, including description (lazy-S, steep wave, etc.), layout and components. Selection of configuration or confirmation of suitability of specified configuration
Connection systems	Descriptions of upper and lower connection systems, including quick disconnection systems and buoy disconnection systems, connection angles and location tolerances
Pipe attachments	Bend stiffeners, buoys, etc., and attachment methods
Attached vessel data	Data for attached floating vessels, including the following: a Vessel data, dimensions, drafts, and the like; b Static offsets; c First (RAOs) and second order motions; d Vessel motion phase data; e Vessel motion reference point; f Mooring system interface data; g Position tolerances.
Interference requirements	Specification of possible interference areas, including other risers, mooring lines, platform columns, vessel pontoons, tanker keel, and so on, and definition of allowable interference/clashing
Load cases	Definition of yearly probability for installation, and normal and abnormal operation specification of accidental load cases and yearly probabilities

Figure 3.2: Table to be provided by the operator to the manufacturer specifying the required riser parameters. [35]

The recommendations form supplements to the mandatory requirements, where the mandatory requirements have to be proven to the customer. The manufacturer is free in choosing how it is to do so, but the most common way is to show analytical and/or numerical results of the various tests of the flexible riser. The minimal functional requirements that shall be demonstrated by the manufacturer are [35]:

1. The pipe shall provide a leak-tight conduit.
2. The pipe shall be capable of withstanding all design loads and load combinations defined herein.
3. The pipe shall perform its function for the specified service life.
4. The flexible pipe materials shall be compatible with the environment to which the material is exposed.
5. The flexible pipe materials shall conform to the corrosion control requirements specified herein.

The ISO 13628-2 or API Spec 17J is the most widely used standard in unbonded flexible riser technology and will therefore form the main reference model in this report with respect to the rules and standards, together with the recommended practices as described in API RP 2RD and API RP 17B.

API Spec 17J prescribes functional requirements, design requirements, material usage, manufacturing requirements, marking and packaging, included documentation and factory acceptance tests for flexible risers. In recent years many of the API-standards have been converted and updated into ISO standards, which ensures better international standardization. Furthermore, currently the ISO standard is nationally adopted by the API/American National Standard. [34]

One of the design requirement prescribed by API Spec 17J applies to the maximum amount of permanent ovalization. In API Spec 17J, a permanent ovalization of 0.2% is considered to be acceptable, where the ovalization is given by [35] :

$$Ovalisation = \frac{D_{max} - D_{min}}{D_{max} + D_{min}}$$

where D denotes the carcass diameter. As a comparison, DNV 0S-F101 has a slightly different specification where the ovality normally is not allowed to exceed 2%, or 0.02, using the following criteria [22]:

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

Another important factor is the minimum bend radius (MBR). This is limited by the allowable strain in the polymeric layers and relative movement of the metallic armour wires during bending. API spec 17J has prescribed MBR requirements with respect to storing, static and dynamic applications of the riser [35]. This is summarized in table 3.1.

Table 3.1: API Spec 17J requirements with respect to the MBR. [36]

	MBR Design criterion
Storage	1.1 times the MBR causing locking in armor wires. 7.7% strain for PE and PA. 7.0% strain for PVDF.
Static applications	1.0 times storage MBR.
Dynamic applications	
Normal operation	1.5 times storage MBR or 3.5% strain for PVDF
Abnormal operation	1.25 times storage MBR or 3.5% strain for PVDF

API Spec 17J is based on working stress design, where the working stresses are compared to the permissible stresses within the elastic regime following Hooks law, i.e. a linear stress-strain response is assumed. The renewed ISO and DNV standard use the limit state method instead, where the stresses within the material is allowed to surpass the yield limit to reach the ultimate tensile strength and thus enters the plastic regime. This means that Hooks law does not apply and a non-linear stress-strain curve is assumed. This makes the design of the riser more economical as material can effectively be used with higher ultimate utilization factors. The disadvantage is that it requires more computational effort and non-linear calculations. [2, 62]

3.3. Analysis for certification

The analysis of flexible riser design mostly consist of three parts. The first is a cross-section analysis where the mechanical properties of the riser are predicted under different operational conditions, as well as determining the load shearing between the individual layers. This cross section analysis is also used to predict the mechanical properties, distribution of stress, failure model, etc.

Second is the global analysis The function of a global analysis is to evaluate the global load effects on the riser in order to determine/approximate the performance of the riser. The global analysis should contain and calculate the static configuration and extreme response of displacement, curvature, force and moment from environmental effects.

An global analysis contains two aspects: a static analysis and a dynamic analysis. The static global analysis determines the equilibrium position and configuration of the system under its own weight, buoyance and static drag forces. The results of this static analysis regarding the equilibrium configuration/position can be used for the dynamic analysis, as this most often forms the best starting point. A dynamic analysis is a time simulation of the motion of the riser under different (time-dependent and changing) loads and load cases.

Thirdly there is the fatigue analysis where the effect of cyclic loads on the riser is considered, one of them is the vortex-induced vibration (VIV). Flexible risers have a large damping factor as (the friction between) the unbonded layers absorb this motion and therefore do not suffer from fatigue damage as a result of VIV, and vibrations in general. However, a detailed fatigue life analysis is required and the manufacturer need to prove that the fatigue life of the riser is ten times the pipes required service life. [5, 35]

The basic analysis used for the design of flexible risers with respect to collapse resistance and behavior is the cross sectional analysis. A lot of research has been done in the local analysis of flexible pipes which can be divided up into three approaches: experimental, analytical and numerical simulations. The experimental data is also used as a reference for the analytical and numerical approaches to validate the results. The manufacturers of flexible risers often use analytical models or Finite Element (FE) models to prove the design to customers and for the validation and certification of flexible risers. [5, 36, 67]

3.4. Conclusion

The riser industry does not use rules for maintaining riser quality, instead manufacturers prove their product to customers. There are different authorities that provide certifications to risers that help in ensuring the customer of the quality of the riser. In order to acquire such a certificate certain specifications as prescribed by these authorities have to be met. The most commonly used are the API standards, which are now also used in the ISO standards. In order to acquire the API certification, the manufacturer has to prove that the riser meets the required riser parameters and flowline requirements given by the customer, and operate safely in the conditions where it will be used. The manufacturer is free in choosing how it is to do so, but the most common way is to show analytical and/or numerical results of the various tests of the flexible riser. This analysis mostly consists of three parts. The first is a cross-section analysis where the mechanical properties are inspected as well as the load shearing between individual layers. This is also used to predict the mechanical properties, distribution of stress, failure model, etc. The second is a global analysis that evaluates the global load effects to determine the performance of the riser. The third is a fatigue analysis inspecting the effect of load cycles. Due to the large damping factors in the riser, due to the friction between the unbonded layers, fatigue is rarely a factor for failure in unbonded flexible risers. Manufacturers often use analytical models or Finite Element (FE) models to prove the design to customers and for the validation and certification of flexible risers.

4 Analysis of the collapse pressure

Flexible risers require extensive analysis in order to be certified, which ensures customers that the product is sufficiently reliable and capable of operating in the required conditions. This analysis can be done experimentally, analytically or numerically. Due to the reliability and relative low costs of numerical analysis, Finite Element (FE) models are most commonly used in the industry.

One of the simplifications commonly used for reducing the calculative intensity of the collapse analysis is by using a symmetry plane. The two most occurring failure modes are either singly or doubly symmetric(further explained in section 5.2.2) : heart mode and eight mode respectively, also shown in figure 4.1. This means that when a double symmetric analysis is done considering a 90°, or quarter, cross-section, only the eight mode can be found. This is something to keep in mind when comparing the results of buckling analyses, as eight mode collapse could be forced due to this simplification.

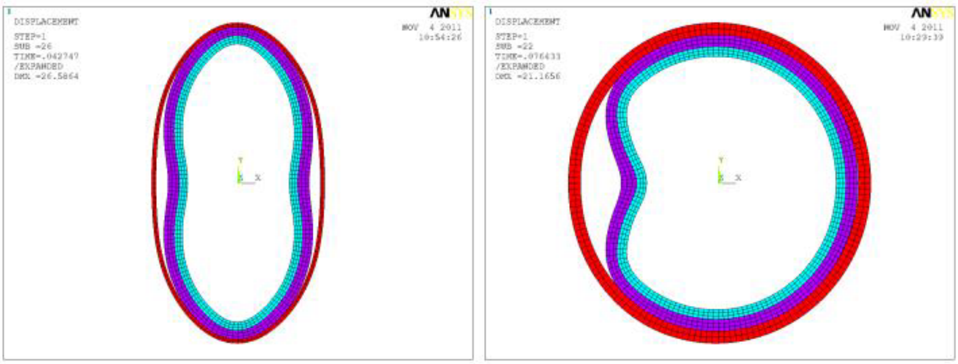


Figure 4.1: Eight mode (left) and heart mode (right) collapse shape. [45]

4.1. Experimental tests

Experimentally proving the collapse strength is the most accurate method for determining the collapse behavior of flexible risers. The downside is that experimental testing is a very costly method, especially as most flexible risers are custom build to suit the application and location. This would mean that each purpose build riser would require extensive experimental testing before being approved by the customer. With it comes that the collapse strength is only found after designing and manufacturing the riser This means that when the collapse strength is found to be insufficient, the design, manufacturing and testing process has to be redone.

Furthermore, many different flexible riser layer compositions and designs are possible and determining the influence of all the possible individual layers experimentally would be infeasible. Therefore analytical and numerical models are most often used that are regularly calibrated and validated using data from experimental tests. [9]

Some experiments have been done where the collapse resistance of flexible pipes was determined, mostly for certification and quality control purposes. These are done in hyperbaric chambers where the pressure is gradually increased while the core of the riser is kept at atmospheric pressure. Often some typical pressures were maintained for a longer period of time to ensure the riser complies with the standard being used, as seen in figure 4.2, after which the pressure is increased until collapse occurs. Furthermore, the pressurization rate is also kept at a constant.

An important parameter of the test samples is the length, often expressed in units of diameter of the test sample. A typical standard length is at least 7.5 times the outer diameter (in accordance with ASTMD2924), which ensures that the end fittings of the sample don't interfere with the buckling behavior. API RP 17B also prescribes a testing methodology for performing buckling collapse tests. Figures 4.3a and 4.3b show examples of test setups, and figure 4.4 shows the results gathered with these test setups. Some results from another buckling collapse test where flexible risers with an internal diameter of 2.5 and 4 inch were tested, following roughly the same testing methodology, are shown in tables 4.1 and 4.2. [9, 16, 36, 54, 55]

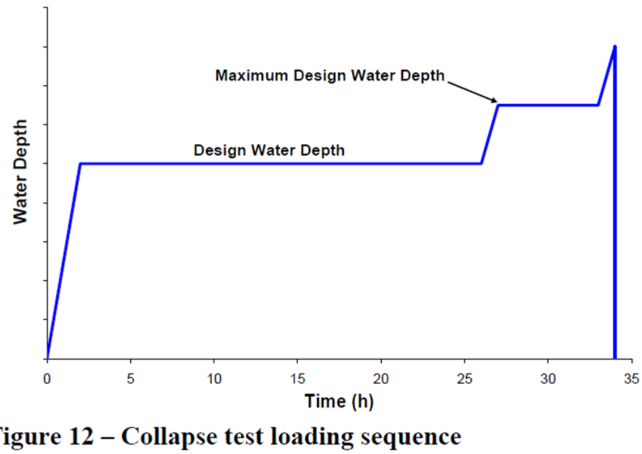
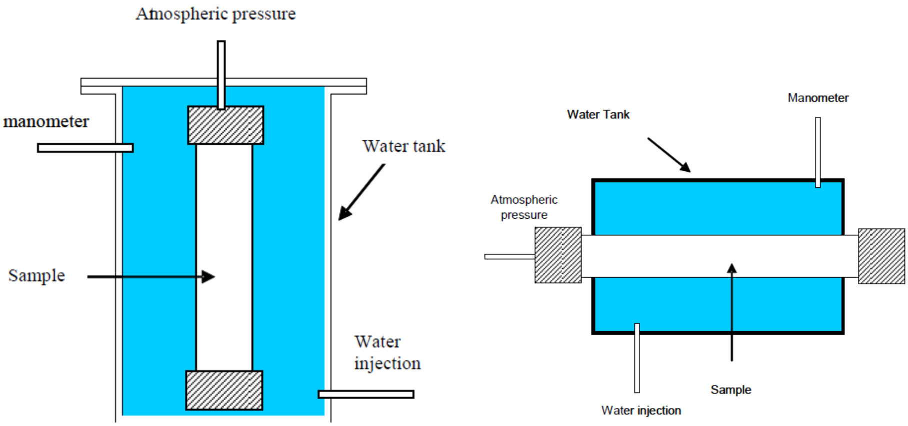


Figure 4.2: Experimental collapse test loading sequence. [54]



(a) Example test setup 1. **(b)** Example test setup 2.

Figure 4.3: Examples of experimental test setups for determining the collapse pressure. [54]

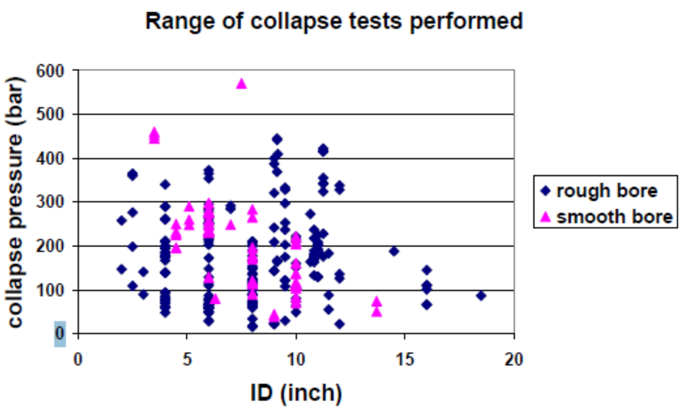


Figure 4.4: Collapse pressure results of experimental tests in a hyperbaric chamber for rough and smooth bore pipes with ranging internal diameter. [54]

Table 4.1: Experimental results of the collapse pressure of 2.5 inch diameter pipes. [55]

2.5 inch sample nr:	Collapse Pressure [MPa]	Fraction Filled
3	12.08	0.79
4	12.41	0.79
5	12.82	0.79
6	13.09	0.79
7	13.18	0.79
8	12.45	0.76
9	15.07	0.75
10	15.50	0.75
Average	13.33	0.78
Standard Deviation	1.27	0.019
Upper Stats Limit	17.14	0.83
Lower Stats Limit	9.25	0.72

Table 4.2: Experimental results of the collapse pressure of 4 inch diameter pipes. [55]

2.5 inch sample nr:	Collapse Pressure [MPa]	Fraction Filled
14	7.07	0.82
15	7.40	0.79
16	7.49	0.82
19	7.67	0.82
20	7.13	0.82
21	7.41	0.82
Average	7.36	0.82
Standard Deviation	0.23	0.012
Upper Stats Limit	8.04	0.85
Lower Stats Limit	6.69	0.78

4.2. Analytical analysis

When regarding the buckling of the carcass layer as a fundamental problem of two concentric rings, where the inner ring is loaded with an external pressure, two kinds failure modes are found. Not surprisingly these were the eight mode and heart mode collapse. In this analysis two types of initial imperfections were considered: single and double symmetric ovalisation. These induced the failure modes, and it was concluded that the ratio of stiffness (thickness) of the two concentric rings and the amount of initial imperfection influenced the failure mode shape in this analytical model. From this research different analytical models for the different failure modes were developed. [41, 51]

4.2.1. Eight mode

The two different collapse modes require different analytical formulation for the critical external pressure. A first approximation of the critical pressure for ovalisation or eight mode collapse for a tubular or a ring was first derived by Timoshenko and Gere in 1961. This method is developed for rigid rings, and since the carcass and pressure armor are wound with a very small pitch (section 5.1.10) they are often idealized as uniform rings. [15] These rings are not flexible but the distance between the rings is variable, and can therefore be approximated by the simplification of rigid rings. They started by using a differential equation for a thin beam with a circular centerline where it is assumed that the radial displacements (u) are small and there are no tangential displacements [71]:

$$\frac{d^2u}{d\phi^2} + u = -\frac{Mr^2}{EI} \tag{4.1}$$

In this equation ϕ is the angular reference for a point on the cross-section of a curved beam from the center of the circular centerline, M is the bending moment in the cross-section of the beam, r is the initial curvature radius of the beam and EI is the bending stiffness of the cross-section of the beam. Using this basic equation two structures can be considered, a 2D ring or a 3D tube having an initial ring ovalization and tube ovalization respectively. Both consider the profiles to be rectangular.

To induce collapse, an elliptical radial ovalization (imperfection) u_1 is introduced, together with an uniform pressure p acting on the external surface. The bending moment in the cross-section becomes:

$$M = pR(u + u_1 cos(2\phi)) \tag{4.2}$$

Ring model

When a rectangular profile is considered equation 4.1 can be rewritten as:

$$\frac{d^2u}{d\phi^2} + u = -\frac{12Mr^2}{Et^3} \tag{4.3}$$

Substituting equation 4.2 in equation 4.3 gives:

$$\frac{d^2u}{d\phi^2} + u = -\frac{12}{Et^3}pR^2(u + u_1cos(2\phi)) \tag{4.4}$$

Solving this equation results in:

$$\frac{u_1p}{p_{cr} - p} \tag{4.5}$$

Where p_{cr} is the critical theoretical pressure load of a ring with no initial ovalization and equals:

$$p_{cr} = \frac{E}{4}\left(\frac{t}{R}\right)^3 \tag{4.6}$$

Tube model

The same method as applied to the ring is used for the tube, however to account for the in plane stresses the Poisson ratio (ν) is included. Equation 4.1 therefore becomes:

$$\frac{d^2u}{d\phi^2} + u = -\frac{12(1-\nu^2)Mr^2}{Et^3} \tag{4.7}$$

Substituting equation 4.2 in equation 4.7 gives:

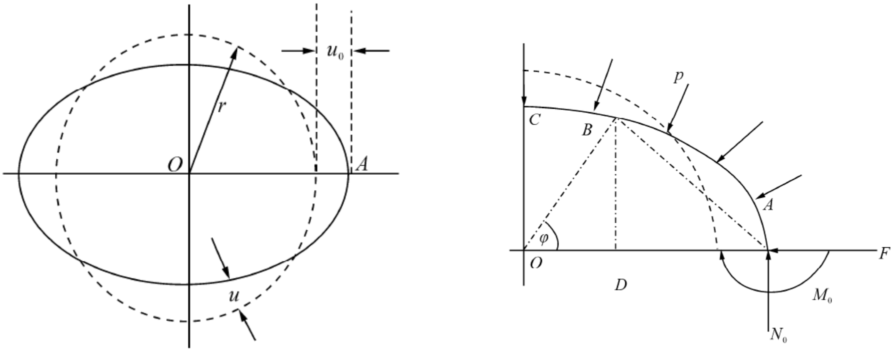
$$\frac{d^2u}{d\phi^2} + u = -\frac{12(1-\nu^2)}{Et^3}pR^2(u + u_1cos(2\phi)) \tag{4.8}$$

where the critical theoretical pressure is found to be [68]:

$$p_{cr} = \frac{E}{4(1-\nu^2)}\left(\frac{t}{R}\right)^3 \tag{4.9}$$

Multi-layer influenced analytical analysis

The analytical method thus far only considers a single circular or tubular structure, e.g. considering only the carcass layer, disregarding the influence of the pressure armor on the collapse resistance. One way to include the influence of the pressure armor is to consider it as a force on the outer surface of the carcass on the longest diagonal, as shown in figure 4.5. This force can be seen as a spring to replicate the increasing force with increasing ovalisation, where the distance u_o represents the compression of the spring. The effect of the pressure armor provides additional protection against buckling [15].



(a) Carcass ring before (dotted line) and after (filled line) deformation.

(b) Quarter section of the carcass ring, with the remaining sections substituted by normal forces and moments. Effect of the pressure armor is added as a force acting on the outer surface of the carcass along the largest diagonal of the ovalized carcass.

Figure 4.5: Analytical model for calculating the collapse pressure of the carcass, also taking into account the effect of the pressure armor and liner. [15]

Calculating the critical pressure for eight mode collapse, including the influence of the pressure armor, for wet collapse can be simplified by considering only a quarter of the cross section as shown in figure 4.5b. The other sections are described with reaction forces, e.g. for the bottom part a force N_0 and moment M_0. The reaction force of the pressure armor is included and represented by a force F. The first step in analytically calculating the collapse pressure is by analyzing the bending moment in any part of section B (indicated in figure 4.5b) as prescribed by:

$$M = M_0 - p \cdot \overrightarrow{OA} \cdot \overrightarrow{DA} + \frac{p}{2} \cdot \overrightarrow{AB}^2 + \frac{F}{2} \cdot \overrightarrow{BD} \tag{4.10}$$

where p is the external hydrostatic pressure. The moment M is positive when it increases the initial curvature. When considering triangle OAB, the section OB can be geometrically rewritten as:

$$\overrightarrow{OB}^2 = \overrightarrow{AB}^2 + \overrightarrow{OA}^2 - 2\overrightarrow{AB} \cdot \overrightarrow{OA} \cdot \frac{\overrightarrow{DA}}{\overrightarrow{AB}} \tag{4.11}$$

which can be rewritten into:

$$\frac{1}{2}\overrightarrow{AB}^2 - \overrightarrow{OA} \cdot \overrightarrow{DA} = \frac{1}{2}(\overrightarrow{OB}^2 - \overrightarrow{OA}^2) \tag{4.12}$$

or:

$$\frac{1}{2}\overrightarrow{AB}^2 - \overrightarrow{OA} \cdot \overrightarrow{DA} = \frac{1}{2}[(r+u)^2 - (r+u_0)^2] \tag{4.13}$$

where u denotes the radial displacement of a section with a positive value when the section moves towards the center of the ring. Equation 4.13 can be substituted into equation 4.10, and assuming that the displacements u are very small compared to the radius, such that the squares of u and u_0 can be neglected, the bending moment is approximated by:

$$M = M_0 + pr(u - u_0) + M_F \tag{4.14}$$

M_F is the moment generated by force F:

$$M_F = \frac{F(r-u)sin(\phi)}{2} \tag{4.15}$$

Where ϕ is the angle between OA and the radial vector of section B, see figure 4.5b. M_F can be linearly approximated by:

$$M_F = M_{FA} + \frac{M_{FC} - M_{FA}}{u_c - u_0} \cdot (u - u_0) \tag{4.16}$$

where:

$$M_F A = (1-\alpha)rF/\pi \tag{4.17}$$

and

$$M_F C = M_F A - rF/2 \tag{4.18}$$

represents the bending moments from force F in sections A and C respectively. α equals the hoop axial stress deformation factor given by:

$$\alpha = \frac{I}{Ar^2} \tag{4.19}$$

I is the area moment of inertia of the ring and A is the area of the cross-section of the ring profile. [75] The contact pressure of two elastic bodies is proportional to its radial displacement according to Hooke's law. Equation 4.16 can be rewritten as:

4

$$M_F = M_FA - \frac{1}{2}\frac{F}{u_C - u_0}r(u - u_0) = M_FA - \frac{1}{4}kr(u - u_0) \tag{4.20}$$

Here k is a constant, analogous to the modulus of a material. Substituting equation 4.20 into 4.14 further into 4.1, and performing some algebraic operations it results in:

$$\frac{d^2u}{d\phi^2} + u(1 + \frac{(p - \frac{1}{4}k)r^3}{EI}) = \frac{(M_0 - M_{FA})r^2 + (p - \frac{1}{4}k)r^3u_0}{EI} \tag{4.21}$$

The general solution of which is:

$$u = C_1sin(q\phi) + C_2cos(q\phi) + \frac{(M_0 - M_{FA})r^2 + (p - \frac{1}{4}k)r^3u_0)}{EI + (p - \frac{1}{4}k)r^3} \tag{4.22}$$

Where C_1 and C_2 are constants and:

$$q^2 = 1 + (p - \frac{1}{4}k)\frac{r^3}{EI} \tag{4.23}$$

From symmetry at the sections A and C, it follows that:

$$(\frac{du}{d\phi})_{\phi=0} = 0 \tag{4.24}$$

and

$$(\frac{du}{d\phi})_{\pi=\frac{\pi}{2}} = 0 \tag{4.25}$$

By applying equations 4.24 and 4.25 to 4.22 the critical pressure P_{cr} is obtained:

$$p_{cr} = \frac{3EI}{r^3} + \frac{k}{4} \tag{4.26}$$

It can be seen from equation 4.26 that the pressure armor increases the wet collapse pressure. Next the interaction between the carcass and inner liner can be included. When the hydrostatic pressure p acts on the outer surface of the liner, the contact pressure p' on the carcass surface is given by:

$$p' = \frac{2p(1-{\mu_L}^2)}{\beta E_L} \tag{4.27}$$

where the subscript L denotes the barrier layer. The coefficient β is defined by:

$$\beta = \frac{(1+\mu_L)(1-2\mu_L)}{E_L}\left(\frac{1-t_L/r_L/2}{1+t_L/r_L/2}\right)^2 + \frac{1+\mu_L}{E_L} - \frac{1}{E}\left(\frac{r}{t}+\frac{1}{2}+\mu\right)\left[1-\left(\frac{1-t_L/r_L/2}{1+t_L/r_L/2}\right)^2\right] \tag{4.28}$$

or

$$\beta = \frac{(1+\mu_L)(1-2\mu_L)}{E_L}\left(1-\frac{t_L}{2r_L}\right)^2 + \frac{1+\mu_L}{E_L} - \frac{1}{E}\left(\frac{r}{t}+\frac{1}{2}+\mu\right)\left[1-\left(1-\frac{t_L}{rr_L}\right)^2\right] \tag{4.29}$$

where t is the thickness of the carcass cross-section and equation 4.29 is more conservative. Combining equations 4.26 and 4.27 results in the modified critical pressure of the carcass:

$$p_{cr} = \frac{\beta E_L}{2(1-{\mu_L}^2)}\frac{3EI}{r^3} + \frac{k}{4} \tag{4.30}$$

When the annulus of the flexible riser is flooded, equation 4.30 can be further rewritten to:

$$p_{cr} = \frac{\beta E_L}{2(1-\mu_L^2)}\frac{fE}{4}\left(\frac{t}{r}\right)^3 + \frac{k}{4} \tag{4.31}$$

where f indicates the fraction fill of the carcass, and is given by:

$$f = \frac{s}{l_p t} \tag{4.32}$$

This formulation is based on the equivalence of area of strip profile and formed cross-section, where s is the cross-section area and l_p the pitch.

The pressure armor and liner can be modeled as two springs connected in series. The ovalization of the pressure armor can be approximated with a sinusoidal function [58]:

4

$$u = u_L cos(2\phi) \tag{4.33}$$

where u_L is the maximum displacement. Substituting equation 4.33 into equation 4.3 results in the following change of curvature in a ring:

$$\kappa = \frac{M}{EI} = \frac{3u_L cos(2\phi)}{r^2} \tag{4.34}$$

Such that the potential energy is given by:

$$E_{pot} = \frac{1}{2} r \int_0^{2\pi} EI\kappa^2 d\phi - 2Fu_L = \frac{9\pi EI u_L^2}{2r^3} - 2Fu_L \tag{4.35}$$

Where the principle of minimum of potential energy can be used to show that:

$$F = \frac{9\pi EI u_L}{4r^3} \tag{4.36}$$

For the pressure armor (subscript pa) it therefore follows that:

$$k_{pa} = \frac{9\pi E_{pa} I_{pa}}{4r_{pa}^3} \tag{4.37}$$

and for the liner:

$$k_L = \frac{9\pi}{4} \frac{E_L I_L}{(1-\mu_L^2) r_L^3} / \left(\frac{2(1-\mu_L^2)}{\beta E_L} - 1 \right) \tag{4.38}$$

Both of which can be rewritten:

$$k_{pa} = \frac{3\pi f_{pa} E_{pa}}{16} \left(\frac{t_{pa}}{r_{pa}}\right)^3 \tag{4.39}$$

$$k_L = \frac{3\pi E_L}{16(1-\mu_L^2)} \left(\frac{t_L}{r_L}\right)^3 / \left(\frac{2(1-\mu_L^2)}{\beta E_L} - 1\right) \tag{4.40}$$

The improved critical carcass collapse pressure that includes the support of the pressure armor in resisting collapse is now given by:

$$p_{cr} = \frac{\beta E_L}{2(1-\mu_L^2)} \frac{fE}{4} \left(\frac{t}{r}\right)^3 + \frac{k_{pa} k_L}{4(k_{pa}+k_L)} \tag{4.41}$$

4

The parameters without a subscript are related to the carcass. And in general the liner reduces the support of the pressure armor when $k_L < k_{pa}$. [15]

4.2.2. Heart Mode

When the carcass is encased by a pressure armor with a much higher stiffness ratio, it can be modeled as a rigid encasement. This results in a heart mode collapse mode as ovalization is impossible. When assuming the hydrostatic buckling of an infinitely long elastic cylinder confined in a rigid enclosure without friction in between the following expression for the critical pressure had been found[9, 28]:

$$p_{cr} = \frac{E}{(1-\nu^2)} \left(\frac{t}{D}\right)^{\frac{11}{5}} \tag{4.42}$$

Where t is the thickness of the cylinder and D the mean diameter. Similarly for a ring this equation can be again modified by disregarding the Poisson ratio[9, 28]:

$$p_{cr} = E \left(\frac{t}{D}\right)^{\frac{11}{5}} \tag{4.43}$$

These equations have also been verified using finite element methods [60]. And have been modified and optimized for different problems and situations where for example geometrical imperfections and an initial gap were included in the analysis [12, 39–41, 66]. When the outer ring is assumed to be linear elastic instead of rigid a nonlinear response is found with a limited load of instability [41].

4.2.3. Equivalent thickness

To more accurately evaluate the buckling collapse of the carcass or pressure armor requires, a rectangular cross-section can no longer be assumed, but the actual cross-section of the carcass or pressure armor should be analyzed. It is also possible to calculate a equivalent thickness of an alternative cross-section. This means that instead of including the exact cross-section of the structure of the carcass, a rectangular cross-section with a certain thickness is assumed such that it has similar bending properties for example. The carcass and the pressure armor layers have complex cross sections (not rectangular) and may present a more complex bending behavior than a rectangular cross section, this method therefore is only an approximation. This equivalent thickness can be estimated by either analytical or experimental methods. Five methods for calculating an equivalent thickness are used.

The first technique for calculating the equivalent thickness for the carcass layer is by making the real carcass profile and a rectangular cross-section have similar bending stiffness per unit length (ei). This method is also referred to as the bending stiffness equivalence method. Once the bending stiffness per unit length is determined for the real carcass profile, the required thickness for a rectangular cross-section can be calculated to match this bending stiffness per unit length. The bending stiffness per unit length for the real carcass profile is [68]:

$$ei = \frac{2EI_{Gmin}}{b} \tag{4.44}$$

$$ei = \frac{2EI_{Gmin}}{b(1-\nu^2)} \tag{4.45}$$

Where E is the Young Modulus, b is the length of the carcass cross section profile and l_{Gmin} is the minimum moment of inertia of the carcass cross profile, which equals:

$$I_{Gmin} = \frac{bt_(eq)^3}{24} \tag{4.46}$$

Combining this with equations 4.44 and 4.45 yields the following bending stiffness for the ring and tube models respectively:

$$ei = \frac{Et_{eq}^3}{12} \tag{4.47}$$

$$ei = \frac{Et_{eq}^3}{(1-\nu^2)12} \tag{4.48}$$

Some modifications to this expression have been done to account for the interlocking superposition of the carcass profile. At first the bending stiffness per unit length of the carcass-profile was multiplied by a factor of 2 to account for this super positioning [46], but the effect of superposition is dependent on the carcass profile. Therefore this value can be anywhere between 1 or 2, representing no superposition and total superposition of the carcass profile respectively. [50] Equations 4.44 and 4.45 can therefore be modified to:

$$ei = E\frac{(1+\varphi)I_{Gmin}}{b} \tag{4.49}$$

$$ei = E\frac{(1+\varphi)I_{Gmin}}{b(1-\nu^2)} \tag{4.50}$$

Where ϕ represents this superposition factor. The value of ϕ depends on the profile geometry and the pitch of the carcass-profile (affecting the superposition of the profiles). The assumed bending stiffness per unit length thus increases with Phi. A simple method to determine the degree of super-positioning is to determine super relative to the pitch of the profile, shown in figure 4.6:

$$\varphi = \frac{super}{pitch} \tag{4.51}$$

Equalizing equations 4.44 and 4.47 (equalizing equations 4.45 and 4.48 has te same result), and rearranging to extract the equivalent thickness results in:

$$t_{eq} = \sqrt[3]{\frac{12(1+\varphi)I_{Gmin}}{b}}. \tag{4.52}$$

The second approach for calculating the equivalent thickness is to consider the carcass as a tube, where the flexional stiffness of the tube is adjusted to match that of the real carcass profile. Again the minimum moment of inertia and same method is used but instead of normalizing this to a unit length it is normalized to a unit area ($A_{profile}$). This method is therefore also referred to as the area equivalence method or equivalent sectional bending stiffness, and this results in the following equivalent thickness[64]:

$$t_{eq} = \sqrt{\frac{12I_{Gmin}}{A_{profile}}} \tag{4.53}$$

The third approach uses a simplified cross sections, with an equivalent cross

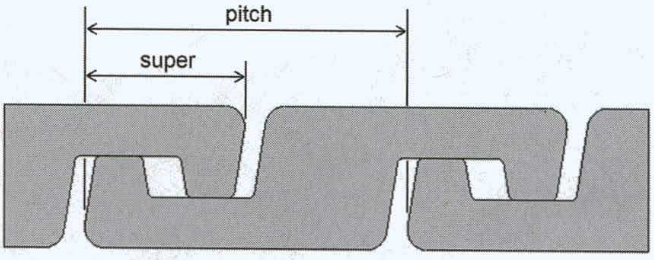


Fig. 4 Pressure armor cross section profile

Figure 4.6: [49]

section area. This means that only the cross section area is equal to that of the real profile but simplified to a rectangular, making the bending stiffness different from the real profile. that is This is therefore a very basic and simplified estimation for calculating the collapse pressure. This method will also not further be analyzed as it is not able to accurately represent the complex structure of a carcass profile. [43, 46, 49]

Two more equivalent thicknesses can be subtracted from the research into numerical-analytical prediction of collapse of flexible pipes. These are the numerically determined equivalents of equation 4.52 and the above explained third equivalent thickness approach. [20, 72]:

$$t_{eq}^3 = \frac{36 I_x n^2 (1-\nu^2)^2 tan(\alpha)^2}{h^3 \pi^2 R^2} \tag{4.54}$$

$$t_{eq} = \frac{nAx}{L_p} \tag{4.55}$$

Where n is the number of carcass strips, I_x the transverse Inertia of the profile, α the laying angle, h the pipe thickness and L_p the pitch of the carcass equal to:

$$L_p = \frac{2\pi R}{tan\alpha} \tag{4.56}$$

For calculating the buckling pressure, it is safer to use the ring model over the tube model as it is more conservative when it comes to calculating the equivalent thickness. Ring behavior results in smaller critical pressures compared to tube behavior. To calculate the critical pressure, the found equivalent thickness can be inserted into equation 4.6 or 4.9. It is also possible to check if plastic stress would occur by calculating the maximum compressive stress due to bending stress in the profile cross-section induced by external pressure. This stress is given by [50]:

$$\sigma_{max} = \frac{qR(w+w_1)\frac{t}{2}}{\frac{(1+\phi)I_{Gmin}}{b}} + \frac{qR}{t} \tag{4.57}$$

This work was the basis for many other research in the buckling analysis of pipes. Depending on the subject being analyses, modifications were made to the original work of Timoshenko and Gere. Some of the most noteworthy changes are the including of corrosion effects by changing the effective thickness of the pipe and the addition of anisotropy effects [7, 8, 15, 18, 50, 51]

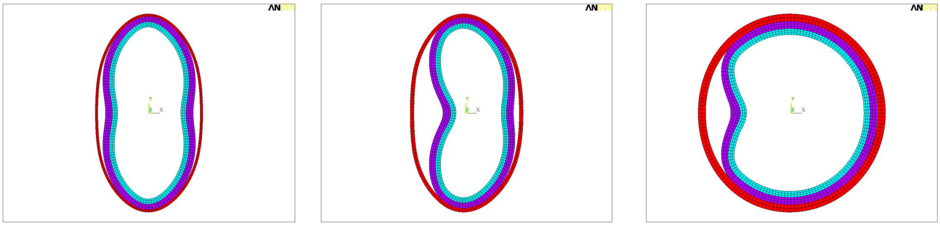
4.3. Finite Element methods

The analytical analysis of the buckling of pipes has evolved throughout the years, becoming more representative to the real world buckling capacity of risers. However, the complexity and nonlinearity of the interaction between the layers of flexible risers in the (ultra-)deep water environments is not yet fully understood, and results in difficulties and constraints that makes it very hard to accurately represent it with an analytical model. Most studies into the buckling collapse of flexible risers therefore use finite element methods or a combination of FE and analytical analysis.

FE analysis has the ability to analyze multiple layers with different properties and their interactions characterizing unbonded flexible risers, something that can have a significant influence in the buckling behavior of flexible risers. FE models often include only two or three layers of the flexible riser to reduce the calculative intensity, and these layers are: the carcass, internal polymeric layer or liner and pressure armor. The latter is sometimes excluded to examine the collapse properties of the carcass as an individual component where the influence of the pressure armor is excluded.

FE methods are also able to predict the local stress in the carcass, as seen in figures 2.1 and 4.8, where stress concentration factors are present if the carcass is properly modeled. This is not possible with analytic methods as the carcass profile has to be simplified in order to be analytically feasible to solve. Furthermore, a well-defined model is capable of producing both eight as well as heath mode collapse, as seen in figure 4.7. This means that factors and parameters influencing the collapse mode shape can be altered within a single model to investigate the sensitivity of the collapse mode to these factors. [45]

FE models require an initial imperfection that initiates buckling, just like with an analytical analysis. If no imperfection is present an infinite stiffness within the model can occur. This leads to highly overestimated buckling collapse pressures. When the value of ovalization is very small, structures only have to display a small displacement before sudden loss of stability occurs. When using FE this results in both very small and high values in the stiffness matrix which results in computational difficulties and errors which are a function of the eigenvalues of the stiffness matrix [30].



(a) Eight mode **(b)** Transitional mode **(c)** Heart mode

Figure 4.7: Results from a single finite element model able to produce multiple collapse modes [45]

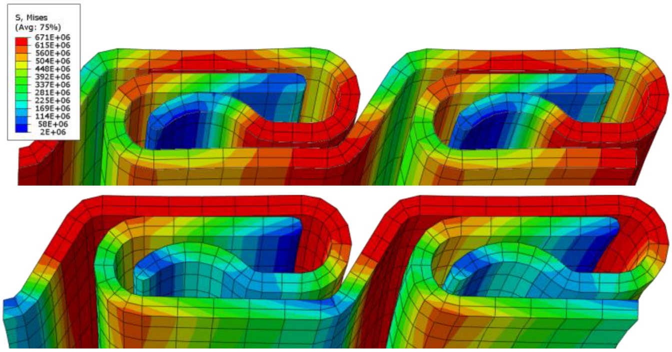


Figure 4.8: Carcass equivalent stress at collapse. The top part shows the profile stress at the maximum diameter of ovalization, the bottom part at the minimum diameter. [18]

Furthermore, the amount of initial imperfection influences the convergence speed, and with it the computation time. Choosing a good initial imperfection is one of the first steps in the setup of the model. The most commonly used imperfection is an ovalization, imposed on the carcass, carcass and liner or all the layers. Sometimes an indentation is simulated by moving one of the nodes slightly closer to the centerline of the tube or circle. [45]

Including the exact carcass profile in a FE analysis can be very computationally demanding, and whether it is worth it depends on the application. It is also possible to use a simplified carcass or pressure armor profile, using a method similar to the analytical method where a rectangular profile cross-section is used with an equivalent thickness, this is also known as an equivalent cross section profile. The advantage FE analysis with an equivalent cross section profile has over an analytical analysis is that the interaction between different layers can be more accurately represented and analyzed. The friction coefficient between the layers of the riser often equals 0.1 in FE analyses, based on experimental work [59].

The results for the buckling collapse pressure can be quite similar when the equivalent thickness method is implemented correctly, as can be seen in figures 4.9 to 4.11. [49] In these simulations two models were compared, the first model

(Cases A1 to A5) is a 3D model where the interlocked carcass, internal polymeric layer (liner) and pressure armor are modeled with their actual cross section. The second model (cases B1 to B5) is a simplification of the first model where the pressure armor is modeled using an equivalent cross section profile. The model only considers a quarter of a ring to study the effect of this simplification on double symmetric eight mode buckling, and also studies the influence of the initial ovalization and gap width (more information of these influencing factors will be detailed in chapter 5). The models were compared using a commonly used dimensionless loading parameter λ for an uniform load which equals [46]:

$$\lambda = \frac{qr^3}{ei} \tag{4.58}$$

Another commonly used simplification in the modeling of the carcass and pressure armor layers is the exclusion of the lay angle, meaning a ring model or a series of ring models is used. The carcass and pressure armor layer are conformed to a helical shape in reality, which does not present any symmetry. In FE analysis symmetry can be a very useful property to significantly reduce the calculating effort by making the boundary conditions coupled. This way infinite repetitions can be simulated without actually rendering an infinitely long pipe. By coupling the degrees of freedom (DOFs) of the opposite sides of a cut from a 3D pipe or ring model, the same displacement field occurs on both of these sides. [50]

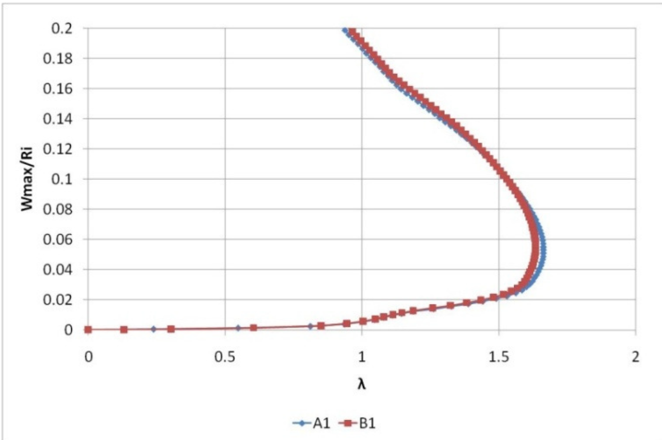


Figure 4.9: Comparison between the results of a 3D FE model incorporating the actual cross-sections of the carcass, liner and pressure armor (A1) and an simplified model where the pressure armor is modeled using an equivalent cross section profile (B1). Both have an initial ovalization of 0.5% [49]

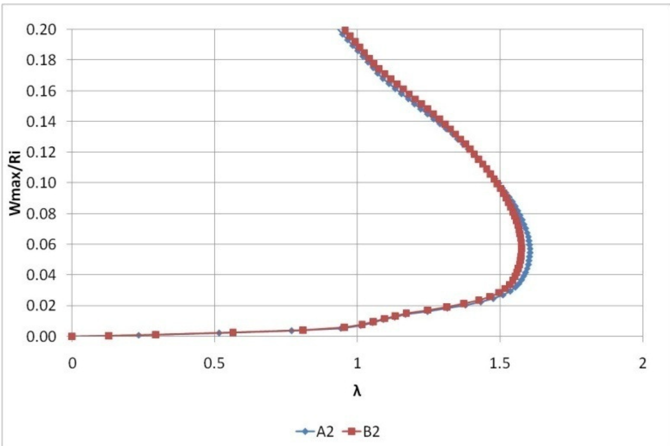


Figure 4.10: Comparison between the results of a 3D FE model incorporating the actual cross-sections of the carcass, liner and pressure armor (A2) and an simplified model where the pressure armor is modeled using an equivalent cross section profile (B2). Both have an initial ovalization of 1% [49]

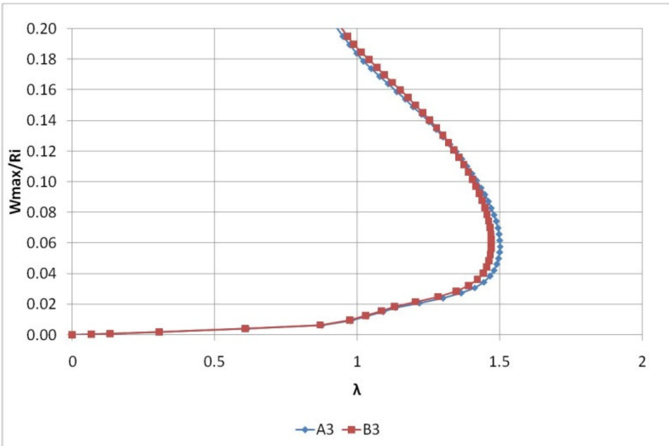


Figure 4.11: Comparison between the results of a 3D FE model incorporating the actual cross-sections of the carcass, liner and pressure armor (A3) and an simplified model where the pressure armor is modeled using an equivalent cross section profile (B3). Both have an initial ovalization of 2% [49]

4.4. Conclusion

Three methods for analyzing the flexible riser are used. The first and most accurate is experimental testing. The major disadvantage of which are the high costs, especially as most flexible risers are custom build to suit the application and location. This would mean that each purpose build riser would require extensive experimental testing before being approved by the customer. With it comes that the collapse strength is only found after designing and manufacturing the riser This means that when the collapse strength is found to be insufficient, the design, manufacturing and testing process has to be redone.

The second is analytical analysis, where different mathematical models are used for eight and heart mode collapse. Analytical models used for determining the critical pressure rely on some simplifications of the complex structure, profiles and layer composition of flexible risers. The most significant is the idealization of the carcass and pressure armor as simple ring or tubular, where the tubular includes the Poisson ratio. Due to the mathematical complexity of multi-layered flexible risers, an simplified model that can be used is that of two concentric rings representing the carcass and pressure armor, where the inner ring is confined by the outer ring while an external pressure acts on the outer surface of the inner ring. Im such a model, the outer ring is represented as a force acing inward on the inner ring along the largest diagonal, where the inner ring has an ovalization. It is also common to simplify te carcass and pressure armor profiles, where an equivalent thickness is calculated. This equivalent thickness is the thickness of a rectangular profile such that it matches a bending or buckling property of the real carcass profile, mostly the bending stiffness per unit length.

The third and most used method for collapse analysis of flexible risers is numerical, or FE analysis. FE analysis has the ability to analyze multiple layers with different properties and their interactions characterizing unbonded flexible risers, something that can have a significant influence in the buckling behavior of flexible risers. FE models often include only two or three layers of the flexible riser, namely: the carcass, internal polymeric layer and pressure armor. FE methods are also able to predict the local stress in the carcass. These are some of the advantages FE models have over analytical models, while also being superior to experimental methods as they are cheaper and more time-efficient as multiple configurations and materials can be modeled, also during the design process.

5

Factors influencing collapse

Flexible risers are complex structures consisting of many different and specifically tasked layers. The unbonded nature allows for relative movement between the layers resulting in some very complex interactions between them. Many different factors influencing the collapse pressure were found, the most mentionable are:

- Curvature
- Gap width
- Ovalization and geometric imperfections
- Polymeric layer (liner) thickness
- D/t ratio
- Material properties
- Axial load
- Pressure armor strength (thickness)
- Carcass profile
- Pitch and lay angle of the carcass and pressure armor
- Friction of layers
- Initial interference between liner and pressure armor
- Mass flow rate

Beside influencing the collapse pressure, some of these factors also influence the collapse mode.

5.1. Collapse Pressure

The collapse pressure is one of the main factors limiting the maximum operating depth for flexible risers. Large safety factor are used to cope with the lack of understanding or ability to simulate or analyze all of the factors influencing the collapse pressure.

5.1.1. Curvature

Flexible risers are often suspended from a floating platform, where the lower end is attached to the seabed. The necessity of relative motion between the seabed and floating platform requires the riser to have a catenary shape, meaning the riser is curved to allow for this relative movement.

Unlike steel risers, the curvature of flexible risers does not induce high stresses in the carcass or pressure armor as the material is not stretched, as long as the bending radius is large enough to prevent the locking of the profiles. The interlocked sections of the carcass and pressure armor are free to move in axial direction until locking. [4]

Bending the riser induces an imperfection with one symmetry plane. This results in reduced symmetry in two ways, the first is in the concentric loading in the plane of curvature and the second is in the cross section of the riser. The bending of the riser changes the surface area of the outside of the riser, where the extended side has a larger area and the compressed side a smaller area. This causes the extended side to have more force acting upon the pressure layer and carcass than the compressed side, this is visualized in figure 5.1. [52, 54]

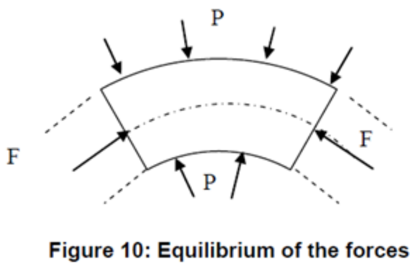


Figure 5.1: Equilibrium of forces on a curved pipe section. [54]

The effect of curvature on the cross section of the riser is the decrease of pitch of each of the rolled layers in the compressed side (region 1 in figure 5.1) and increase of the pitch in the extended side (region 2 in figure 5.1). This causes a radial stiffness change where the radial stiffness in the compressed side is increased as the carcass and pressure layer wounds are closer together and therefore have a larger radial stiffness per unit length as there are more windings per unit length. The opposite effect occurs at the extended side where the radial stiffness is therefore reduced.

Both of these asymmetries make the extended side more susceptible for collapse. The collaboration of both the loading and cross sectional asymmetry re-

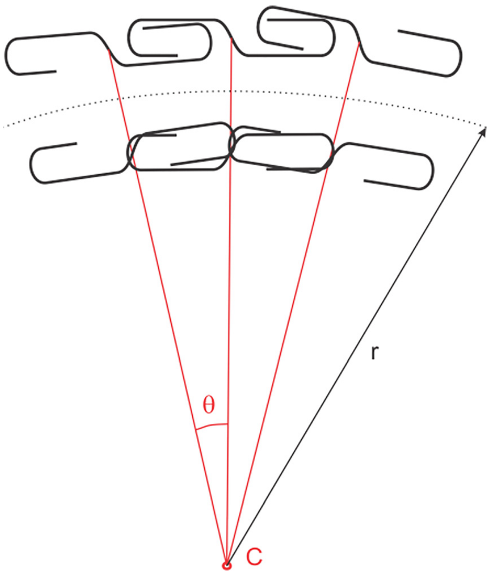


Figure 5.2: Center of curvature of the approximated collapse models, and the effect on the carcass cross-section for the inner and outermost sections. (exaggerated curvature—small r) [52]

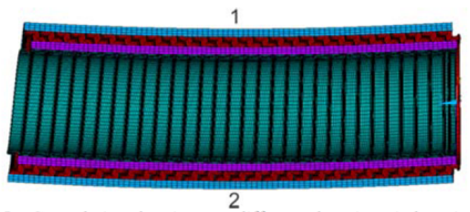


Figure 5.3: Curved pipe with an asymmetric pitch in the two different regions 1 and 2. [51]

duces the wet pressure collapse. The less stiff extended section will collapse first and therefore initiate heart mode collapse, this is also shown in figures 5.4 and 5.5, where the riser was first bent to the MBR before increasing the pressure. This study on bending sensitivity was done for the 8 inch riser with a gap of 1.4 mm, as this was considered the most representative riser configuration. [4, 51]

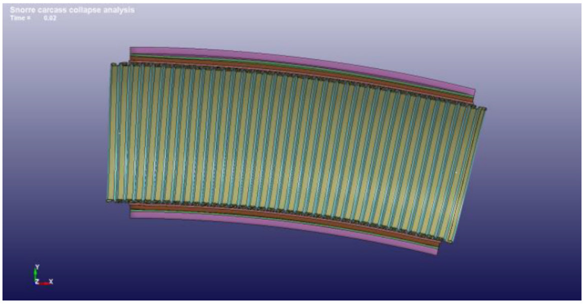


Figure 5.4: Simulation of a 8 inch riser bent to MBR. [4]

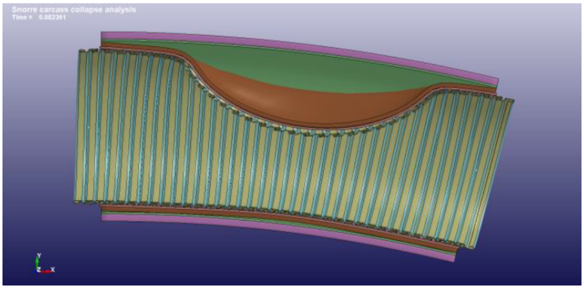


Figure 5.5: Simulation of the collapse of a bent 8 inch riser. [4]

Bending a flexible riser greatly reduces the wet collapse pressure compared to a straight riser. Finite element methods show that reductions of more than 10% possible, depending on the diameter and bending radius of the riser, see figures 5.6 and 5.7. An Finite Element analysis of an 8 inch riser being bent to MBR before increasing the pressure showed an decrease in collapse pressure of 24% compared to the straight model. This result is also shown in figure 5.8. The influence of curvature on the collapse resistance is even more predominant in smooth bore pipes compared to rough bore pipes, and has to be regarded as one of the major influencing factors on the critical collapse pressure.[4, 51, 54]

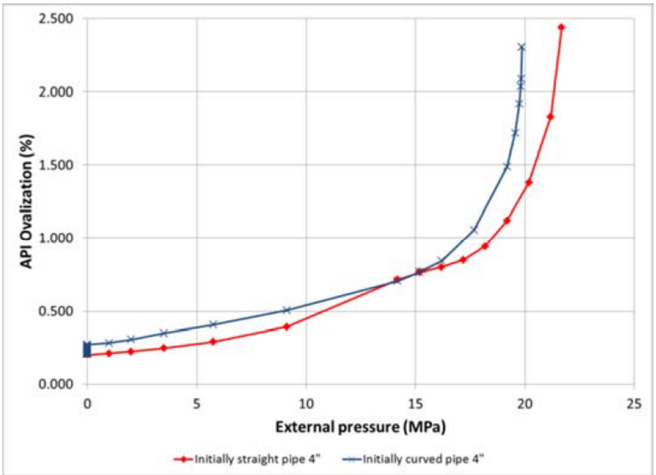


Figure 5.6: API ovalization vs External pressure comparison between a straight and initially bent 4 inch flexible riser. [51]

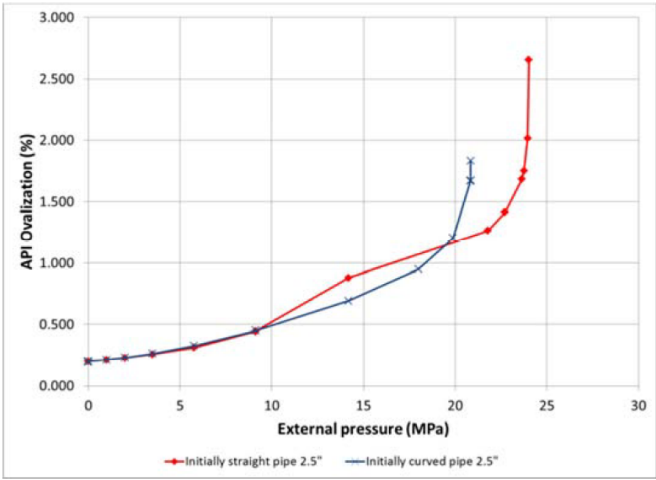


Figure 5.7: API ovalization vs External pressure comparison between a straight and initially bent 2.5 inch flexible riser. [51]

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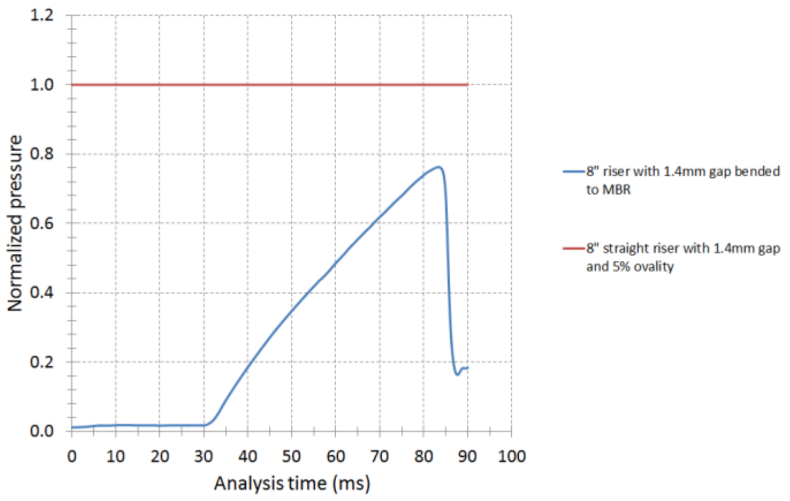

Figure 5.8: Analysis time vs. normalized pressure, bended to MBR 8” riser. [4]

5.1.2. Gap Width

Flexible risers composed of different unbonded layers are sometimes susceptible for gaps between these layers, illustrated in figure 5.9. There are two ways in which a radial gap between the pressure armor, liner and/or carcass can occur: volume change of the polymer liner and extrusion of the liner into the adjacent interlocked layers. Many polymers will either swell or experience a volume loss when subjected to working conditions. [4]

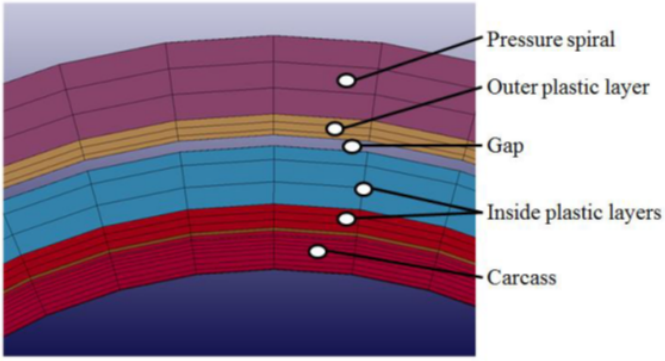

Figure 5.9: Illustration of the layers modeled in a FE model including a gap between two of the layers. [4]

The effect of the gap width can be studied by imposing an initial gap between the liner and pressure armor in a FE analysis. This process can be repeated for different gap widths to study the effect of the gap width size on the wet collapse pressure. A gap width induces a smaller initial radial stiffness of the riser as there is no direct support from the pressure armor preventing the deformation of the carcass. Inducing a gap width therefore reduces the collapse resistance as the carcass is free to deform until the gap is filled. When a gap exists only on a part of the circumference, an ovalization of the carcass can be induced. [49]

Studies into the effect of the size of the gap width have been done, some results are shown in figures 5.10, 5.11 and 5.12. These FE studies were done for risers with different diameters where the gap width was varied. The data of figures 5.10 and 5.11 was also normalized to the results of models with a gap width of 1.4 mm, and the normalized collapse pressures of these are shown in tables 5.1 and 5.2 respectively.

These results all show a decrease of the collapse pressure with increasing gap width. These studies therefore concluded that the critical pressure decreases significantly with increasing gap width. This is to be expected as a larger gap width allows for more deformation of the carcass before being supported by the pressure armor. This addition of a gap decreases the strength (or capacity) of the riser as the carcass is no longer (fully) supported by the pressure armor in resisting external pressure. [4]

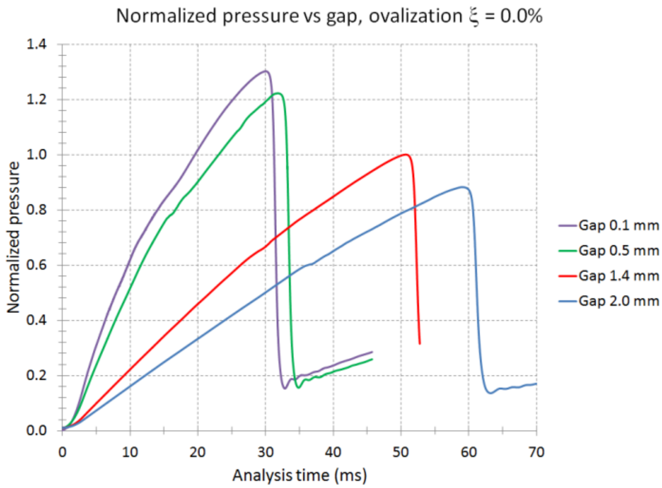

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Figure 5.10: Analysis time vs. normalized pressure, gap sensitivities on an 8 inch flexible riser with no ovalization. The pressure is normalized to the results where the gap width equals 1.4 mm. No analysis of a gap width of 0 mm was done, as it was assumed the gap width reduces the collapse pressure. The focus of this analysis was to investigate the influence of the gap width size. [4]

Table 5.1: Influence of the gap width on the collapse pressure of an 8 inch flexible riser, normalized to the model with no ovalization and a gap with of 1.4mm. [4]

Ovalization (%)	0.0	0.0	0.0	0.0
Gap Size (mm)	0.1	0.5	1.4	2.0
Normalized collapse pressure	1.302	1.222	1.0	0.882

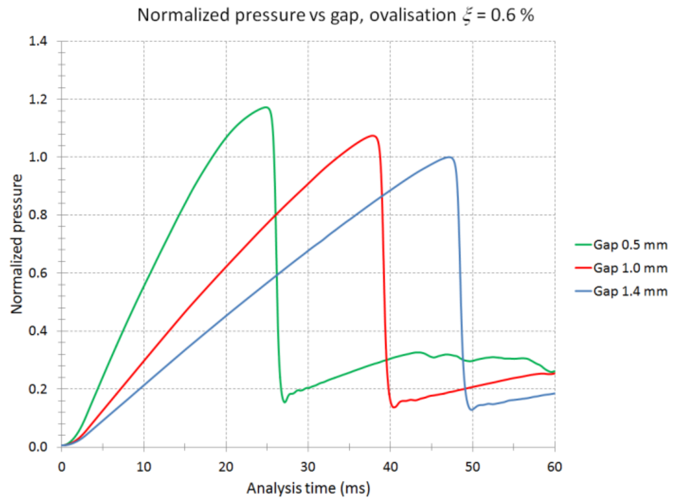

Figure 5.11: Analysis time vs. normalized pressure, gap sensitivities on an 6 inch flexible riser with 0.6% ovalization. [4]

Table 5.2: Influence of the gap width on the collapse pressure of an 6 inch flexible riser, normalized to the model with 0.6% ovalization and a gap width of 1.4mm. [4]

Ovalization (%)	0.6	0.6	0.6
Gap Size (mm)	0.5	1.0	1.4
Normalized collapse pressure	1.172	1.074	1.0

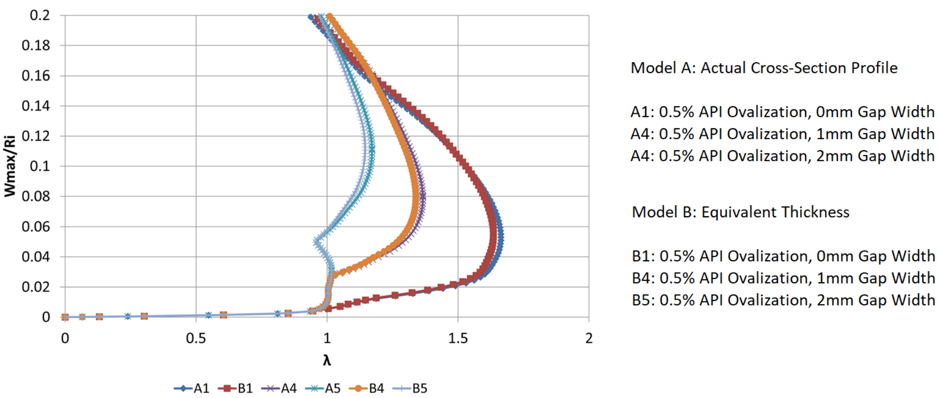

Figure 5.12: Results of a FE analysis of a flexible riser using two different models, one with the actual cross section and one using equivalent thickness. Both models included a 0.5% API Ovalization and the gap width was varied to 0, 1 and 2mm. [49]

5.1.3. Ovalization and geometric Imperfections

The maximum amount of ovalization or imperfectness in the circularity of the cross section is prescribed by all the standards for flexible risers. API prescribes a maximum ovalization of 0.2%. The effect of ovalization is simulated with FE analysis methods where it is it is either applied to the carcass, liner and pressure armor or only the carcass and liner. [4] In the case of the latter this will result in a gap, ovalization and gap width are therefore closely related.

The ovalization of the carcass layer results in contact forces of the pressure armor which tries to prevent the carcass from collapsing. The amount of ovalization that is allowed before these contact forces become significant can determine the wet collapse pressure as well as the shape of the collapse mode.[50, 52, 68]

The ovality of the pressure armor or carcass have a different amount of influence on the wet collapse pressure. The carcass has a slender profile compared to the pressure armor, and is therefore more susceptible to deformations. This also means that the ovalisation of the pressure armor requires more energy as the bending stiffness of the pressure armor profile is much higher. The effect of an fixed percentage of ovalization of the pressure armor therefore does not compare to the same percentage of ovalization of the carcass. The effect of ovalization of the pressure armor is of much larger influence then that of the carcass, this can also be seen by comparing figures 5.13 and 5.14. [18, 50]

From figures 5.13 to 5.15 it is clear that ovalization influences the collapse pressure quite significantly. The collapse pressure results of 5.15 were normalized to the 0% ovalization results and are shown in table 5.3. The normalized collapse pressures show that an ovalization of 0.6% decreases the collapse pressure by 5% for an 6 inch flexible riser compared to a perfectly circular flexible riser. The same effect was observed for 8 and 9 inch risers using the same method.[4]

These results clearly illustrate the importance of producing flexible risers with a very small ovalization, as well as handling the risers such that defects and ovalization are prevented as much as possible. The API standard for a maximum ovalization of 0.2% is therefore a very reasonable standard for ensuring the critical collapse pressure is not dominated by ovalization.

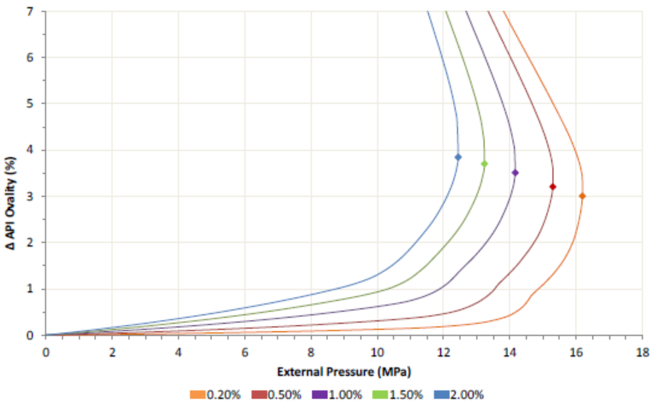

Figure 5.13: Influence of the ovalization of the carcass on the collapse pressure. [18]

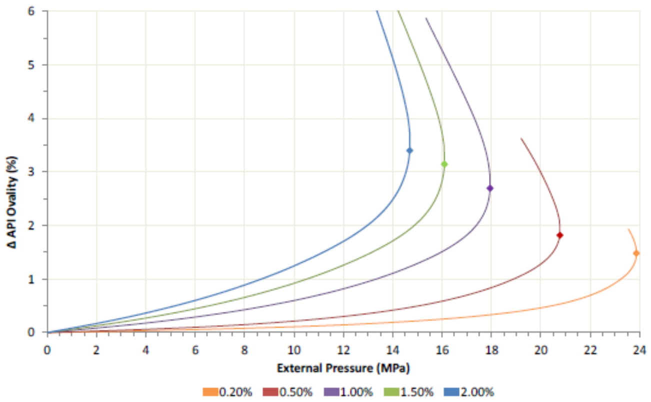

Figure 5.14: Influence of the ovalization of the pressure armor on the collapse pressure. [18]

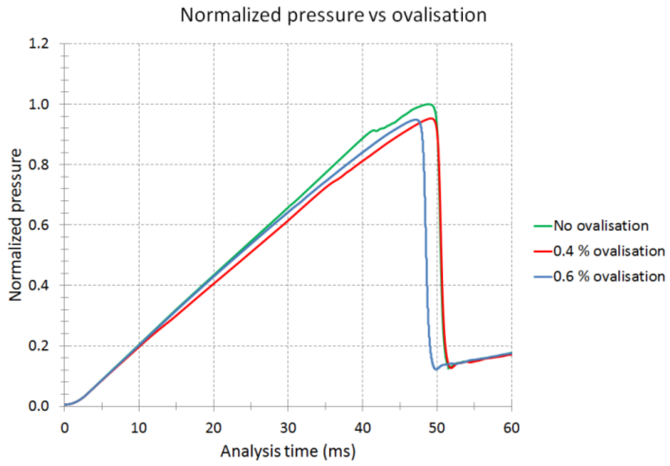

Figure 5.15: Influence of the ovalization of the carcass on the collapse pressure. [4]

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Table 5.3: Influence of the ovalization on the collapse pressure of an 6 inch flexible riser, normalized to the model with no ovalization and a gap width of 1.4 mm. [4]

Ovalization (%)	0.0	0.4	0.6
Gap Size (mm)	1.4	1.4	1.4
Normalized collapse pressure	1.0	0.953	0.949

5.1.4. Polymeric layer thickness

The influence of the polymeric liner depends on the material used and the thickness of this layer. When an easily compressible liner is used it could have an similar effect to that of a gap. A commonly used liner material is Nylon 11 or Polyamide 11 (PA11), which is a very strong polymer and therefore not very compressible.[17] This also reduces the influence it has on the collapse pressure. The effect of the thickness of the polymeric liner thickness is shown in figure 5.16, where a 1mm and 5mm thick liner are compared. This also shows that the polymeric layer has no significant influence on the wet collapse pressure.[51]

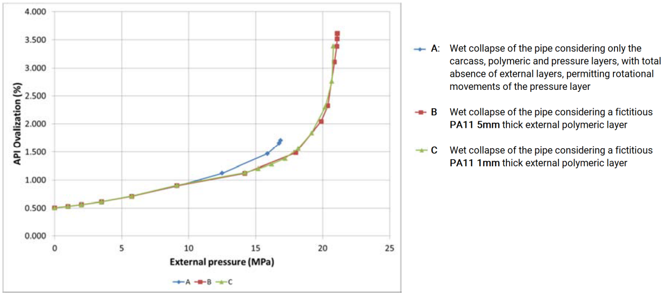

Figure 5.16: Ovalization versus External Pressure for wet collapse of three different models, where the difference between models B and C show the effect of changing the polymeric layer thickness. [51]

5.1.5. D/t Ratio

The D/t ratio is the ratio between the diameter and the thickness of a specific layer. In general, for a single layer pipe the collapse pressure decreases with increasing D/t. With flexible pipes the same response is found where the critical pressure decreases with increasing internal diameter. Increasing the thickness of the carcass and pressure armor increases the critical pressure. The scaling of flexible risers to allow for larger or smaller internal diameters is always a combination of changing both the diameter of a layer as well as the thickness. Determining the appropriate diameter and thickness is done early on in the design process as the internal diameter is prescribed to the manufacturer by the client. [15]

5.1.6. Material Properties

Material selection for the collapse resisting layers is one of the main parameters in the design process. The strength of the material is often simulated using two models. The first is the linear elastic response up to the yield strength, where the stresses linearly increase with the strain. The relationship between the stresses (σ) and the strain (ε) is defined by the Young's modulus (E). The Young's modulus is a measure for the stiffness of the material in the elastic region. Once the stresses in the material exceed the yield strength this elastic model does no longer accurately represent the material behavior, and therefore another model is required. This second model describes the plastic, non-linear, behavior of the material. [52]

One of the main parameters for collapse resistance is the material strength of the collapse resistant layers. [54] Apart from the general material properties, there is also anisotropy within the material. The carcass layer is cold formed which affects the yield strength. The amount of deformation and the bending radius both influence the yield strength and the carcass profile is therefore a complex structure to analyze, as it is composed of sections with different bending radii. There is no homogeneous yield strength and it is difficult to calculate the exact location where buckling occurs. Furthermore, it is hard to determine the exact yield strength of the material in areas where a small bending radius is present. Often however a general estimated average yield strength is used to simplify calculations. [4, 18]

Using steel with a higher yield strength therefore prolongs the linear elastic re-

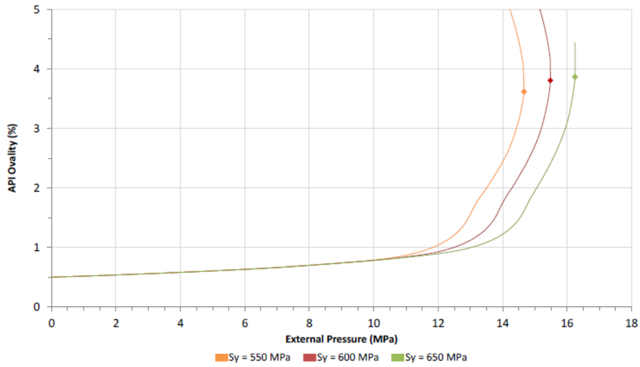

Figure 5.17: Carcass Yield Strength sensitivity. [18]

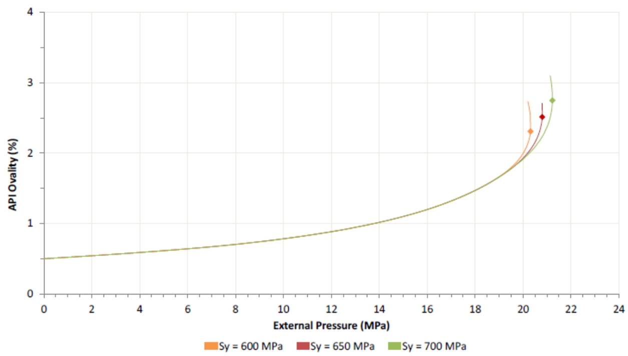

Figure 5.18: Pressure armor Yield Strength sensitivity. [18]

sponse, increasing the critical collapse pressure which occurs in the plastic regime. This can also be seen in figures 5.17 and 5.18, where the effect of an increased yield strength is studied for both the carcass as well as the pressure armor respectively. The yield strength seems to have a larger effect of the carcass layer, this could be due to the more slender structure of the carcass making it more vulnerable to yielding. Furthermore, for collapse to occur the carcass has to deform plastically opposed to the pressure armor that could remain intact.

5.1.7. Axial Load

Risers are suspended from the surface to the seabed, this induces axial tension forces. These forces are mostly handled by the tensile layers, but if there is an axial force in the carcass layer it can affect the collapse pressure quite significantly. Bending also generates axial forces due to friction with other layers. Axial forces negatively influence the collapse pressure. By applying an axial force on one end of the carcass and fixing the other the influence of this axial pressure can be investigated. This was done for an 6 and 8 inch riser using a FE model and the results are shown in figures 5.19 and 5.20. Tables 5.4 and 5.5 show the riser parameters and the normalized reduction in collapse pressure for the 6 and 8 inch risers with respect to the reference case with no axial force applied to the carcass.

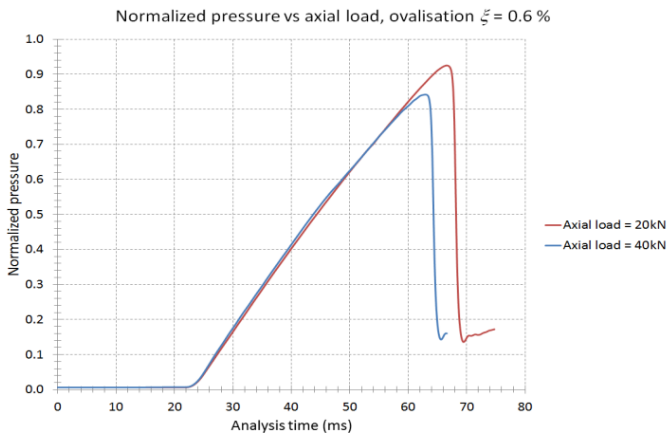

Figure 5.19: Influence of axial load on the collapse pressure, comparison between axial loads of 20 and 40kN on an 6 inch flexible riser. [4]

Table 5.4: Influence of axial load on the collapse pressure of an 6 inch flexible riser, normalized to the model with 0.6% ovalization and a gap width of 1.4 mm without any axial load. [4]

Axial Load (kN)	0	20	40
Ovalization (%)	0.6	0.6	0.6
Gap Size (mm)	1.4	1.4	1.4
Normalized collapse pressure	1.0	0.925	0.841

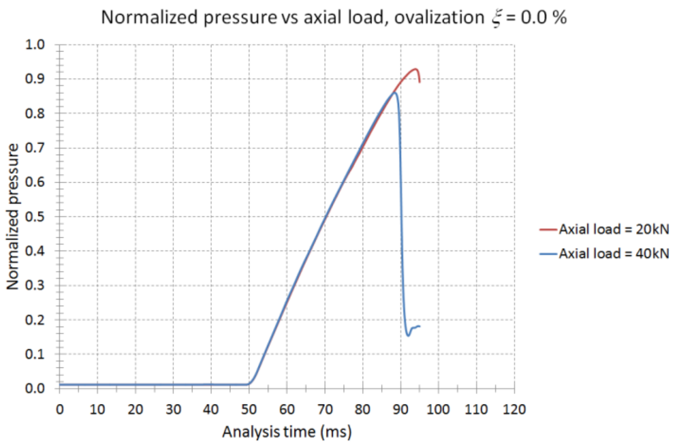

Figure 5.20: Influence of axial load on the collapse pressure, comparison between axial loads of 20 and 40kN on an 8 inch flexible riser. [4]

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Table 5.5: Influence of axial load on the collapse pressure of an 8 inch flexible riser, normalized to the model with 0.6% ovalization and a gap width of 1.4 mm without any axial load. [4]

Axial Load (kN)	0	20	40
Ovalization (%)	0.6	0.6	0.6
Gap Size (mm)	1.4	1.4	1.4
Normalized collapse pressure	1.0	0.929	0.860

5.1.8. Pressure armor strength (thickness)

Under wet collapse conditions, where external pressure is acting directly on the liner and carcass layer, it is the pressure armor that helps the carcass layer in retaining its circular shape. It adds radial stiffness preventing the carcass layer from expanding outwards, a phenomenon that occurs with the ovalization of the carcass. By keeping the carcass layer circular under larger external pressure it increases the wet collapse pressure. The amount of radial stiffness it adds depends on the profile and strength of the pressure armor. Often it is only the (equivalent) thickness of the pressure armor that is varied to simulate an increase or decrease of radial stiffness of the pressure armor. Increasing the thickness results in a higher radial stiffness and consequently a higher wet collapse pressure, this is also illustrated in figure 5.21. This figure shows that the pressure armor thickness, or strength, can have a huge influence on the collapse pressure. It can also be seen that increasing the collapse pressure by increasing the pressure has a limit, this is when the collapse mode of the carcass is forced to heart mode. From this point the radial stiffness is to large to allow any radial expansion of the carcass, meaning the effect of the pressure armor is saturated.

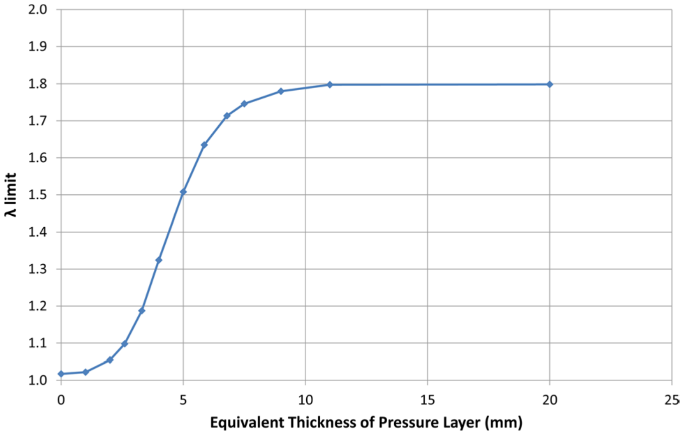


Figure 5.21: Some values of equivalent thickness of pressure armor and the corresponding obtained wet collapse pressures. [49]

The effect of the relative radial stiffness can be investigated by changing the pressure armor thickness while keeping the carcass layer unchanged in a FE model. The results of a FE study into the effect of the pressure armor thickness used this principle and the results are shown in figure 5.22. Three collapse modes were found, the heart mode collapse, eight mode collapse and a transitional collapse mode. This transitional collapse mode is a combination of the two primary collapse modes. This research showed that with increasing pressure layer thickness, or radial stiffness, the collapse pressure also increases. It also shows that for relative small radial stiffness of the pressure armor the eight mode collapse is preferred, while for the large relative radial stiffness heart mode collapse is predominant, this will be further discussed in chapter 5.2.

One can further notice the monotonic shape of the curve, where an increase in pressure armor thickness increases the carcass collapse pressure. The response on the pressure layer thickness is not linear as it is only had a relevant influence up to a certain point when it reaches a saturation point. [49]

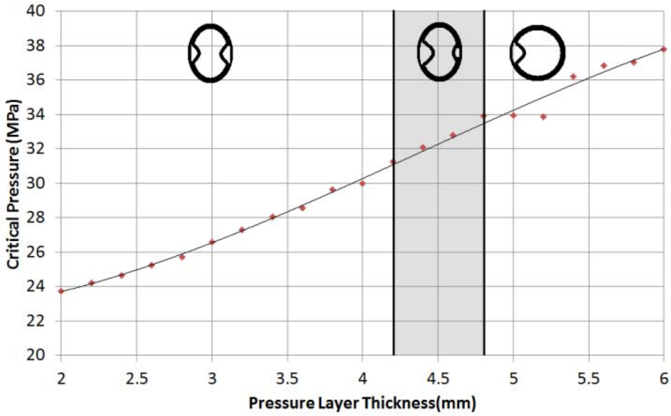


Figure 5.22: Influence of the pressure layer thickness on the critical pressure, as well as the collapse mode shape for a singly symmetric ovalization of 2%. [45]

5

5.1.9. Carcass profile

The carcass profile is a complex structure, and the shape can be optimized for different applications. Manufacturers may develop their own profile which are often kept confidential together with the material selection and production method. In general, the profile has an interlocking “S” shape. Changing the shape parameters of this general profile gives an insight into the sensitivity of the collapse pressure on the shape of the carcass profile.

Some parameters that can be changed are the inclination of the front and rear legs, as shown in figures 5.23 and 5.24 respectively, and the slope angle which is shown in figure 5.25. The sensitivity of the collapse pressure was investigated for both 6 and 8 inch diameter pipes using a FE method. The inclination of the front and rear legs was changed by 10°and the slope by 2°in both directions.

Front leg

Bending the font leg up by 10°decreases the collapse pressure compared to the normal profile by a small amount: 4% for the 8 inch pipe and 0.6% for the 6 inch pipe. Bending the front leg down increases the collapse pressure by 4% for the 8 inch pipe and by 8.4% for the 6 inch pipe. Bending down the front leg moves it further away from the central axis of the profile increasing the moment of inertia and thus the bending stiffness, the opposite effect can be seen when bending the front leg up.

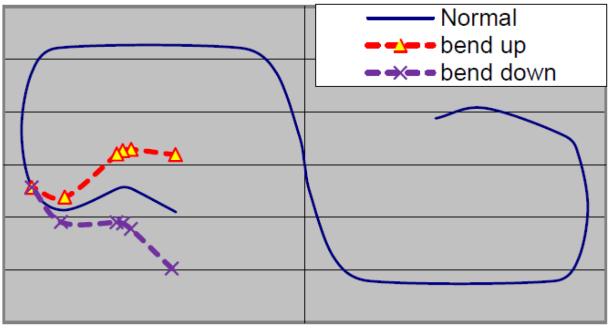


Figure 5.23: Illustration of varying the front leg profile angle by 10°up and down. [76]

Table 5.6: Influence of varying the front leg profile angle on the collapse pressure of an 6 and 8 inch flexible riser, normalized to the nominal profile without bending. [76]

Pipe ID: Profile:	Nominal profile	Bend up by 10°	Bend down by 10°
6 inch	1.0	0.994	1.084
8 inch	1.0	0.96	1.04

Rear leg

Using a similar approach for the rear leg showed the same results, however as the rear leg is in the opposite side of the central axis the effect is opposite of that of the front leg considering bending it up and down. When the leg is bent up is moves away from the central axis thus increasing the bending stiffness by increasing the moment of inertia, and therefore the collapse pressure. Bending the rear leg down has the opposite effect. However, the effect of bending the rear leg has a much smaller influence on the collapse pressure compared to that of the front leg with a maximum change in collapse strength of 3%.

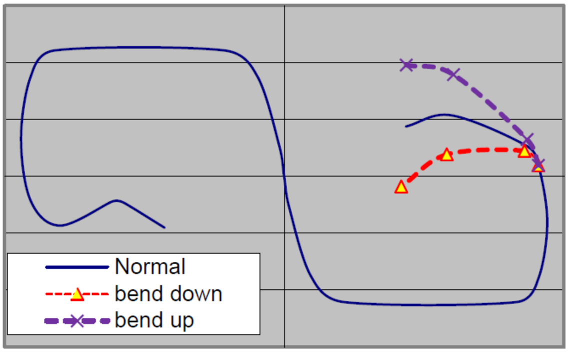


Figure 5.24: Illustration of varying the rear leg profile angle by 10°up and down. [76]

Table 5.7: Influence of varying the front leg profile angle on the collapse pressure of an 6 and 8 inch flexible riser, normalized to the nominal profile without bending. [76]

Pipe ID: Profile:	Nominal profile	Bend up by 10°	Bend down by 10°
6 inch	1.0	1.0	0.99
8 inch	1.0	1.03	1.01

Slope angle

The third parameter to be varied is the slope angle. This can be done similar to the bending of the front and rear legs, but as is changes a longer section the bending can be reduced to 2% in either direction to simulate a significant profile change. Varying the slope angle may change collapse strength compared to the nominal profile, where it appears that bending down increases the collapse strength. For the 6 inch diameter pipe the collapse pressure showed an increase of 5.8% and for the 8 inch pipe an increase of 6.6%. Bending up seems to have little effect where a reduction in collapse pressure of 2% is observed for the 8 inch diameter pipe and an increase of 1.2% for the 6 inch diameter pipe. In this case bending down seems to follow the same principle where the moment of inertia is increased.

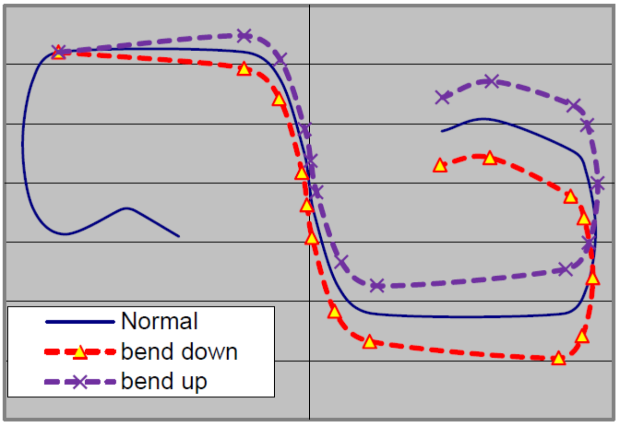


Figure 5.25: Illustration of varying the slope angle profile by 2°up and down. [76]

Table 5.8: Influence of varying the slope angle on the collapse pressure of an 6 and 8 inch flexible riser, normalized to the nominal profile without bending. [76]

Pipe ID: Profile:	Nominal profile	Bend up by 2°	Bend down by 2°
6 inch	1.0	1.012	1.058
8 inch	1.0	0.98	1.066

5.1.10. Pitch and lay angle of carcass and pressure armor

The carcass and pressure armor consist of a metallic profile that is helically wounded. The pitch of this helical shape is the distance that is covered in axial direction per winding, also shown in figure 5.26. When the pitch equals 0, the profile follows a perfect circular shape. In general the pitch is small compared to the length of the pipe, and many models used exclude the pitch from the analysis. [50]

However, the pitch can have a significant influence on the wet collapse pressure. A perfect circle has a much higher collapse strength compared to a helical shape as it excludes forces in axial direction. Furthermore, the collapse pressure decreases with increasing pitch, this can also be seen in figures 5.27 and 5.28 where the effect of the pitch of the carcass and pressure armor was investigated using a FE model. The pitch influences the initial lockup of the profile into itself, which as a results influences the critical pressure. When the pitch is small the profile locks up into itself before plastic deformation occurs. This lockup increases the radial stiffness as the profile sections corporate in resisting collapse. This prolongs the linear elastic response and allows the structure to mobilize more before plastic deformation occurs. [18]

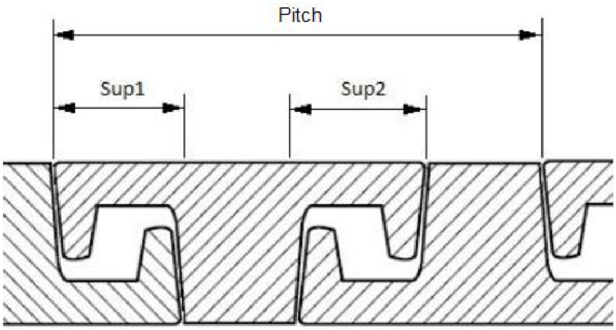


Figure 5.26: Illustration of the pitch and super of an helically winded pressure armor, the same holds for the carcass. [45]

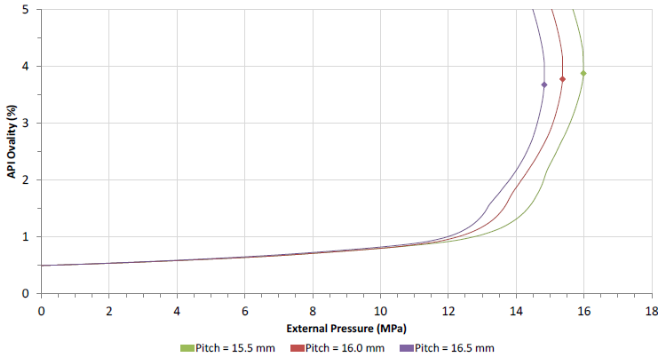


Figure 5.27: Influence of the carcass pitch on the collapse pressure. [18]

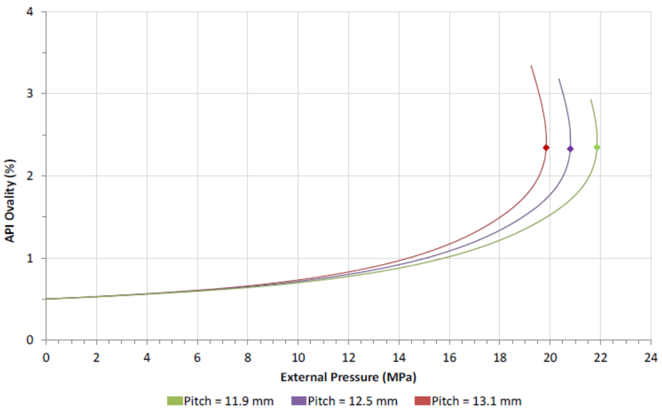


Figure 5.28: Influence of the pressure armor pitch on the collapse pressure. [18]

5.1.11. Friction of layers

Unbonded flexible risers allow movement between the different layers. However, due to the tight concentric configuration and the used materials these layers experience friction when moving relative to each other. When modeling unbonded flexible risers using FE methods different contact models can be used, especially with respect to the contact mechanism between the pressure armor layer, liner and carcass affect the buckling capacity of the riser. [15]

One way of testing the influence of friction is by changing the friction factor of the carcass and pressure armor in a FE model. The results of such a test showed that friction increases the critical collapse pressure. This friction only has an effect when self-contact occurs, and therefore the friction factor of the pressure armor is more influential as this structure is always in contact with itself. The carcass profile only makes self-contact after an initial collapse, which results in a smaller influence of the friction on the collapse pressure as yielding has already occurred before friction has had an influence. The influence of friction in the carcass layer and pressure armor are shown in figures 5.29 and 5.29 respectively. This also shows that there is a significant increase of 10% in collapse pressure when the friction coefficient of the pressure armor equals 0.3 compared to the frictionless case, while the friction coefficient of the carcass seems to have little effect on the collapse pressure.

5

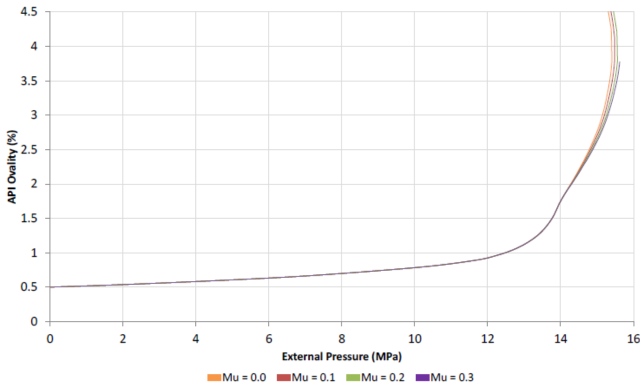


Figure 5.29: Influence of the carcass friction factor on the collapse pressure. [18]

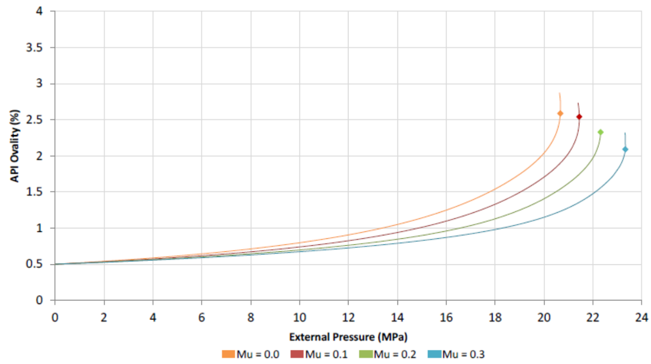


Figure 5.30: Influence of the pressure armor friction factor on the collapse pressure. [18]

5.1.12. Initial interference between liner and pressure armor

During the manufacturing of an unbonded flexible risers, the layers are build up from the carcass to the external sheath. In this process the pressure armor is rolled on top of the carcass and liner, which causes an initial contact pressure between the liner and pressure armor as the liner is effectively compressed between the carcass and pressure armor. The significance of this pressure can be numerically studied as an initial interference effect.

The result of this study is shown in figure 5.31 and shows that the initial interference does influence the collapse pressure, where a larger interference results in a higher collapse pressure. The interference has an opposite effect of that of the gap width, where a higher interference reduces the free motion of the carcass layer thus preventing ovalization.

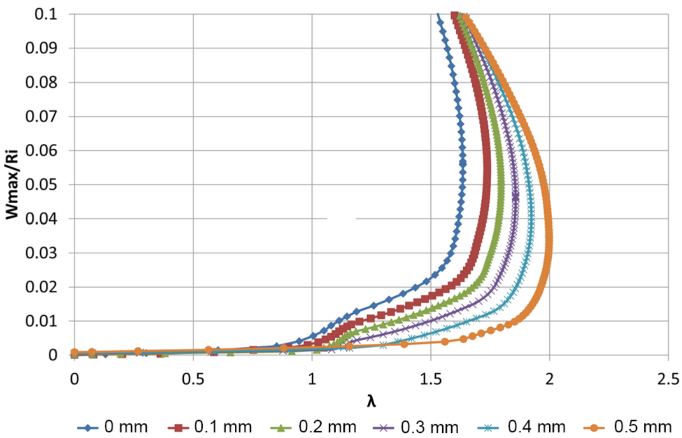


Figure 5.31: Influence of the Initial interference between the liner and pressure armor on the collapse pressure. An initial ovalization of 0.5% was present. [49]

5.1.13. Mass flow rate

Depending on the gas or fluid being transported, the mass flow rate (MFR) might influence the collapse pressure. A higher MFR is created by having a larger pressure gradient, which means a internal pressure at the bottom section where there is also the highest external pressure and a very low pressure at the top section. A numerical study into the effect of the MFR showed that the it can influence the collapse pressure, also shown in figure 5.32 and 5.33, where a higher MFR increases the collapse pressure. [4]

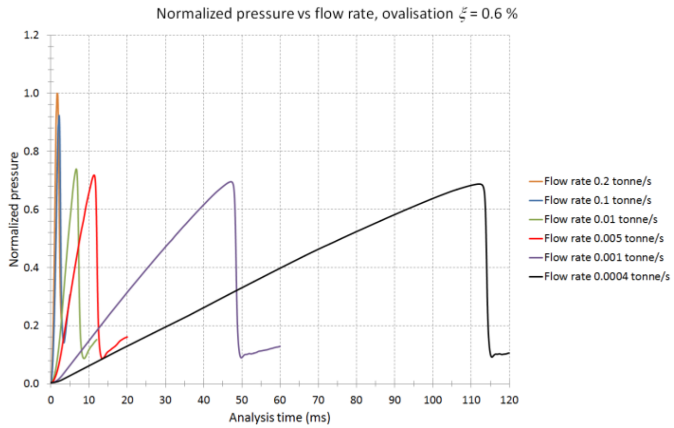


Figure 5.32: Influence of the mass flow rate on the collapse pressure. An initial ovalization of 0.6% was present. [4]

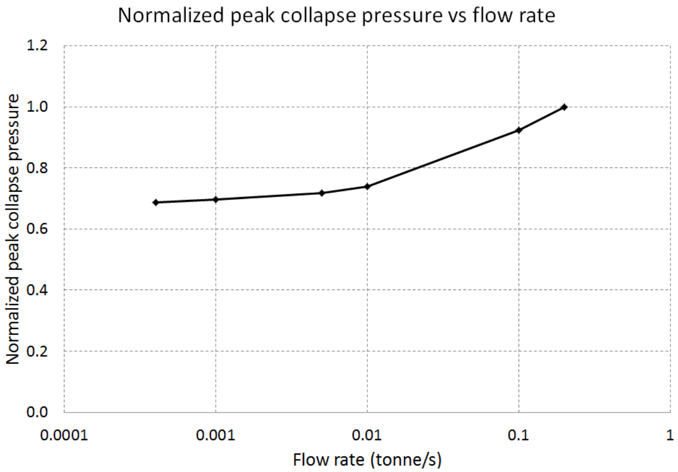


Figure 5.33: Summary of the peak collapse pressures as found in 5.32. [4]

5.2. Collapse Mode Shape

There are many factors influencing the collapse pressure of flexible risers, but only some of them influence the collapse mode shape. The following factors can most significantly change the collapse mode shape:

- Curvature
- Ovalization
- Pressure armor strength

There are two principles found that influence the collapse mode shape. The first is the amount of symmetry, this can be either in the loading or in the cross-section shape. The second is by influencing the radial stiffness of the layers surrounding the carcass, most significantly the pressure armor. Curvature and ovalization of the riser both change the symmetry within the riser which changes the collapse shape. The relative strength of the pressure armor influences the radial stiffness.

5.2.1. Radial stiffness of pressure armor

Collapse of the carcass is obstructed by the radial stiffness of the layers surrounding the it. These layers prevent the carcass from expanding outward, and thus from ovalizing in a doubly symmetric way. All of the layers of the flexible riser contribute to the radial stiffness, but the pressure armor has the most significant influence as its main purpose it to resist radial forces giving it a very high radial stiffness compared to the other layers.

The radial stiffness of the pressure armor is often related to the thickness of the pressure armor, as a larger thickness results in a higher radial stiffness and where the thickness is a simple parameter to be changed to inspect the influence of radial stiffness rather than linearly changing the actual radial stiffness.

The radial stiffness is mostly dependent on the profile and thickness of the pressure armor, where the radial stiffness increases with the moment of inertia of the pressure armor profile and the thickness. As mentioned before, a high radial stiffness of the pressure armor relative to the carcass prevents the carcass from deforming in a doubly symmetric shape. This is because the pressure armor is resisting the outward expansion of the carcass along its largest diagonal.

The natural tenancy of a pipe is to collapse in eight mode, and thus a doubly symmetric ovalization. Eight mode collapse is also known as four hinge collapse, where only four parts of the cross section deform plastically and effectively function as hinges, also marked in figure 5.34. The other sections of the cross section remain intact and largely undeformed. This also explains why the eight mode collapse is naturally preferred over heart mode collapse, as only these four point have to deform meaning it requires much less energy to collapse than the heart mode shape, where much larger part of the cross section has to plastically deform.

The radial stiffness of the pressure armor resists the naturally preferred eight mode collapse. When this resistance is to large relative to the outward pressure exerted by the carcass, the carcass is prevented from collapsing in eight mode shape and is forced into heart mode collapse. This amount of resistance thus can determine the shape of collapse, together with the shape and amount of initial ovalization as described in section 5.2.2.

With a small thickness of the pressure layer, or more precise a small radial stiffness relative to the carcass, resistance against doubly ovalization is insufficient to force the carcass into heart mode collapse, and eight mode collapse will therefore naturally occur. When the thickness of the pressure armor in increased, it becomes more likely that heart mode collapse occurs. This can also be seen in figure 5.35, showing the results of a FE analysis into the collapse mode shape of a flexible riser where the model was run several times while changing the thickness of the pressure armor layer. The model included a 0.2% singly ovalization and consisted of three individually modeled layers: carcass (innermost), polymer liner and a pressure armor (outermost). The thickness of the pressure armor layer was varied between 2 and 6 mm with increments of 0.2 mm. [9, 45]

The results in figure 5.35 show three types of collapse: eight mode, heart mode and a transitional mode. This transitional shape is most closely related to an eight mode collapse influenced by the initial singly ovalization. Furthermore, it was also mentioned that the differentiation between eight mode and the transitional mode was very subtle and the frontier between them is therefore imprecise and dependent on the judgment of the analyst. [45] The change from transitional mode to the heart mode is much more distinct, and is therefore chosen to be the main focus for studying the effect of the pressure armor on the collapse mode shape.

This transitional collapse (figure 4.7b) mode is found between eight mode and heart mode collapse conditions, and could be the result of the radial strength of the layers surrounding the carcass increasing with increasing deformation. Meaning that initially eight mode collapse was occurring but when the ovalization and radial expansion developed, the radial strength of the surrounding layers increased as it requires more force to further deform these layers, eventually restricting eight mode ovalization and inducing heart mode.

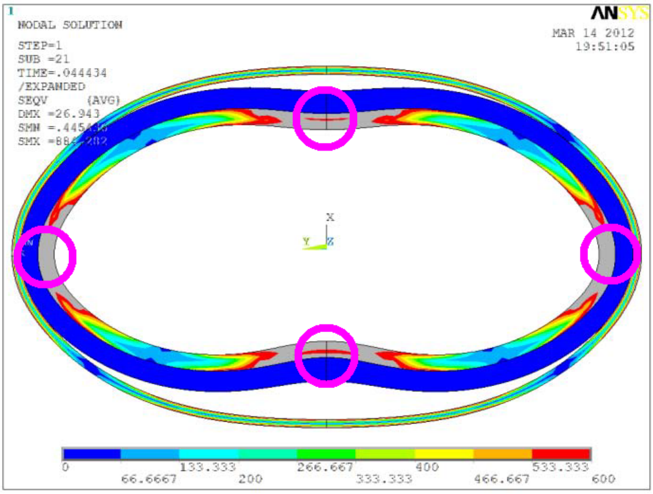


Figure 5.34: Eight mode collapse shape, also known as four hinge collapse. These four "hinges" are marked with purple circles. [45]

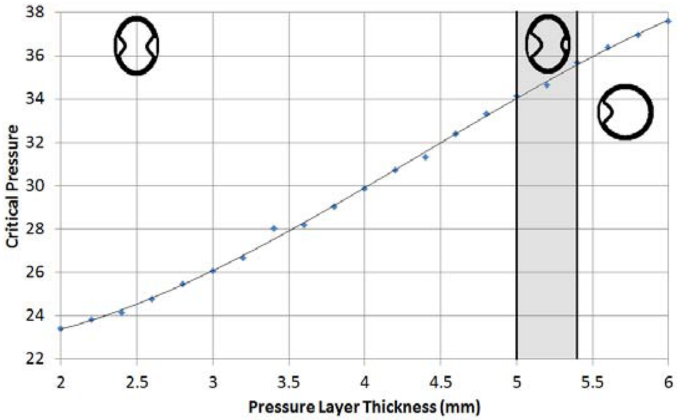
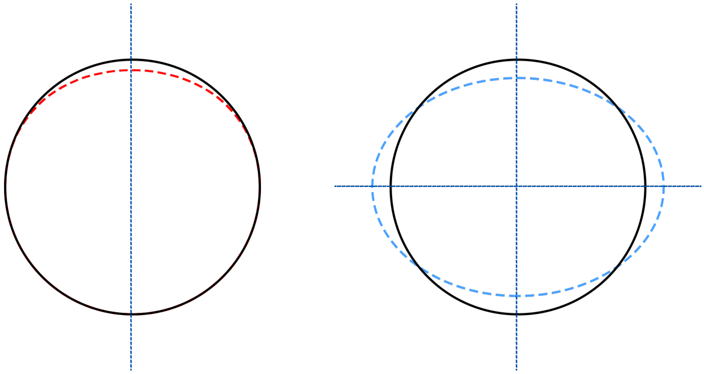


Figure 5.35: Results of an FE analysis on the collapse pressure and mode shape of a flexible riser consisting of a carcass, polymer liner and pressure armor. The thickness of the pressure armor was increased from 2 to 6mm with increments of 0.2mm with an initial singly symmetric ovalization of 0.2%. [45]

5.2.2. Singly or doubly symmetric ovalization

A perfect circular riser does not exist, there is always some imperfection, often in the shape of an ovalization, that reduces the symmetry. In general two types of ovalization are known to influence the collapse mode shape. These are singly and double symmetric ovalization and are illustrated in figure 5.36. Heart mode has a singly symmetric shape and by imposing a singly symmetric ovalization, the probability of heart mode collapse is increased. The same holds for eight mode collapse which has a doubly symmetric shape. The amount of symmetry in the cross section or loading influences the amount of symmetry of the internal stresses, and this way influences the collapse mode shape.

Large singly symmetric loading or deformation creates an uneven distribution of forces throughout the cross-section, where one section of the riser experiences larger stresses. This is the same region where the ovalization is located, e.g. the top section of 5.36a. This results in this section being more likely to fail as larger stresses induce larger deformations due to strain. This deformation increases the amount of ovalization, making this a self strengthening effect. Buckling will occur at one point where the internal stresses exceed the yield strength of the material and sudden large plastic deformation occurs due to the self strengthening effect, resulting in heart mode collapse. [54]



(a) Singly symmetric **(b)** Doubly Symmetric

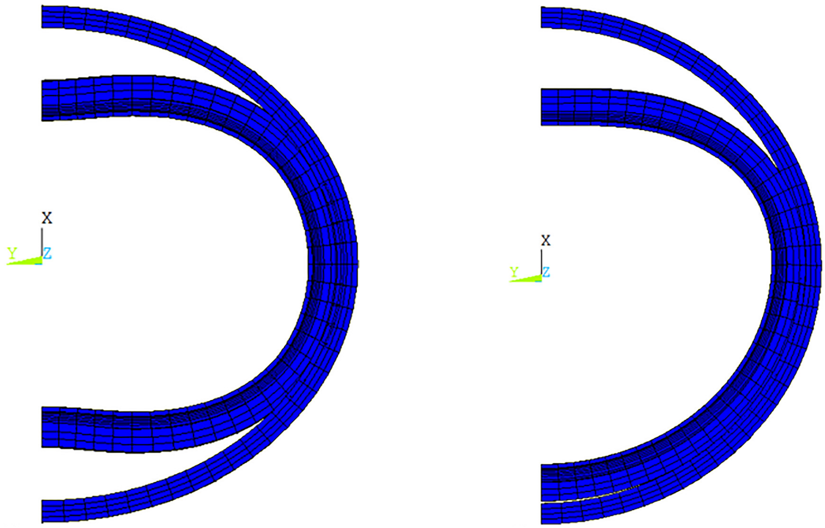
Figure 5.36: Illustration of Singly and Doubly symmetric ovalization with symmetry planes.

This also means that when a singly symmetric ovalization is present, the flexible riser still tends to collapse in eight mode shape for small thickness or radial stiffness of the pressure armor. With a doubly symmetric ovalization, eight mode shape is preferred over heart mode shape for larger thickness or radial stiffness of the pressure armor than with singly symmetric ovalization. This can also be seen by comparing figures 5.35 and 5.39. When a singly ovalization of 0.2% is imposed, the carcass collapsed in heart mode shape when the pressure armor was thicker than 5.4 mm. Compared to the situation where a doubly symmetric ovalization ov 0.2% is imposed where a pressure armor thickness of 6mm is still not enough to force the carcass into heart collapse. This does not mean that heart mode collapse does not occur with doubly symmetric initial ovalization, only that a pressure armor thickness of 6mm was not sufficient in this simulation. If the pressure armor thickness would be further increased heart mode collapse would occur.

This shows that even when a doubly symmetric initial imperfection is induced, singly symmetric heart mode collapse can still occur. When a singly symmetric imperfection is imposed it is also still possible to have doubly symmetric eight mode collapse. This is because it is not only the shape of imperfection but also the severity of the deformation that influences the collapse mode shape, beside the relative stiffness of the surrounding layers. With increasing severity of singly ovalization, the required thickness for heart mode collapse decreases. This effect can be seen in figures 5.38a to 5.38d, where the pressure armor thickness where heart mode shape collapse occurs decreases from 5.4 mm with 0.2% ovalization to 4.6 mm with an ovalization of 3%.

Beside the initial ovalization of the carcass due to imperfections, another type of asymmetry can be present to influence the collapse mode shape, e.g. curvature. Figure 5.37 shows the influence of the amount of curvature on the collapse mode shape. When the riser is bent to its MBR it collapses into the heart mode, while for a curvature of 100 times its MBR it collapses into the eight mode. Curvature of the riser induces singly asymmetric loading as described in section 5.1.1 and increases the probability of heart mode collapse with decreasing bending radius.

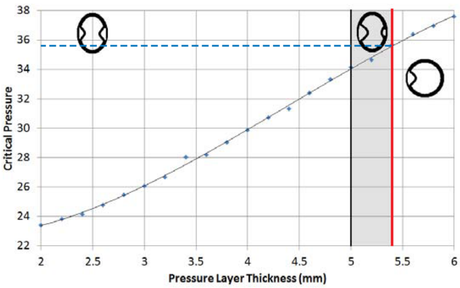
A similar effect is observed with the amount of ovalization, a large initial singly symmetric ovalization increases the probability of heart mode collapse. This can also be seen in figures 5.38a to 5.38d, with increasing singly symmetric ovalization the external pressure required for heart mode collapse to occur decreases from about 35.5 MPa for 0.2% ovalization to about 33 MPa for 3% ovalization. Also the pressure layer thickness where heart mode collapse occurs decreases, meaning that heart mode collapse can occur in a larger region of pressures and with a larger spectrum of pressure layer thicknesses (as heart mode collapse only occurs when the pressure layer has a sufficiently large thickness or radial stiffness, more on this in section 5.2.1).



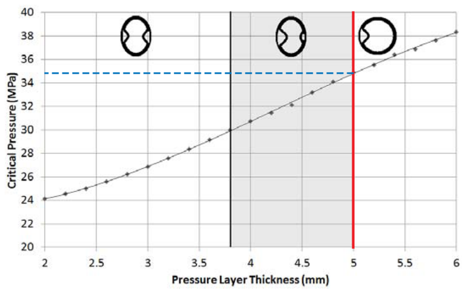
(a) Eight mode collapse when bent to 100 times the MBR.

(b) Heart mode collapse when bent to MBR.

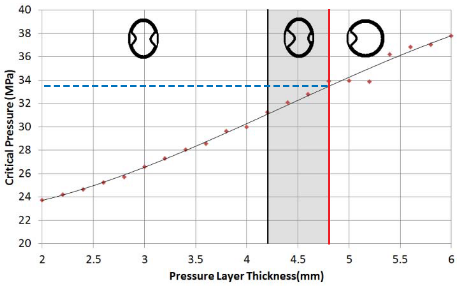
Figure 5.37: Results of a FE analysis of a flexible riser where the influence of the curvature was studied. With a radius of curvature 100 times larger than the MBR an eight mode collapse mode was found, when the radius of curvature was set to the MBR collapse in the heart mode shape was found. [52]



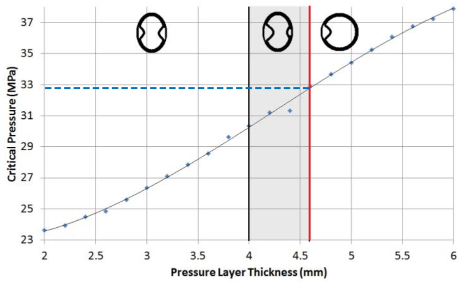
(a) Initial singly ovalization of **0.2%**.



(b) Initial singly ovalization of **1%**.



(c) Initial singly ovalization of **2%**



(d) Initial singly ovalization of **3%**.

Figure 5.38: Results of an FE analysis on the collapse pressure and mode shape of a flexible riser consisting of a carcass, polymer liner and pressure armor. The thickness of the pressure armor was increased from 2 to 6 mm with increments of 0.2 mm with four severities of initial singly symmetric ovalization. The pressure armor thickness required for heart mode collapse decreases with the amount of singly ovalization.[45]

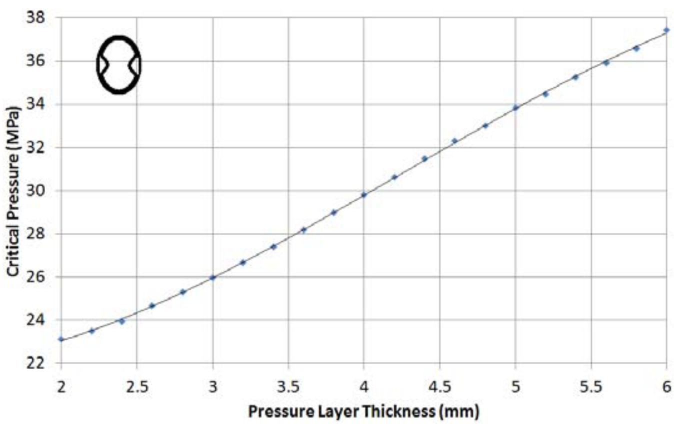


Figure 5.39

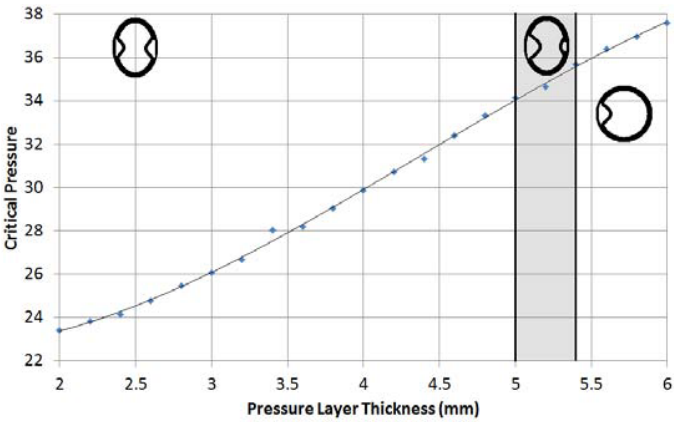

38
36
34
32
30
28
26
24
22
Critical Pressure
2
2.5
3
3.5
4
4.5
5
5.5
6
Pressure Layer Thickness (mm)

Figure 5.40

5.3. Conclusion

As mentioned before, and shown in this chapter, there are many factors that can influence the collapse response. But not all of these factors are similarly relevant. It can be concluded that bending, ovalization (especially of the pressure armor), gap width, axial loading, initial interference and the pressure armor strength are the most influential and relevant factors with respect to influencing the collapse pressure. Other factors that were quite influential but considered irrelevant as they are mostly design considerations that can not simply be changed included the D/t ratio, material properties (submissive to material selection), carcass profile (manufacturer depended and often confidential making it difficult to be analyzed) and the mass flow rate. Most of these factors (possibly excluding carcass profile) are chosen or predetermined based on other design or operational aspects and requirements. The pitch or lay angle and polymeric layer thickness had very little influence on the collapse pressure.

Some of these factors were found to also influence the collapse mode. By inducing a curvature, specific ovalization shape or changing the strength of the pressure armor relative to the carcass the collapse mode can be changed. These factors influenced the collapse mode by one of two principles. The first principle is the amount of symmetry, either in the loading or in the cross-section shape. The second is the combined radial stiffness of the layers surrounding the carcass, where the pressure armor is the most influential.

Two types of symmetry are of interest, these are singly or doubly symmetric. This is because these coincide with the amount of symmetry found in heart mode (singly symmetric) and eight mode (doubly symmetric) collapse. By inducing singly symmetry within the riser, the probability of heart mode collapse can be increased, and the same holds for doubly symmetry and eight mode collapse.

The radial stiffness of the layers surrounding the carcass form a obstruction to ovalization of the carcass, where it is impeded from expanding outward. This forces the carcass to collapse in heart mode, when the radial stiffness of the pressure armor is large relative to that of the carcass. When the pressure armor has a relatively small radial stiffness, the ovalization and outward expansion of the carcass will deform the pressure armor with it. This allows the naturally preferred eight mode collapse to still occur.

6 Conclusion

The collapse of unbonded flexible risers is a complex mechanism due to the multi-layered concentric structure composed of different materials, where each layer is assigned a specific task. Many of these interactions are non-linear and can only be investigated using FE analysis or experiments. Analytical analysis can be a good method for making a rough estimate of the collapse pressure, as long as it concerns simple structures consisting of not more than one or two concentric layers with simple cross-sections where the material behavior remains in the elastic regime. Flexible risers consist of many more layers, making FE analysis the preferred method for analyzing and certifying flexible risers as experimental testing is often too expensive.

The collapse response of unbonded flexible risers was found to be influenced by many different factors. The most mentionable factors influencing the critical collapse pressure include:

- Curvature
- Gap width
- Ovalization (and geometric imperfections)
- Polymeric layer thickness
- D/t ratio
- Material properties
- Axial load
- Pressure armor strength (thickness)
- Carcass profile
- Pitch and lay angle of the carcass and pressure armor

- Friction between layers
- Initial interference between polymeric layer and pressure armor
- Mass flow rate

Not all of the above mentioned factors are similarly relevant. It can be concluded that bending, ovalization and imperfections (especially of the pressure armor), gap width, axial loading, initial interference between the liner and pressure armor or carcass and the pressure armor strength are the most influential and relevant factors with respect to influencing the collapse pressure. Other factors that were quite influential but considered irrelevant as they are mostly design considerations that can not simply be changed included the D/t ratio, material properties (submissive to material selection), carcass profile (manufacturer depended and often confidential making it difficult to be analyzed) and the mass flow rate. Most of these factors (possibly excluding carcass profile) are chosen or predetermined based on other design or operational aspects and requirements. The pitch or lay angle and polymeric layer thickness had very little influence on the collapse pressure.

Pipes naturally tend to collapse in eight mode, but of these thirteen factors three were found to significantly influence the collapse mode shape. These include the curvature, ovalization and pressure armor strength, where the gap width can be seen as a factor allowing ovalization to occur such that it indirectly influences the collapse mode shape. The way these influence the collapse mode shape can be separated into two principles.

The first is by creating an amount of symmetry either in the loading or in the carcass shape. Curvature induces both ovalization and singly asymmetric loading and can significantly influence the collapse mode shape. The asymmetry in the carcass shape is mostly singly or doubly symmetric ovalization, where singly symmetric ovalization complements the heart mode and therefore increases the probability of heart mode collapse. Doubly ovalization has a large resemblance with the eight mode shape, and can similarly induce eight mode collapse.

The second principle is by changing the radial stiffness of the layers surrounding the carcass, most significantly the pressure armor as this layer has the highest radial stiffness where its main purpose is to withstand external radial pressure. The radial stiffness is closely related to the thickness of the pressure armor where a small thickness results in a low radial stiffness and a hight thickness in a large radial stiffness. When the pressure armor has a high radial stiffness compared to the carcass it restricts the outward expansion of the carcass present with eight mode collapse, forcing the carcass into heart mode collapse. With a relatively small radial stiffness the pressure armor can ovalize with the carcass and allow it to collapse in eight mode shape.

The goal of this project was to make an overview of the development of theories and approaches deployed for the research on the collapse modes of flexible UDW risers subjected to external pressure, where the following aspects had to be investigated:

- Development of deep sea riser industry, including riser types, structural configuration, material, relevant rules /standards etc.
- Main factors that affect the critical pressure and collapse modes of the risers.
- Available theories and approaches (analytical, numerical, experimental methods etc.) deployed to identify the reasons of collapse modes of risers.
- Possible collapse mechanisms of the risers when influenced by the those factors.
- Possible inter-relationship among the main factors that influence the collapse modes.

All these aspects have been reviewed upon in this report. However, the inter-relationships among the factors influencing the collapse behavior requires more investigation, as most studies have been on the influence of individual factors. Flexible risers are complex structures where all of the layers interact with each other, which also means that all the influencing factors combined prescribe the collapse behavior, where the properties of all of the individual layers play a role.

Nevertheless, an interesting relation between the pressure armor thickness and amount of ovalization was found, where a thicker pressure armor (resulting in a higher radial stiffness) allowed for a larger amount of ovalization to occur before heart mode collapse was induced. Another interesting observation was that the pressure armor has a larger influence on the carcass collapse pressure than the carcass layer itself, an effect that was found in the sensitivity studies of the friction of layers, pitch and lay angles and ovalization. The recommendation is therefore to focus further investigations into improving the wet collapse capacity of flexible risers on the pressure armor.

Another observation related to the pressure armor (or radial strength in general) was the transitional collapse (figure 4.7b) mode, which is found between eight mode and heart mode collapse conditions. This could be the result of the radial strength of the layers surrounding the carcass increasing with increasing deformation. Meaning that initially eight mode collapse was occurring but when the ovalization and radial expansion developed, the radial strength of the surrounding layers increased as it requires more force to further deform these layers, eventually restricting eight mode ovalization and inducing heart mode. If this is actually the case should be further investigated.

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