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Preface

This thesis is written as a final closure for my master degree program in Offshore Technology; Subsea Technology. The thesis was conducted from January to June in 2013, at the University of Stavanger. The thesis is about how buoyancy is interpreted in general and when calculating top tension in risers and drill pipes.

I would like to thank my supervisor Arnfinn Nergaard for his patience during this thesis. Many hours have been spent discussing the topic of this thesis. I would also like to thank professors Rune W. Time, Bernt S. Aadnoy and Bjørn H. Hjertager for their contributions during this thesis.

A special thanks goes to technician Arne Haaverstein from IRIS for his help making the pressure vessel needed to conduct the Bridgman experiment.

Last but not least I would like to thank my girlfriend Malene Skogly for her patience, support and understanding during this semester.

This thesis has helped me understand how buoyancy and hydrostatic forces contributes when calculating the needed top tension in risers and drill pipes.

Morten Reve

Summary

This thesis highlights that there is a wide spread use of the term “buoyancy” in the petroleum industry which can lead to misunderstandings. It is also evident that terms like “effective tension”, “true wall tension” and “apparent weight” induces more misunderstandings if not understood correctly.

It has been shown that there is different ways of interpreting buoyancy forces and how they act on a submerged object. Several experiments have been used as illustrations to show that fluid need non-vertical sides to create a lift force (buoyancy force) on an object immersed in fluid although the object is displacing fluid.

Further on it has been shown that the effective tension concept used in marine riser calculations can be misinterpreted because of the different buoyancy understandings. When calculating the effective tension the influence of the horizontal pressure acting on the riser is accounted for. In other terms, the effective tension is a three dimensional stress calculation which gives the needed top tension force (in one dimension) to prevent buckling as an answer. This calculation can be interpreted as if there were a “buoyancy force” present along the entire length of the riser, which is a contradiction to what has been presented in this thesis.

An experiment has been conducted to show that effective tension concept gives correct results of the internal three dimensional stress state and thus is not just a “fictitious stress” or “fictitious force” as mentioned in several papers, but a stress state which can lead to failures if not accounted for.

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Chapter 1. Introduction

This thesis' purpose is to investigate how the term "buoyancy" is interpreted from a volume perspective and a pressure perspective. These two interpretations are then used to show that there is a need for clarification of the term when dealing with top tension calculations on marine risers and drill pipes.

An experiment will be conducted; the results from the experiment will be used to show that effective tension can be used to calculate the internal stress state of an object. We will also create an analogy on how the effective tension concept, a concept used to calculate the top tension needed to prevent a riser string to buckle, can be interpreted.

The thesis will be divided into 8 chapters. Chapter 1 contains the introduction, objectives and the background for the thesis. In chapter 2 we will have an introduction to the oil and gas industry in general and a more specific introduction to marine riser systems and components of such systems. Chapter 3 will introduce us to the effective tension concept as derived by C. P. Sparks. In chapter 4 we take a closer look at how text books in physics introduce students to buoyancy, discuss the two views and in the end define the term "buoyancy". Chapter 5 will introduce two schools of understanding buoyancy in the petroleum industry which is based on the discussion in chapter 4. A literature study on the subject has been performed. Chapter 6 will be devoted to an experiment conducted to show how effective tension describes a "real" force. In chapter 7 there will be a discussion based on the results from the experiment. Chapter 8 will contain a conclusion and a recommendation for further work.

1.1 Objectives

- Give an introduction to the offshore petroleum industry and marine riser systems
- Introduce the effective tension concept – a concept used to calculate the top tension needed to prevent a marine riser from buckling
- Investigate how buoyancy is understood in basic physics and illustrate that there is a need for clarification of the term "buoyancy" when and object is placed at the bottom of a fluid container
- Based on the buoyancy investigation and a literature study, show that misinterpretation of the term "buoyancy" when doing calculations using the effective tension concept can occur
- Show that effective tension is not a fictitious force by conducting an experiment. Discuss the results from the experiment
- Prepare conclusions and recommendations for further work

1.2 Background

When doing calculations on top tension needed to prevent marine risers from buckling the effective tension concept is often used. [1] There seems to be much confusion on what effective tension means, and how to derive the equations used. To further complicate things other terms like "effective axial force", "true wall tension", "apparent weight", "buoyancy method", "pressure area method" are introduced. "True wall tension" can again be referred to as "true force", "true tension", "real tension", "actual tension" or "absolute tension" and effective tension can sometimes be termed "fictive tension" or "absolute tension less end cap load". It has even been called "the effective weight of the effective mass". [1][2]

C. P. Sparks is the author often referred to when effective tension is mentioned in offshore standards. [1][3] In his book [1] he states that "*a suspended riser will see a buoyancy force equal to the weight of the fluid displaced, which for a vertical riser of uniform cross section is equal to the pressure x area ($p_e A_e$) acting at the riser lower end. Note, however, that the buoyancy force acts at the centroid of the submerged volume, at the midheight of the submerged length, not at the riser lower end*". [1] In this thesis we will review this statement and the effective tension concept and present how we can understand effective tension.

In practice a marine riser will never be perfectly vertical or stand firmly on the sea bottom. In fact it is very hard to perform measurements on a real riser determining the tension forces which are acting, as there is as so many different parameters in play (currents, waves, rig motion).

During this thesis we will only be looking at perfectly vertical bars of uniform cross section. We will only investigate the pressure area term ($p_e A_e$) in the effective tension equation presented by Sparks, thus the bars will be solid with no internal volume. This will exclude any form of stress induced by drilling mud inside the riser. We will also exclude any auxiliary lines. All cases are static cases.

In a newer book [4] the effective tension is described as; "The effective force is not a true force in that it cannot be measured with a strain gauge or weight indicator." Further on it is described as "The effective force is a fictitious quantity …". In this thesis we will try to show that by calculating the effective force one can predict failures. [4]

Chapter 2. Introduction to the oil and gas industry

Parts of chapter 2 have been used in an earlier project at the University of Stavanger. [5]

2.1 Offshore development

Oil has been used for over five thousand years. In the middle east oil seeped up from the ground. This oil was used for waterproofing boats and baskets, paint, provide light and even in medication. [6]

Whale oil has been used as source of light in recent times. But because of the increased use of whale oil the whale population dropped and this increased an already high oil price further. [6]

The demand for oil was then much higher than the supply and many companies and individuals were looking for a larger and more lasting source of the what-to-be-known-as “black gold”. The answer came with the development of drilling for crude oil onshore. The demand for oil did still rise and this led to the exploration companies to look for oil below the seabed. [6]

Prior to the Second World War drilling offshore was limited to shallow waters of Lake Maracibo, Venezuela and the swamps and coastal area of Louisiana in the US. After the Second World War there was a significant change in the oil industry as America was making its transition from a war-time to a peace-time economy. Until then the government had controlled the oil-price, but now the states started to dispute over the offshore shelf mineral rights. There was a large public demand for oil and gas, and the companies encountered challenges e.g. underwater exploration, weather forecasting, tidal and current prediction, drilling location determination and offshore communications. [6]

Despite these challenges, the first well was drilled from a fixed platform offshore out-of-sight of land in 1947. The combination of a barge and platform was a significant breakthrough in drilling-unit design for offshore use. This event marked the beginning of the modern offshore industry as it is known today. [6]

In the Gulf of Mexico, the first oil well structures to be built in open waters were in the water depths of up to 100m and based on a piled jacket structure, in which a framed template has piles driven through it to pin the structure to the sea bed. To this, a support frame was added for the working parts of the rig, such as deck and accommodation. These structures were the fore-runners for the massive platforms in many locations around the world, including the North Sea. [6]

There was a high activity level in the oil industry in the 1960s. Many new offshore oil and gas fields were discovered in the Gulf of Mexico, the Southern North Sea, the South China Sea and in the Gulf of Suez. [6]

Two “oil shocks” in the 1970s led to dramatic increases in oil prices and a perception that oil was in short supply. [6]

In the 1970s and the early 1980s there were years of unprecedented offshore activity. The high oil prices and a perceived need to increase security of oil supplies facilitated the installation of giant platforms in the hostile waters of the Northern North Sea and offshore Alaska, and in the Gulf of Mexico. [6]

In 1986 a collapse in the oil price put the future of such offshore field developments into question. As a response the industry came up with innovative solutions that enabled new developments to go ahead. The oil price was low for almost twenty years, but after 2004 there has been a significant increase in both oil and gas prices. There has also been a significant increase of costs for exploration, development and operation after 2004. Safety and environmental issues has also been a growing concern which the industry has been facing. [6]

The industry has in recent times been looking towards smaller fields, often with complex geology, and remote and frontier areas, for example deep water. This is because it is not likely that there will be discovered many new “giant” offshore fields although large volumes of oil and gas lies offshore. To develop these resources economically, the industry has to find new solutions that combine cost-effectiveness over the lifetime of the project with improved safety and environmental performance. [6]

2.2 Current activities and trends

The offshore oil and gas industry has for the last few decades developed very high activity in the North Sea, the Gulf of Mexico, the South China Sea, offshore Brazil and offshore West Africa. The North Sea has been the largest producing region of the offshore oil industry. There are a few deep water fields being developed offshore Norway and West of Shetland, but most reservoirs are in water less than 200 meters deep. Development of satellite fields is a major feature in this region – generally small accumulations of oil and gas which lie close to existing production facilities. [6]

The current focus of the industry includes the continental shelf of Gulf of Mexico, offshore Brazil and West Africa (e.g. Nigeria and Angola) where water depths reach some 3000 meters. Many of the record breaking developments in deep offshore drilling have been in these locations. [6]

“Frontier” areas such as offshore Alaska and the Barents Sea are also in the focus of the industry. Remoteness, deep water, high winds, floating ice and sub-zero temperatures are some of the challenges the industry faces in these areas. Some arctic regions are frozen up to 10 months of the year, putting severe limitations on the drilling activities. [6]

Western companies have increased their interest in the former USSR’s hydrocarbon resources after the opening of the country. More than a fifth of the world’s offshore oil and gas resources could be located here according to some estimates. To date only a small part of the area has been explored. In the Sea of Okhotsk north of Japan and in the Barents and Kara Seas in the Russian Arctic there have been a number of oil and gas field discoveries. [6]

2.3 Deepwater development challenges and trends

Expanding into deepwater frontier areas with no existing infrastructure has always been a challenge. [6]

Many achievements, which are comparable with the space industry, have lead to the possible development of the offshore oil industry in hostile waters. Many fields are located far from land and operations are extended to even more remote locations. New fields are being explored in ever deeper and harsher waters, like the Norwegian Sea, the Atlantic Ocean west of Scotland and the Barents Sea. [6]

2.4 Drilling

A marine drilling riser system is used during a drilling operation offshore. The drilling process can differ from operation to operation but the basic steps in each operation can be summarized as shown in figure 2.1:

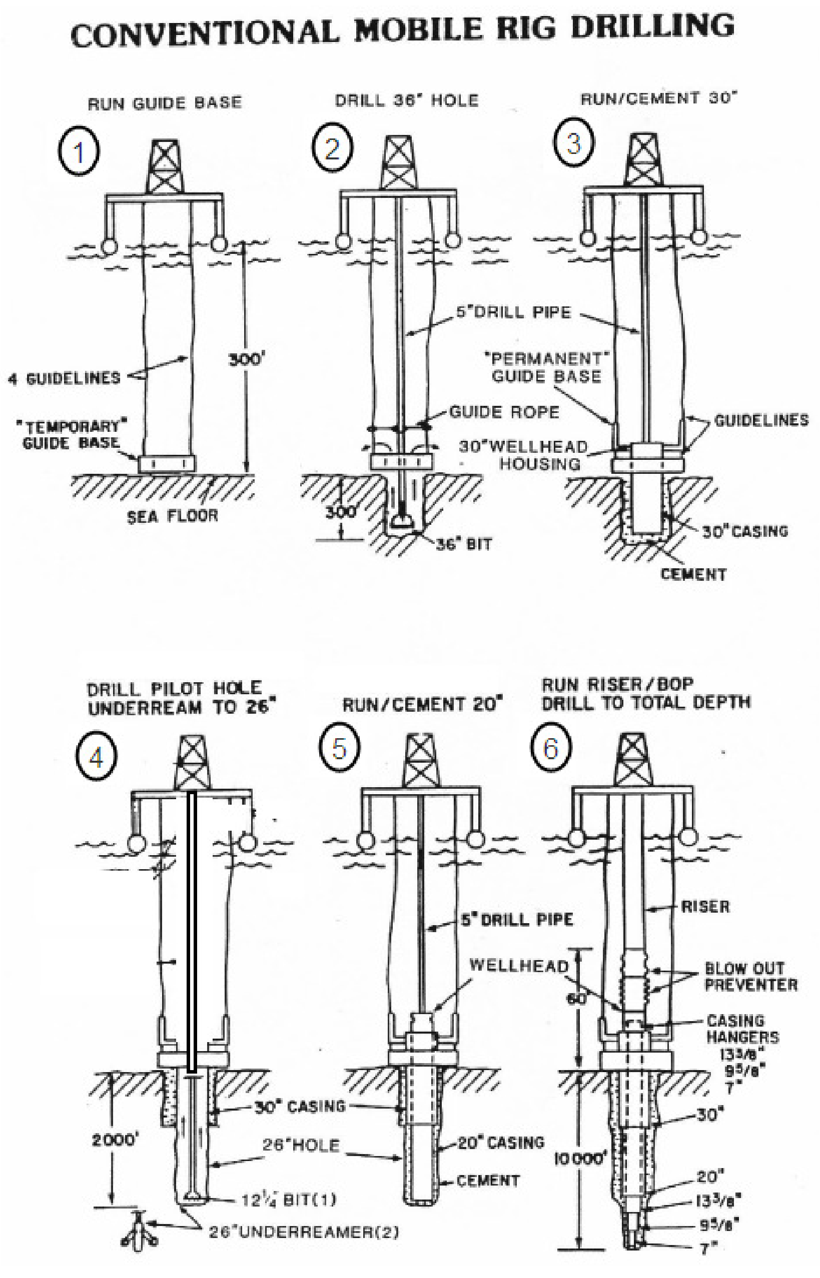


Figure 2.1: Generalized steps of a drilling operartion [7]

- Step 1: Lower a temporary guide base which is connected to 4 guide wires. This will help guide different equipment in the following steps.
- Step 2: Start drilling through the seabed with a 30” or 36” drill bit to around 120m. This drilling sequence is done without a riser. Use a rope to guide the drill pipe in place as shown in figure 2.2

- Step 3: After the drilling in step 2 is completed the 30” conductor will be lowered into the hole and cemented. The upper part of the conductor is the wellhead housing. A permanent guide base is installed.
- Step 4: A 26” hole is drilled to approximately 500m.
- Step 5: The 20” casing with wellhead is lowered and cemented in place.
- Step 6: The subsea Blow-out Preventer (BOP), Lower Marine Riser Package (LMRP) and the marine riser is installed and the drilling continues. Drill mud will now circulate through the marine riser bringing the drill cuttings up to the platform where it is removed and the mud is pumped down the well again. The BOP, LMRP and marine riser will be connected to the wellhead for the rest of the drilling operation. [6][8]

2.5 Marine riser system

Marine risers were first used to drill from barges offshore California in the 1950s. In 1961 an important landmark occurred when drilling took place from the dynamically positioned (DP) barge CUSS-1. Since those early days, risers have been used for four main purposes [1]:

- Drilling
- Completion/workover
- Production/injection
- Export

In each of the four groups of risers there is a large variety of details, dimensions and materials. In this thesis we will focus only on the drilling riser. Drilling risers can be divided into low-pressure and high-pressure risers. Their difference is discussed in section 2.6.1 and 2.6.2. [1]

2.5.1 Low-pressure drilling riser

A low-pressure riser is open to atmospheric pressure in the top end. This is the standard drilling riser used today. Because it is open in the top end the internal pressure can never be higher than that owing to the drilling-mud weight. Drilling risers are made of up joints which can vary between 15-23 meters (typically) in length. The nominal diameter of the central tube is usually 21” and on the outside the central tube is equipped with several auxiliary lines, see figure 2.2. If the BOP is closed due to a kick in the well the kill and choke lines are used to communicate with the well and to circulate fluid. The booster line is used to inject fluids in the lower end of the riser to accelerate the drill cutting flow back to the surface. The two small auxiliary lines shown in figure 2.2 can be hydraulic lines, which is used to power the subsea BOP. [1]

Figure 2.2: Drilling risers with buoyancy modules to the left - 3D figure of the riser, drill string and kill/choke/auxiliary lines to the right [9][10]

Riser joints can be fitted with syntactic foam buoyancy modules to increase the buoyancy force which reduces the weight of the joint in water, see figure 2.2. The upper part of a drilling riser is usually fitted with modules with the exception of the splash zone. This to reduce the hydrodynamic forces the waves will induce on the riser in this area. The design pressure for the modules increases with the water depth implying a stronger, heavier and denser design. This favors installing buoyancy modules at the top end of the riser. [1]

Air-can buoyancy modules have been used in the past. The advantage with these was that they could be optimized for each individual drilling campaign. The disadvantage was that they added a level of complexity. [1]

The connector in the top and the bottom of the riser joint is another feature which can have many different designs. [1]

The auxiliary lines are supported by guide clamps. The design of the guide clamps is critical because they prevent the lines from buckling. Another good practice which ensures that the safety level is kept is to design the lines so they cannot break out of their housing at the connector level, even if the line should buckle. [1]

2.5.2 High-pressure drilling riser

A high-pressure drilling riser is used when the BOP is located at the surface, as was the case for the CUSS-1 in 1961. In the event of a kick the BOP is accessible for closing from the drill rig and no subsea choke and kill lines are necessary. Because of the lack of choke and kill lines along the submerged riser the architecture is much simpler than a low-pressure riser. The riser must be designed to take the full well pressure. However, when drilling with a surface BOP there Is potentially more risk, unless an adequate seal and disconnect system can be provided in case of an emergency. [1]

High pressure risers with surface BOPs have been used to drill from many tension leg platforms since the 1980s, such as Hutton (1984), Heidrun, Mars, RamPowell, and URSA, and from some spars. In the case of the Heidrun TLP, the drilling riser was made of titanium. [1]

In the early 1990s a high-pressure slimline (small-diameter) riser with surface BOP was proposed for the Ocean Drilling Program, to allow scientific drilling with mud circulation in ultra deep water (>4000 m). But the project was not pursued. High-pressure risers with surface BOPs have more recently been used to drill a large number of wells from semisubmersibles in moderate environmental conditions. The concept continues to be developed for deeper water and harsher environments. [1]

2.6 Components of a Marine Drilling Riser System

The marine drilling riser system is a continuation of the well bore from the seabed to the surface. It connects the subsea BOP Stack to the drilling vessel. [11]

According to the American Petroleum Institute (API) specification 16F [11] the main function of the marine riser system is to:

- Provide for fluid communication between the drilling vessel and the BOP Stack and the well:
  - Through the main bore during drilling operations
  - Through the choke and kill lines when the BOP Stack is being used to control the well
  - Through the auxiliary lines such as hydraulic fluid supply and mud boost lines
- Guide tools into the well
- Serve as a running and retrieving string for the BOP Stack

The main components of a marine riser are shown in figure 2.3.

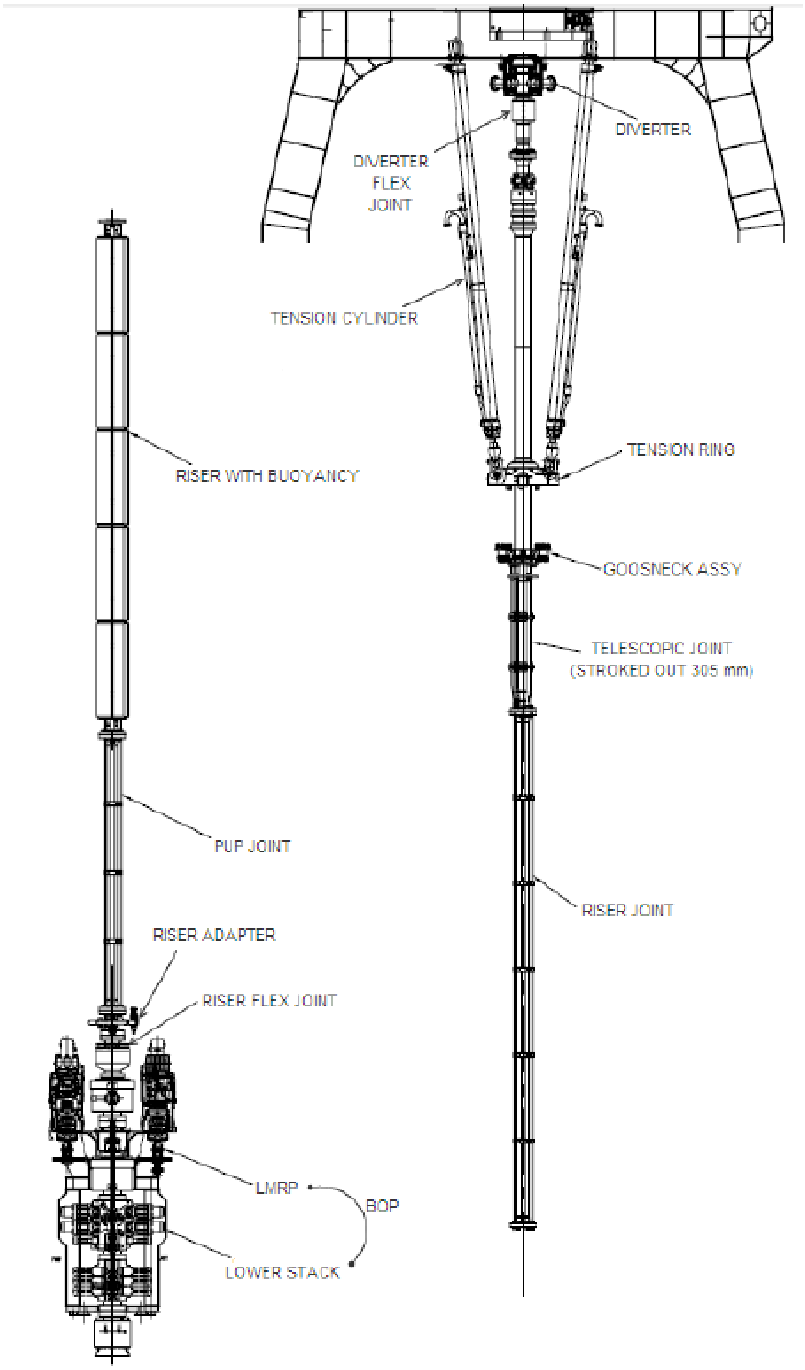


Figure 2.3: Main components of a marine riser [8]

2.6.1 Upper Marine Riser Package (UMRP)

The upper part of the riser string including the riser tensioner system is called the UMRP. [12] The UMRP includes:

- The diverter system

- Upper flex joint

- Self-tensioned slip joint (telescopic joint) and tensioner ring

- Riser rotation bearing joint

2.6.1.1 Diverter
The diverter diverts the drill mud and cuttings from going vertically up from the riser system and potentially be blown out on the drill floor in case of a kick, routing the fluid horizontally out and into flowlines connected to the mud system. [13]The surface diverter is mounted on top of the UMRP but it is not part of the marine riser system. [12]

2.6.1.2 Upper Flex/ball joint
The upper flex joint or ball joint is positioned on surface level in between the diverter and the inner barrel of the slip joint. The joint allow some misalignment of angle between the riser system and the drilling vessel (roll pitch and offset motions of the vessel). [12]

2.6.1.3 Telescopic joint and tensioner ring
The telescopic joint, or the slip joint as it is often named, consists of an outer and inner barrel. The outer barrel is attached to the riser string and is held in tension by wire ropes or hydraulic cylinders from the top end of the outer barrel to the tensioners. [13] The inner barrel is connected to the upper flex/ball joint and can move freely inside the outer barrel to compensate for the drilling vessels horizontal and vertical movement. Between the outer and inner barrel there is a packer element which seals of the annulus, this prevents fluid leakage from the riser. On the top end of the outer barrel there is typically mounted or incorporated a tensioner ring. The main function of the tensioner ring is to transmit the support load from the riser tensioner lines to the outer barrel. Some tensioner ring systems also allow for rotation of the vessel around the riser. The telescopic joint usually have terminal fittings for connecting the choke, kill and auxiliary line drape hoses to the rigid lines used on the riser joints. [12]

2.6.1.4 Riser Rotation Bearing Joint
The riser rotation bearing joint is mounted at the bottom of the telescopic joint. It allows the drilling vessel to rotate around the riser's vertical axis and minimizes the torque transferred from the riser to the telescopic joint. It typically consists of a roller bearing system, built in locking device and hydraulic motors. The hydraulic motors and the built in locking device is used for precise rotational control and preventing inappropriate rotation of the riser. [12]

2.6.2 Riser joints
See chapter 2.5.1

2.6.3 Lower Marine Riser Package (LMRP)
The lower marine riser package is an assembly located at the bottom of the drilling riser, but above the BOP. The LMRP provides releasable interface between the riser and BOP stack. [12]

Typical component in a LMRP are: [11]

- Lower Riser Adapter
- Flex/ball joint bypass lines
- Lower flex/ball joint
- Hydraulic connectors for mating the riser to the BOP stack

2.6.3.1 Lower Riser Adapter
The lower riser adapter is the connection between the lower most riser joint and the lower flex/ball joint mounted on the lower marine riser package. [11]

2.6.3.2 Flex/Ball joint bypass lines

The bypass lines are mounted on kick outs on the riser adapter. They bypass the flex/ball joint and terminate in the BOP. [11]

2.6.3.3 Lower flex/ball joint

See section 2.7.1.2 as upper and lower flex joint is basically the same. [11]

Chapter 3. Effective Tension Concept

To be able to present the effective tension concept correct, section 3.1 is solely based on chapter 2 and chapter 3 from the book "Fundamentals of Marine Riser Mechanics" [1] written by C. P. Sparks.

In section 3.2 we will discuss the concept based on the assumptions mentioned in section 1.2.

3.1 Effective Tension

"All codes of practice require the global behavior of pipes and risers to be calculated using effective tension. This is generally defined in one of two ways: [1]

- *Some codes quote equation (3.1)*
- *Some codes mention that effective tension is the axial tension calculated at any point of the riser by considering only the top tension and the apparent weight of the intervening riser segment."*

$T_e = T_{tw} - p_i A_i + p_e A_e$ *(3.1)*

Where

$T_e = Effective\ tension$

$T_{tw} = True\ wall\ tension\ in\ air\ at\ point\ of\ interest$

$p_i = internal\ fluid\ pressure\ at\ point\ of\ interest$

$A_i = internal\ cross - sectional\ area$

$p_e = External\ fluid\ pressure\ at\ point\ of\ interest$

$A_e = External\ cross - sectional\ area$

To calculate needed top tension in a riser system to prevent buckling Sparks introduces "The Effective Tension Concept". This concept includes the influence of the tension in the riser walls, internal and external pressure and the weight of the pipe. [1]

3.1.1 Archimedes' Law

"Archimedes' Law states in its most general form states that when a body is wholly or partially immersed in a fluid, it experiences an upthrust equal to the weight of fluid displaced. This is illustrated in figure 3.1, in which a body is shown fully immersed in a fluid". [1]

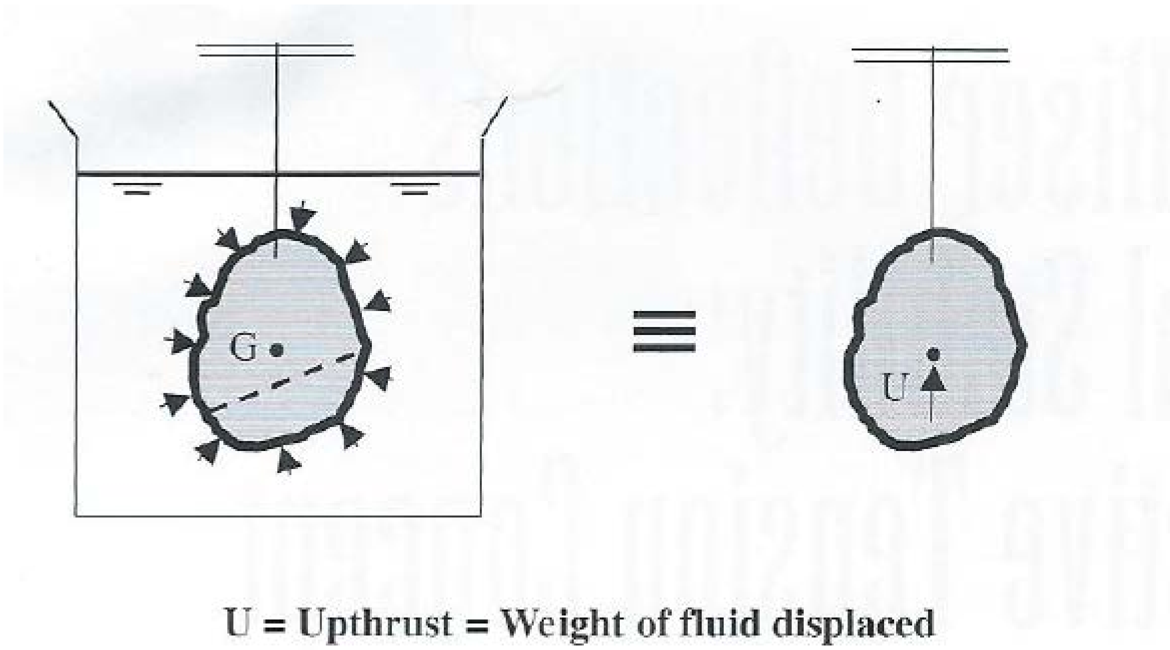


Figure 3.1: Archimedes' Law [1]

The argument taught to school children is that the pressure field is just able to maintain the displaced fluid in equilibrium, as shown in figure 3.2. Thus, it must provide an upthrust U equal to the weight of the fluid displaced W_f. Furthermore, since this upthrust can produce no rotation, it must act at the centroid of the displaced fluid, which is also the center of gravity G. Hence, it will also act at the centroid of the submerged body. [1]

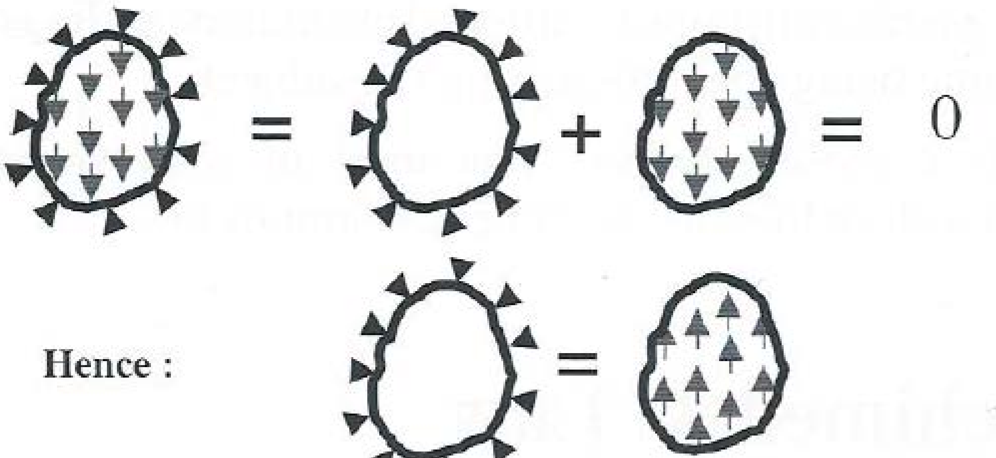


Figure 3.2: Pressure and weight acting in a fluid [1]

Thus, if the true weight of the body is W_t, the tension in the string will be given by the following, where $W_t - W_f$ is generally called the apparent weight W_a: [1]

$T = W_t - U = W_t - W_f$ *(3.2)*

There are a number of important points to make about Archimedes' Law: [1]

- *The law can be applied directly only to pressure fields that are completely closed. Note that for a suspended or floating body, the pressure field appears not to be closed; however, since the pressure at the surface is zero, the field can be considered to be closed.*
- *The law cannot be applied directly to parts of submerged bodies, such as that below the dotted line in figure 3.1.*
- *The law says nothing about internal forces or stresses.*
- *The closed pressure field, when combined with the distributed weight of the displaced fluid, can produce no resultant moment. The fluid would not be able to support the associated stresses."*

3.1.2 Archimedes' Law by superposition

Archimedes' Law can also be deduced by superposition. This may be too abstract for school children, but leads to the same results more clearly and directly. Since superposition will be used extensively in this book, it will be first used here to rederive Archimedes' Law. [1]

In figure 3.3, the two systems shown (the submerged body and the displaced fluid) are both in equilibrium under the combine loads that include the effects of tension, pressure, and weight. Hence, if the two systems are superimposed and the forces on the displaced fluid are subtracted from those on the submerged body, the resulting equivalent system will also be in equilibrium. [1]

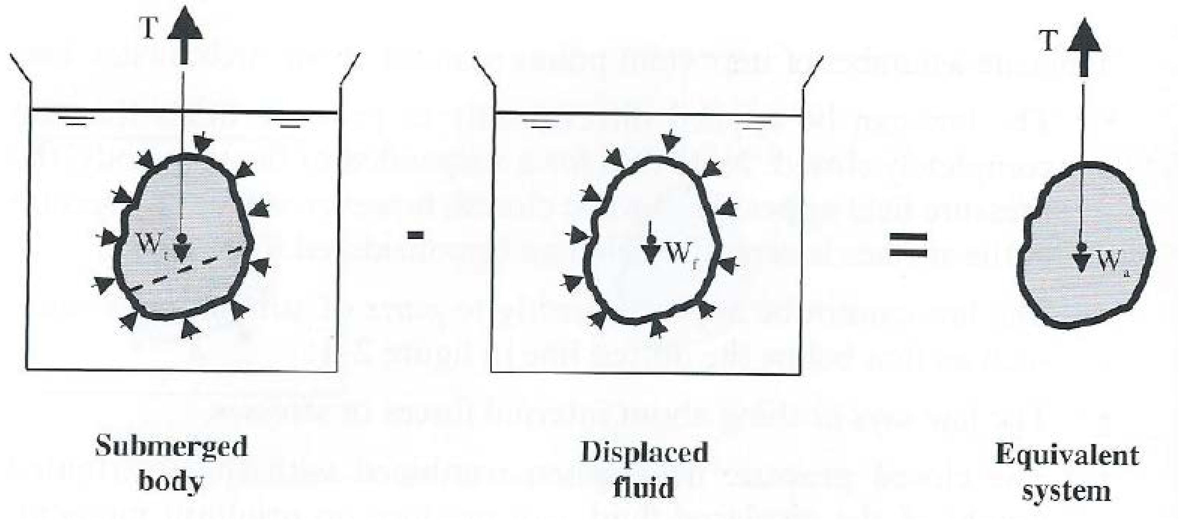


Figure 3.3: Archimedes Law by superposition [1]

Superposition of the two systems allows the identical pressure fields to be eliminated. All that remains in the resulting equivalent system is the tension T in the string and the apparent weight W_a, which is then simply the difference between the weights of the submerged body and the displaced fluid, as given by equation (3.3). [1]

$W_a = W_t - W_f$ *(3.3)*

Any two systems can be superimposed in this way. The only requirement is that they both be in equilibrium. In the preceding, there is no need to specify that densities must be constant or that the upthrust acts at the centroid of one or other of two systems. The argument can be applied directly to cases where the submerged body does not have a constant density; where the body is suspended *across the interface between fluids of different densities, or where the density of the displaced fluid may vary vertically according to some law. As long as the displaced fluid segment represents exactly the fluid displaced by the submerged body, superposition can be used directly. [1]*

3.1.3 Internal forces in a submerged body

In the calculation of the internal forces on a part of a submerged body, the problem is to take into account the pressure filed that is not closed. Figure 3.4 shows the forces acting on the segment below the dotted line in figures 3.1 and 3.3. The resultant of the pressure filed acting on the underside of the segment is unknown and cannot be determined directly using Archimedes Law. [1]

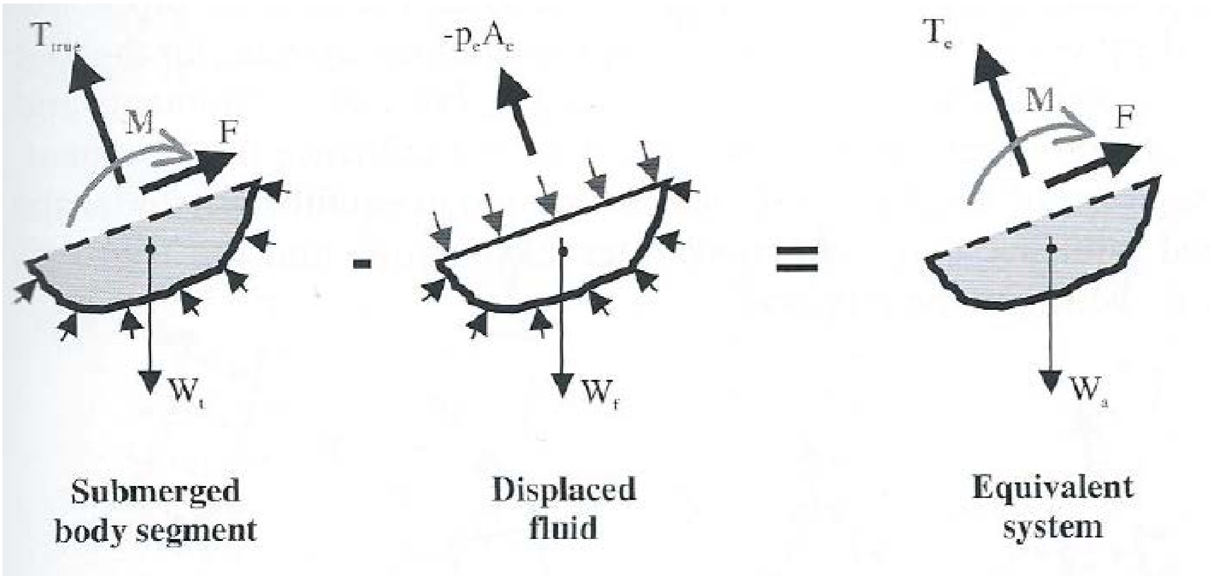


Figure 3.4: Internal forces acting on a submerged body segment [1]

Nevertheless, superposition allows the internal forces to be determined very simply. The middle sketch of figure 3.4 shows the forces acting on the displaced fluid segment including the closed pressure field. If these forces are subtracted from the forces on the body segment, the pressure field acting below the body is conveniently eliminated. However, the force $p_e A_e$, owing to the pressure acting on the section, remains (where p_e is the pressure in the fluid and A_e is the cross-sectional area of the section). Since convention requires tension to be positive, this must be shown as a tensile force: $-p_e A_e$. [1]

The equivalent system (figure 3.4, right-hand sketch) shows the resultant of the superposition. Once again, the apparent weight W_a is given by equation (3.3), where the weights W_t, W_f and W_a correspond to the segment, rather than the whole body. Thus, the apparent weight W_a is in equilibrium with an effective tension T_e, a shear force F, and a moment M, which can be found by resolving forces normal and parallel to the section and by taking moments. The shear force F and the moment M are the same as on the body segment. (For the applications considered in this book, the minute moment created by the very slight pressure gradient across the section can be neglected.) The effective tension T_e is then related to the true tension T_{true} as shown in equation (3.4). [1]

$T_e = T_{true} - (-p_e A_e) = T_{true} + p_e A_e$ *(3.4)*

According to convention, tensile forces are positive. However, according to a further convention, pressures are also positive. The positive sign in the right-hand side of equation (3.4) results from the *contradiction between the two conventions. The effective tension T_e is nevertheless the difference between the tensions acting on the body segment and the displaced fluid segment, just as apparent weight is the difference between their weights". [1]*

3.1.4 Curvature, Deflections and Stability of Pipes and risers under pressure

"The preceding arguments can be extended to the case of pipes and risers under pressure. Figure 3.5 shows equivalent force systems for the case of a pipe subjected to internal pressure p_i. For clarity, moments and shear forces have been omitted, but that does not influence the argument. A pipe segment of length δs is shown curved and in equilibrium under the combined influence of pipe weight, internal pressure, and the true wall tension T_{tw} acting on the pipe wall. [1]

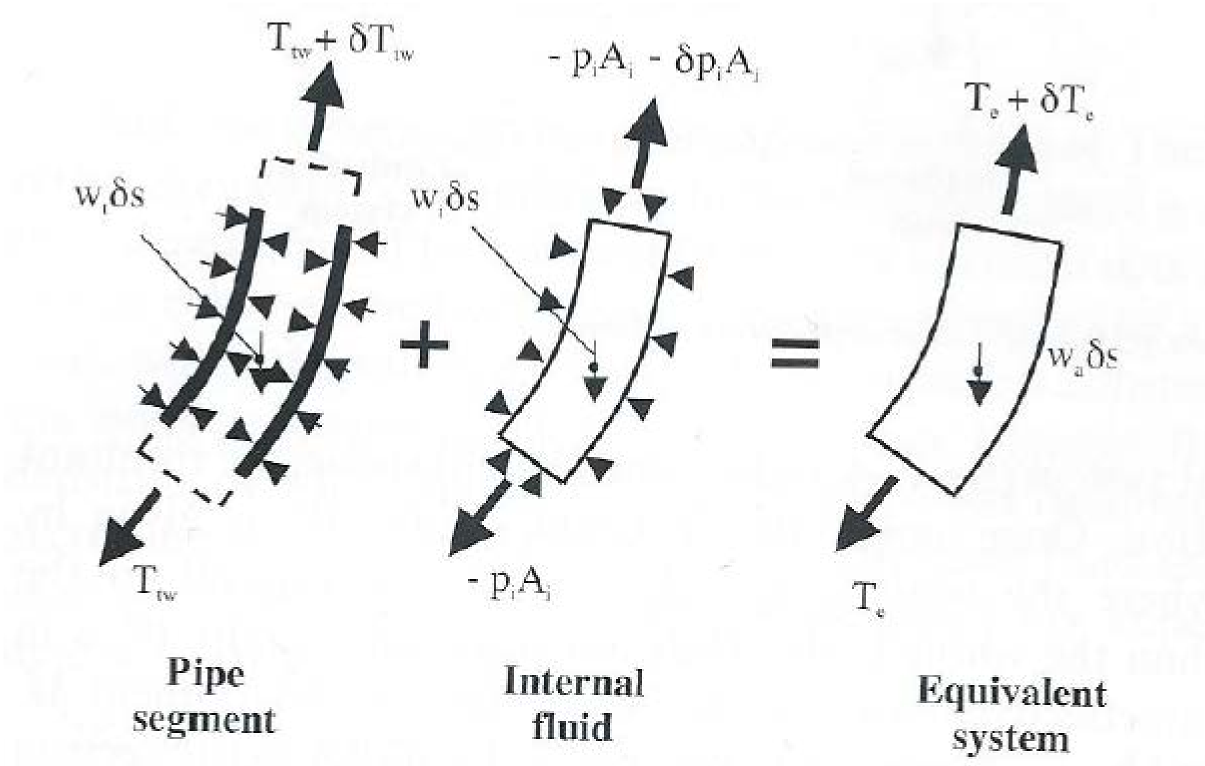


Figure 3.5: Pipe with internal fluid - equivalent force system [1]

The pressure field acting on the internal fluid column is closed and in equilibrium with the weight of the internal fluid. The lateral pressures acting on the pipe wall are equal and opposite of those acting on the internal fluid. Hence, by superposition and addition of the two force systems, those lateral pressures are eliminated. However, the axial "tension" in the fluid column $-p_iA_i$ remains (where p_i is the internal pressure and A_i is the internal cross-sectional area of the pipe). This leads to the equations for the effective tension T_e and apparent weight w_a of the equivalent system: [1]

$T_e = T_{tw} + (-p_iA_i)$ (3.5)

$w_a = w_t + w_i$ (3.6)

When external pressure p_e is also present, the same approach can still be used, as shown in figure 3.6. By the addition of the force systems acting on the pipe segment and the internal fluid and then the subtraction of the force system acting on the displaced fluid, all lateral pressure effects are *eliminated. In figure 3.6, w_t, w_i, w_e and w_a are the weights per unit length of the tube, the internal fluid column, the displaced fluid column and the equivalent system, respectively. [1]*

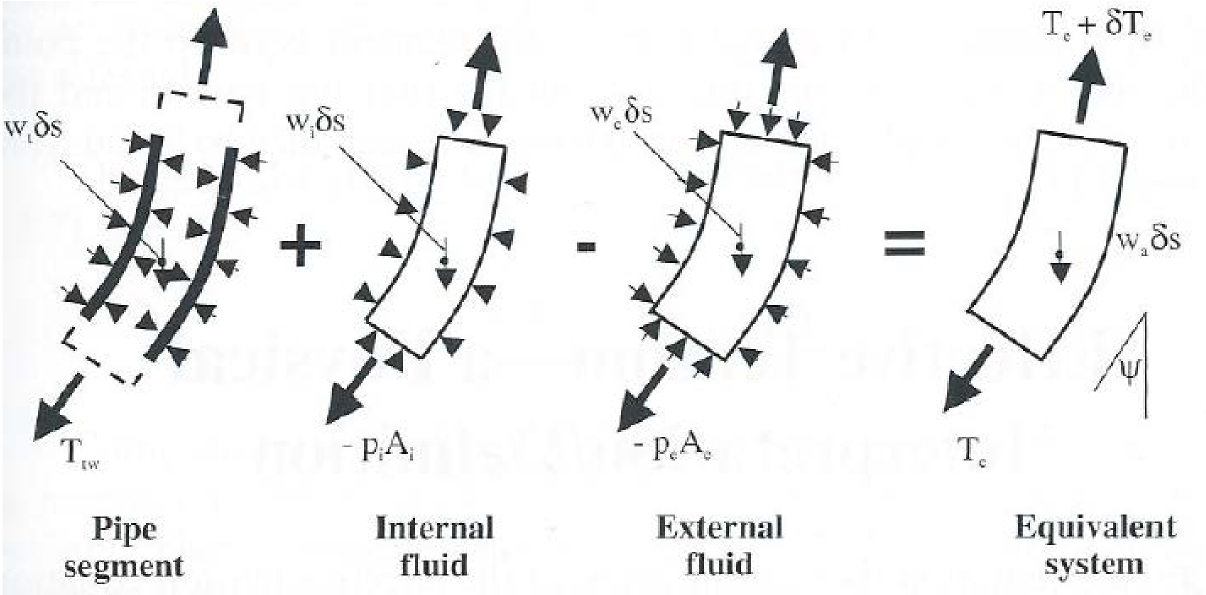


Figure 3.6: Pipe with internal and external fluids - equivalent force systems [1]

The equations for the effective tension T_e and the apparent weight w_a then becomes

$T_e = T_{tw} + (-p_i A_i) - (-p_e A_e)$ *(3.7)*

$w_a = w_t + w_i - w_e$ *(3.8)*

Furthermore, the two concepts are related, as can be seen from the right-hand sketch in figure 3.6. For an element of length δs, resolution of forces in the axial direction gives

$\frac{dT_e}{ds} = w_a \cos \Psi$ *(3.9)*

which for small angles with the vertical becomes $\frac{dT_e}{ds} = \frac{dT_e}{dx} = w_a$. [1]

Since for any fluid the combined effects of its weight and enclosed pressure field can produce no resultant moment anywhere, the bending effects of forces on the equivalent system are precisely the same as those on the pipe segment. Therefore, the simplest way to take into account the effects of internal and external pressure on pipe or riser curvature, deflection and stability is to use effective tension and apparent weight in the corresponding tensioned-beam calculations. [1]

The effective tension, at any point along a riser, can be obtained most simply by considering the equilibrium of the segment between the point and the riser top end, taking into account the riser top tension and the segment apparent weight. The true wall tension T_{tw} can then be found from equation (3.7)". [1]

3.1.5 Confusion regarding buoyancy

In chapter 3, "Application of Effective Tension – Frequent Difficulties and Particular Cases", of Sparks' book [1], he again discusses buoyancy;

"The Archimedes upthrust, or buoyancy, acting on a submerged body was recalled at the beginning of chapter 2. It is a volumetric force in the sense that it is the resultant of the closed pressure field acting on the enclosed volume. It is equal to the weight of fluid displaced by the body and acts at the centroid of the submerged volume. The concept can be applied to any fully submerged body. It can also be applied to any suspended or floating body – such as a suspended riser or a ship hull – for which the pressure fields can be considered to be closed. For a suspended riser, the buoyancy is equal to the weight of fluid displaced, which for a vertical riser of uniform section is equal to the pressure x area ($p_e A_e$) acting at the riser lower end. Note, however, that the buoyancy force acts at the centroid of the submerged volume, at the midheight of the submerged length, not at the riser lower end. Ships would capsize if buoyancy acted at the keel level instead of at the centroid of their displaced volume. [1]

Confusion arises when discussing the buoyancy of part of a submerged object (see figure3.4), such as a segment of riser, since it is subject to a pressure field that is not closed. The confusion is particularly flagrant if the riser pipe concerned is vertical and of uniform section. The wall of such a riser is continuous. Hence, the fluid pressure will act only horizontally on the segment and will have no vertical component. It is tempting to say that such a riser segment has no buoyancy. Since the segment can be positioned anywhere along the riser length, that would imply that the entire riser has no buoyancy, except at the surface at the lower end. That plainly does not agree with Archimedes' conception of buoyancy as an upthrust acting on a submerged volume. [1]

The confusion is further increased when considering the stability of a vertical uniform riser connected to the sea bed, since the external fluid pressures the do not even apply a vertical force to the surface at the riser lower end! Yet if the riser has negative apparent weight (i.e., is lighter than water), it will remain vertical and stable even if the top tension is reduced to zero. How does it do that without collapsing in a heap on the seabed? Some would argue that if the riser did depart from vertical, forces with vertical components would be generated which would return it to the vertical". [1]

3.2 Effective tension - Discussion

Equation (3.7) and (3.8) derived by Sparks will in this thesis be reduced to equation (3.11) and (3.12) since the annulus and drilling mud is excluded due to the assumptions made in section 1.2.

$T_e = T_{tw} - (-p_e A_e)$ (3.11)

$w_a = w_t - w_e$ (3.12)

The required top tension T to prevent the riser from buckling at any point (lower end in practice), considering the whole length of the riser and the effective tension equal to zero at the bottom of the pipe, will then become

$T = w_a = w_t - w_e = w_t - p_e A_e$ (3.12)

Where

$T = Minimum\ top\ tension\ to\ prevent\ buckling$

$w_a = apparent\ weight\ of\ riser$

$w_t = weigth\ of\ riser\ in\ air$

$w_e = weight\ of\ displaced\ fluid$

$p_e = external\ pressure\ at\ lower\ end\ of\ riser$

$A_e = external\ cross - section\ at\ lower\ end\ of\ riser$

Comparing equation (3.12) with equation (3.13) given by API 16 Q [14] to calculate the minimum slip ring tension

$T_{SRmin} = W_s f_{wt} - B_n f_{bt} + A_i(d_m H_m - d_w H_w)$ (3.13)

Where

$T_{SRmin} = Minimum\ Slip\ Ring\ Tension\ (to\ avoid\ buckling\ at\ point)$

$W_s = Submerged\ Riser\ Weight\ above\ considered\ point$

$f_{wt} = Submerged\ Weight\ Tolerance\ Factor\ (min$ 1,05)

$B_n = Net\ Lift\ of\ Buoyancy\ Material\ above\ considered\ point$

$f_{bt} = Buoyancy\ loss\ and\ Tolerance\ Factor\ (max$ 0,96)

$A_i = Internal\ Cross\ Section\ Area\ of\ Riser\ and\ aux.lines$

$d_m = Drilling\ Fluid\ Weight\ Density$

$H_m = Drilling\ Fluid\ Column\ above\ considered\ point$

$d_w = Sea\ Water\ Weight\ Density$

$H_w = Sea\ Water\ Column\ above\ considered\ point$

Note that W_s can be written as

$W_s = w_t - (A_e - A_i)d_w H_w$ (3.14)

Where

$w_t = Riser\ Weight\ in\ air\ above\ considered\ point$

$A_e = External\ Cross\ Section\ \ Area\ of\ Riser\ and\ aux.lines$

And by combining equation (3.13) and (3.14) and neglecting the tolerances and lift from buoyancy material we get

$T_{SRmin} = W_t - (A_e - A_i)d_w H_w + A_i(d_m H_m - d_w H_w) = W_t - A_e d_w H_w + A_i d_m H_m$ (3.15)

Equation (3.15) is equal to equation (3.12) if we consider a solid riser with no internal cross section;

$T_{SRmin} = w_t - A_e d_w H_w = w_t - p_e A_e$ (3.16)

It is easy to show that $p_e A_e$ at the lowest point of the riser is numerical the same as if we would calculate the buoyancy force.

In chapter 4 we will take a closer look on how buoyancy can be understood and define what a buoyancy force is. This definition will show that the term $p_e A_e$ in equation (3.16) is not a buoyancy force.

Chapter 4. Buoyancy

In this chapter we will take a closer look on how buoyancy is introduced in modern day physics. We will then review two different methods to interpret buoyancy by calculating buoyancy as pressures on flats, and by calculating buoyancy as weight of displaced volume. Different experiments trying to show that one school is right and the other school is wrong will be presented. In the end of the chapter we will define how a buoyancy force will be interpreted in the rest of this thesis.

Buoyancy is widely used and understood throughout the world. But as shown in this chapter it also generates a good topic of discussion. First we will look at Archimedes' Principal describing buoyancy.

4.1 Archimedes' Principle

Some 2000 years ago the great Archimedes defined how buoyancy works on an object immersed or partly immersed in a fluid. Archimedes did consider several cases, or propositions; this includes a solid which is lighter than the fluid it is immersed in, a solid which is equal in weight as the fluid it is immersed in and finally a solid which is heavier than the fluid it is immersed in. [15]

The propositions Archimedes wrote: [15]

- "Proposition 3: Of solids those which size for size, are of equal weight with a fluid will, if let down into the fluid, be immersed so that they do not project above the surface but do not sink lower"
- "Proposition 4: A solid lighter than a fluid will, if immersed in it, not be completely submerged, but part of it will project above the surface"
- "Proposition 5: Any solid lighter than a fluid will, if placed in the fluid, be so far immersed that the weight of the solid will be equal to the weight of the fluid displaced"
- "Proposition 6: If a solid lighter than a fluid be forcibly immersed in it, the solid will be driven upwards by a force equal to the difference between its weight and the weight of the fluid displaced"
- "Proposition 7: A solid heavier than a fluid will, if placed in it, descend to the bottom of the fluid, and the solid will, when weighed in the fluid, be lighter than its true weight by the weight of the fluid displaced"

All the propositions listed above have been proved and accepted. A mathematical explanation has been derived and follows the same propositions exactly.

Modern text books of physics could name the force mentioned in proposition 6 "buoyancy" [16][17], "buoyancy force" [18], "buoyant force" [19][20][21][22] or "upthrust" [1][23].

A modern way of describing buoyancy is by using a figure, like figure 4.1, and state that the buoyancy acting on the body partly or fully immersed in fluid is the net sum of the vertical forces (the horizontal forces are equal and opposite, canceling out each other) acting on the lower side of the body and the vertical forces acting on the top side of the body. The forces are naturally calculated from the pressure acting on the lower and upper part of the body; this is shown in equation (4.1).

$$B = \Delta F = F_l - F_u = A * \Delta p = A * (p_l - p_u) = A * h * \rho_{fluid} * g \quad (4.1)$$

This is known as the piston-force understanding of buoyancy.

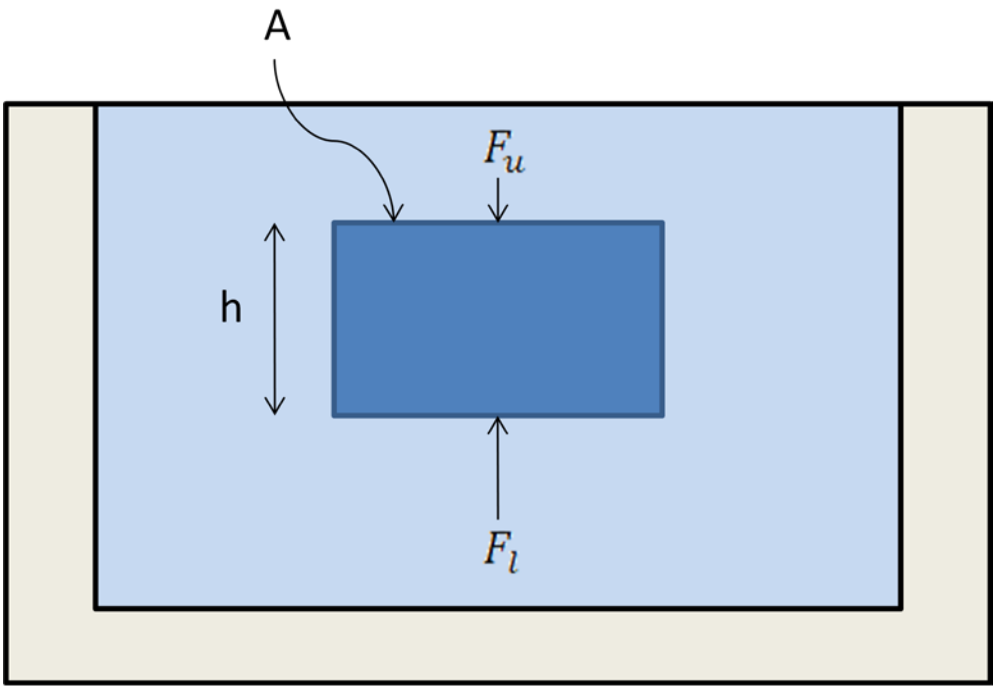


Figure 4.1: Body immersed in fluid, showing only the vertical forces from the fluid

A different way to describe the same buoyancy force is to state that the buoyancy force is equal to the volume of the displaced fluid multiplied with the fluids density. The mathematical expression is shown in equation (4.2).

$B = V_{displaced\ \ fluid} * \rho_{fluid} * g$ (4.2)

This is known as the volume understanding of buoyancy.

It is simple to show that equation (4.1) is equal to equation (4.2). These two different methods of calculating buoyancy will almost always give the same answer when calculating the buoyancy force acting on a body.

It will also be stated that the buoyancy force acts through the center of buoyancy, which is equal to the center of gravity of the displaced fluid.

In the physic text books investigated in this thesis both the above approaches are used. [16][17][18][19][20][21][22][23]

4.2 Archimedes's Principle in practice

To prove Archimedes' Principle many different experiments can be conducted. One of the simplest but most useful experiments which illustrate the effect of buoyancy is shown in figure 4.2.

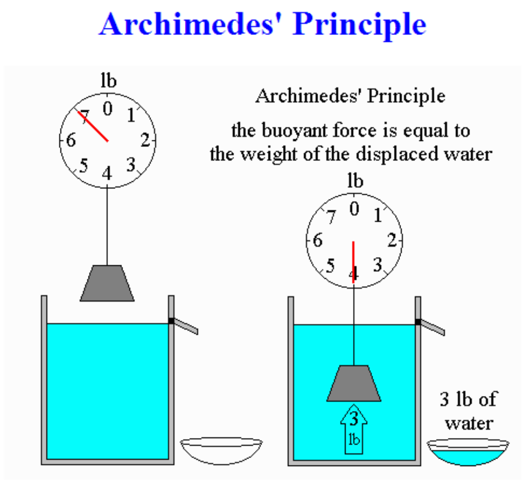


Figure 4.2: Experiment showing Archimedes' Principle [24]

The body hanging in the air has a weight of 7 lb as shown by the left hand sketch of figure 4.2. When the body is lowered into the beaker it will displace water (the water level will raise). The water that the body displaces will be drained into the bowl next to the beaker. When the body is completely immersed in fluid, a volume of water equal to the volume of the body will be in the bowl. As the weight shows in the right-hand sketch of figure 4.2 the body, when immersed in fluid, is only weighing 4 lb. The figure also shows that the water in the bowl is weighing the same as the "missing" 3 lb, or in other words; the buoyancy force is equal to the weight of the displaced fluid.

But if the object is standing firmly on the ground, with no fluid between the ground and the object, will there be any buoyancy force acting on it?

Using the volume interpretation of Archimedes Principle will give a buoyancy force equal to the displaced fluid. That the object is still displacing fluid is evident. But by using the piston force approach one would get a negative buoyancy force, or a force directed downwards.

In the following of this section we will review a couple of experiments or thought experiments used by recent papers dealing with this problem. We will also show an experiment conducted by Goins in 1980 and an experiment conducted at the University of Stavanger during this master project.

4.2.1 Experiment 1: "Do fluid always push up objects immersed in them?" [25]

Figure 4.3 shows a hands-on experiment which is used to show that the buoyancy force will "disappear" if there is no fluid underneath the object immersed in fluid. Figure 4.3 a) shows a glass box filled with water. On the bottom of the glass box there is a glass prism whose upper surface is optically polished. Below the surface of the water there is floating a table tennis ball with an attached thin polished glass plate. When the ball is pushed down and makes contact with the prism, as shown in figure 4.3 b), the buoy remains standing although there is an upwards force still acting on the ball. [25]

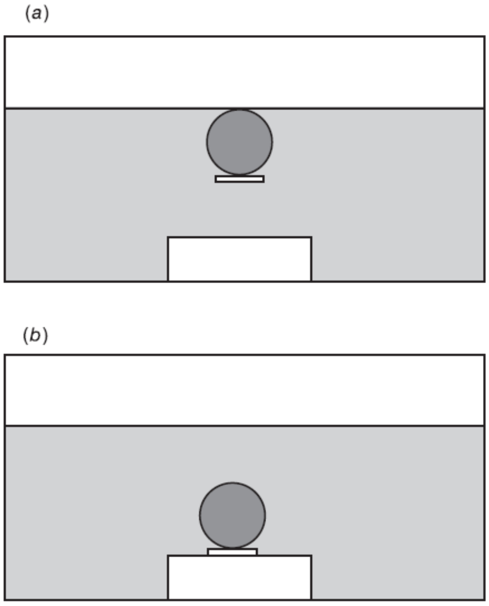


Figure 4.3: Experiment setup [25]

The author's explanation of this is simple, as there is no water underneath the plate attached, the water cannot exert a force directed upwards on the plate. Instead, there is a force from the fluid above the plate, directed downwards (the weight of the fluid column above the plate), keeping the plate attached to the prism. The authors conclude after this experiment that the "*widespread definition of buoyancy dealing with the amount of water displaced by the body immersed in a fluid is deficient*". *It is convenient for calculating the magnitude of the buoyancy force but can be misleading and give an erroneous result if applied without caution".* [25]

Comments: In figure a) the ball seems to be neutral buoyant, and this could lead one to think that the ball will stay in equilibrium if it is moved down into the water. But the authors do comment in the paper that the mass of the ball can be manipulated by adding or removing water inside of the ball, and thus manipulate how large the buoyancy force will be with regards to the mass of the ball.

It can be discussed how long the ball will be kept attached to the prism in practice, as in the end the ball will float up. This can be used as an observation to confirm that the buoyancy force finally counteracted some additional force that held the ball down and thus the original Archimedes Principle explained by the volume understanding of buoyancy is still valid.

4.2.2 Experiment 2: "Just what *Did* Archimedes Say About Buoyancy?" [26]

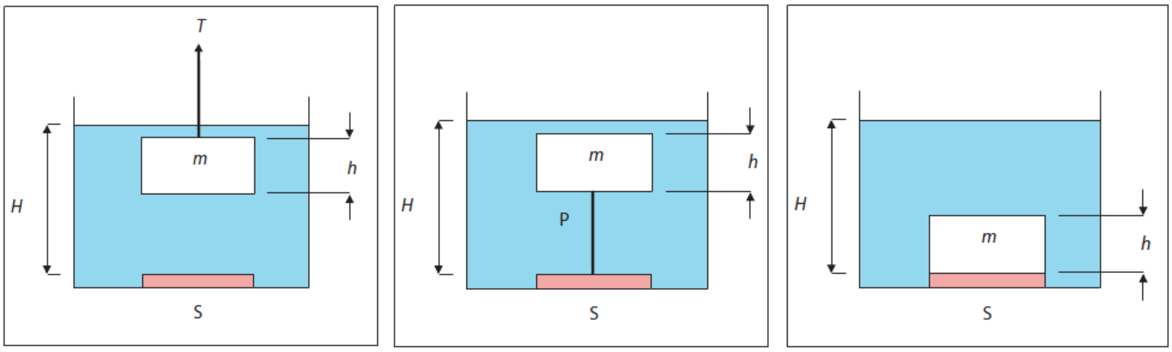


Figure 4.4: Experiment setup [26]

In figure 4.4 the experiments setup is shown in three steps from left to right. In step one a body with horizontal cross-sectional area equal to A has been lowered into a beaker. Resting on the bottom of the beaker is a scale with identical cross-sectional area as the body. The scale is designed to read the total force F on A (excluding the force due to atmospheric pressure). Readers are referred to the original paper for further details on the setup. [26]

Step 1:
The force T (direction upwards) holding the body in equilibrium, is equal to the weight of the body subtracted by the buoyancy force; [26]

$$T = mg - \rho_{fluid}\, ghA = mg - \rho_{fluid}\, gV_{displaced}$$

The force F read from the scale will be the weight of the water column directly above the scale; [26]

$$F_1 = \rho_{fluid}\, gHA$$

Step 2:
The body is now balanced by a pin which mass is neglected. There will clearly act a buoyancy force on the body in this case. The force P (direction downwards) holding the body in equilibrium will be equal to: [26]

$$P = mg - \rho_{fluid}\, ghA$$

The force F read from the scale will be; [26]

$$F_2 = \rho_{fluid}\, gHA + P = \rho_{fluid}\, gHA + (mg - \rho_{fluid}\, ghA)$$

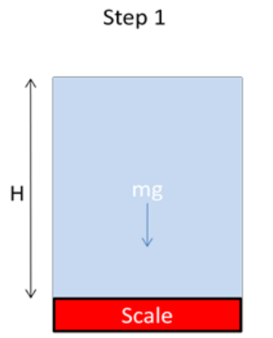
Step 3:
The body is now resting on the scale. The scale reading will be: [26]

$$F_3 = \rho_{fluid}\, gHA + (mg - \rho_{fluid}\, ghA)$$

Comments:

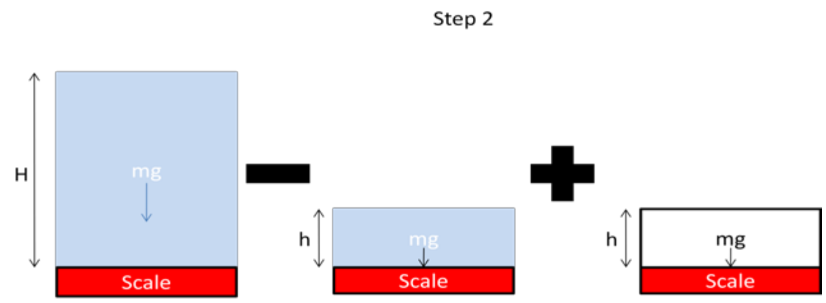
This shows that the scale will read the same in step two and step three ($F_2 = F_3$). In step two there is no doubt that there is a buoyancy force acting on the body. People favoring the volume understanding of buoyancy could use this as an example to prove that the buoyancy force really is volume force acting *through* the body and not *externally* on the body.

But after a closer look on what the scale actually is measuring, step by step, by imagining that the water is turned into a solid, as shown in figure 4.5, figure 4.6 and figure 4.7, it is not conclusive in this matter.



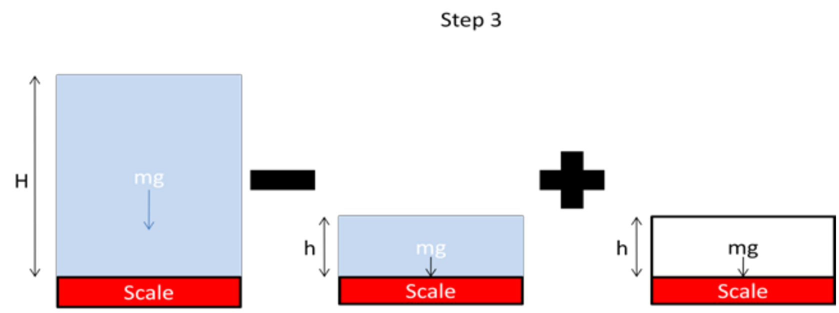
$$F_1 = mg_{water\ column}$$

Figure 4.5: Step 1



$$F_2 = mg_{water\ column} - mg_{displaced\ fluid} + mg_{body}$$

Figure 4.6: Step 2



$$F_3 = mg_{water\ column} - mg_{displaced\ fluid} + mg_{body}$$

Figure 4.7: Step 3

The author writes; “*The above procedure works no matter what the submerged object’s density may be. If its density is less than that of the fluid, the apparent weight would be negative and provision would have to be made to keep the object in force contact with the scale (e.g., by gluing it down or tethering it with a thread) during the first measurement. It wouldn’t do to have the object float away!”.* [26]

The question that then appears is; would the body really float away under the given boundary conditions (no fluid underneath the body in step 3)?

4.2.3 Experiment 3 – Goins [27]

In 1980 Goins conducted an experiment to prove if it is the displaced fluid that causes buoyancy or if it is the fluid pressure exposed to a projected area that causes buoyancy. The experiment setup is shown in figure 4.8. It consists of two steel tubes of the same weight placed in an open tank. One tube had an external bevel and the other an internal bevel. [27][28]

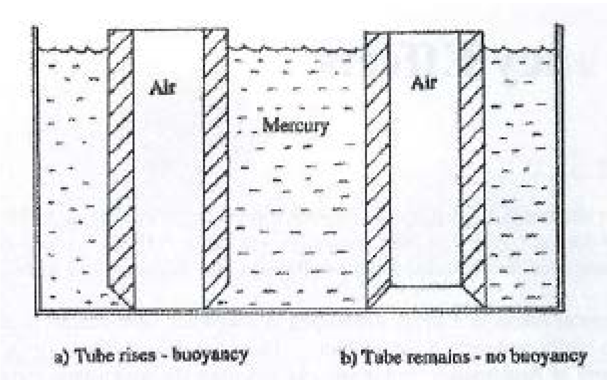


Figure 4.8: Experiment setup [28]

The tubes were held down as mercury was filled in the tank. When the tubes were released the cylinder with the external bevel rose (figure 4.8 a)) while the tube with the internal bevel (figure 4.8 b)) did not rise at all.

Comments: The result from this experiment is really hard to argue against as it is so conclusive. Actually, the internal bevel tube displaces a bit more fluid than the external bevel tube, yet it is only the external bevel tube that rises from the ground.

4.2.4 Experiment 4 – Bar through water container

This experiment was conducted in the lab at the University of Stavanger during the time this thesis was written. The basic setup is shown in figure 4.9. An empty container is placed over a scale. There is a hole in the bottom of the container were a solid bar with uniform cross-section A can be passed through. The density of the rod does not depend on the result in this experiment, but let's assume it is lower than the density of water. The clearance between the rod and the hole in the container is so narrow that no fluid will penetrate through as the water is poured into the container (a seal ring can be used). In fig 4.9 b) the scale will read the weight of the bar. As water is poured into the container, shown in figure 4.9 c), the scale will still read the weight of the bar, although it is displacing water. When the water level is higher than the length of the bar inside the container, the scale reading will start to add the weight of the water column right above the bar, as illustrated in fig 4.9 d).

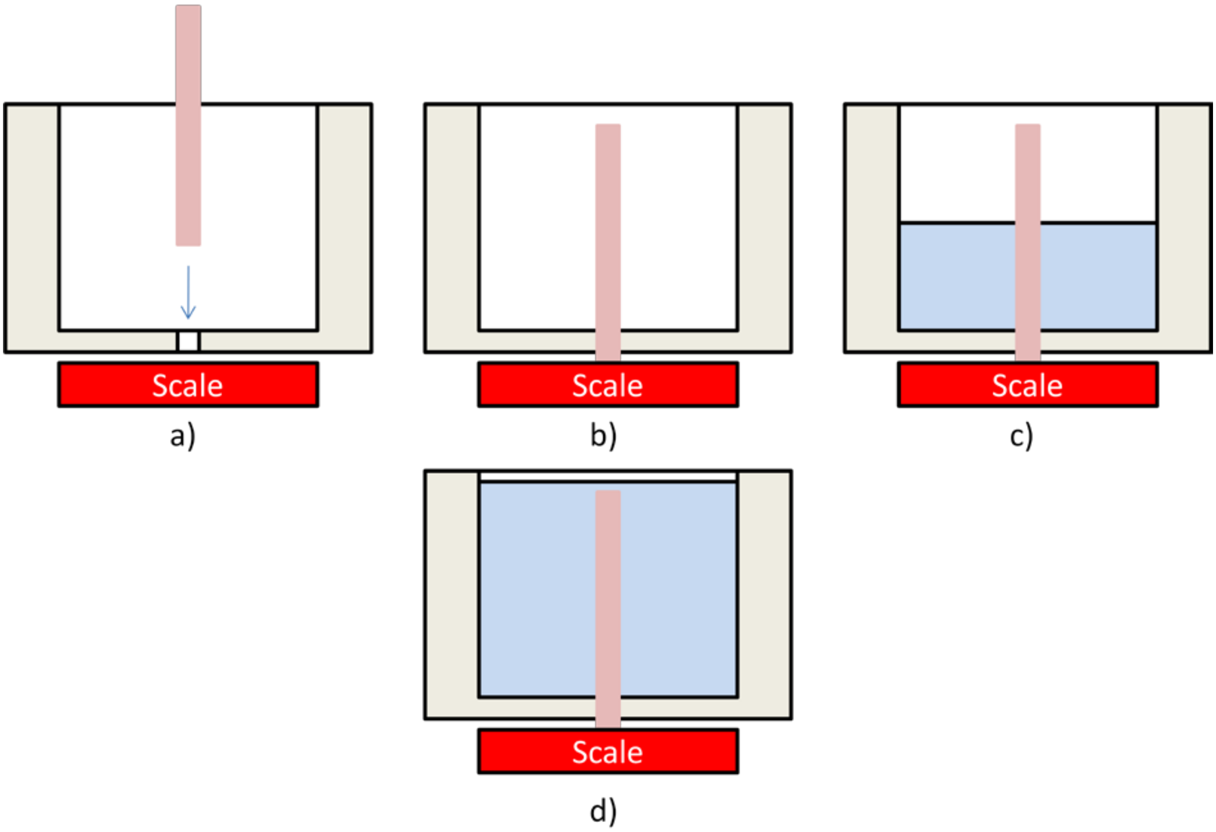


Figure 4.9: Experiment setup

Comments: The result from this experiment, as in the Goins experiment, is illustrating very convincingly how buoyancy works. If the bar was made of a really light material one could conclude that it actually would float up as the water was poured into the container, by using equation (4.2).

4.3 Discussion and conclusion

The two first experiments are from two recent papers [25][26] discussing buoyancy. There are more papers [29][30][31][32][33][34] discussing the same topic; some of that discussion will be referred to in this section.

The seemingly trivial question on how a buoyancy force should be defined comes in effect when looking at the different experiments shown above. In experiment 1 it could be argued that other forces (e.g. a "suction" force) come in effect, as pointed out in the paper. [25]

In experiment 2 it is demonstrated quite convincingly that the scale will read the same in both step 2 and 3. As there is no doubt that there is a buoyancy force acting on the block in step 2 this could be used to illustrates that the buoyancy force is equal to the volume displaced in both step 2 and step 3 although there is no fluid underneath the body in step 3.

Carl E. Mungan comments on the two papers [25][26], along with other papers [30][31][32] concerned on how buoyancy should be understood, in 2006 [29]. The abstract of the paper reads; "*I propose that buoyant force be generally defined as the negative of the total weight of the fluids that are displaced, rather than as the net force exerted by fluid pressures on the surface of an object. In the case of a body fully surrounded by fluids, these two definitions are equivalent. However, if the object makes contact with a solid surface (such as the bottom of a beaker of liquid), only the first,* *volumetric definition is well defined while the second definition ambiguously depends on how much fluid penetrates between the object and the solid surface".*

This proposal seems to be based on a pedagogical view of the matter; to not confuse students when introduced to buoyancy. This is a good proposition in most cases, but as shown in this chapter, it could lead to a flawed interpretation of what a buoyancy force really is.

The paper describing experiment 1 replies to this paper stating that there is no need to change the original definition of buoyancy. [33] "*One should merely keep in mind that the rather widespread formulation of Archimedes' principle that 'any body completely or partially submerged in a fluid (gas or liquid) at rest is acted upon by an upward, or buoyant, force the magnitude of which is equal to the weight of the fluid displaced by the body' is deficient.*"

Describing the result from experiment 3 Goins wrote "*This illustration leads to some interesting but not always accepted conclusions. In figure 4.8 b) there is fluid displaced by air and steel, but no buoyancy. In other words, the common concept that buoyancy is equal to the weight of fluid displaced is true only sometimes! Buoyant forces exist only when there is an exposed end or cross-sectional area to which hydrostatic pressure can be applied vertically*". [27][28]

Experiment 4 is merely an extension of experiment 3 which excludes any discussion regarding if there is any fluid between the bar and the bottom of the water container. Consequently both experiments show the same result.

To be able to distinguish between the volume understanding of buoyancy and the piston force understanding of buoyancy we will define "buoyancy force" in accordance with the piston force school;

"Buoyancy force is an external force directed in opposite direction of gravity. To calculate the buoyancy force the net pressure (acting upwards) on the bottom of a submerged object is subtracted by the net pressure (acting downwards) on the top of the submerged object."

Using this definition we can conclude that there can be no buoyancy force acting on a perfectly vertical riser of uniform cross-section when it is standing on the sea bottom, although it is still displacing fluid.

Chapter 5. Two schools of understanding buoyancy in the petroleum industry

This chapter will review how tension is calculated based on two schools and compare them. This is an extension of the buoyancy discussion in chapter 4, but now directed towards marine risers only.

5.1 Background

When assessing how forces act on a marine riser or on a drill pipe under influence of hydrostatic forces, available literature seems to have different understandings. According to Aadnoy [28] there are "*two schools of thought*". He describes these two schools as "*those who believe in the principle of Archimedes, and those who calculate the hydrostatic forces starting with the bottom surface of the string. The latter is called the piston-force approach.*" [28]

Arnfinn Nergaard, who is supervising this thesis, has made a short description of the differences of the two schools; [35]

"*Piston-force School*

- *Any object, fully or partly immersed in fluid is exposed to a buoyancy force equal to the weight of the displaced fluid. This is conditional upon the presence of a horizontal flat exposed to hydrostatic pressure.*
- *The resultant buoyancy force is calculated as the sum of all hydrostatic forces acting on the external surface of the object.*
- *An object with vertical sides without an underside flat subjected to hydrostatic pressure sees no buoyancy force.*

Principle of Archimedes School

- *Any object, fully or partly immersed in fluid is exposed to a buoyancy force equal to the weight of the displaced fluid.*
- *The resultant buoyancy force is calculated as the product of the volume and the density of the displaced fluid.*
- *An object with vertical sides without an underside flat subjected to hydrostatic pressure is subject to the full effect of buoyancy. "*

As seen these two schools are based on the two different methods of understanding buoyancy as discussed in chapter 4. And again, the two schools main difference comes in affect when observing an object with vertical sides standing on bottom, as shown in figure 5.1, with no fluid between the object and the bottom.

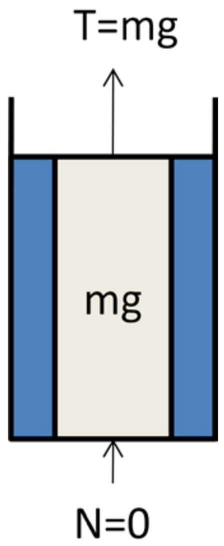


Figure 5.1: Solid bar standing on bottom with no fluid in between

In the following three sections the two schools are reviewed and compared.

5.2 The principle of Archimedes school

People favoring the principle of Archimedes school will state that “When a body is submerged into a fluid, the buoyancy force equals the weight of the displaced fluid”. This principle is simple and often used to explain why boats float. [28]

How this school understands buoyancy and internal tension force in a solid steel bar is shown in the following example:

In the left hand side of figure 5.2 a solid bar is hanging in vacuum, and the right-hand side shows the distribution of the internal tension. The force T to keep the bar in equilibrium is equal to the weight of the bar mg. The bar has a length L and a uniform cross-sectional area A. The internal force at any given point in the bar is calculated by

$$T_{int} = \rho_{steel} * g * A * \Delta L$$

where ΔL is the length from the bottom of the bar to the point of interest.

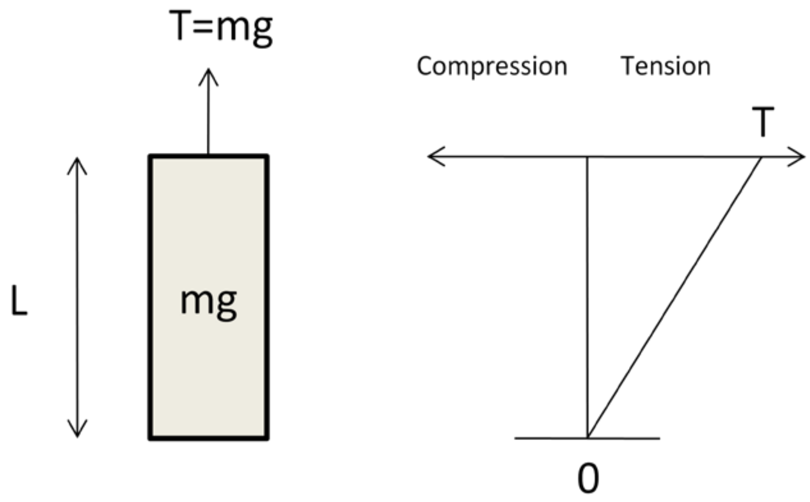


Figure 5.2: Solid bar hanging in vacuum and internal force distribution

If the bar is immersed into a fluid, T will be reduced by the buoyancy force B, which is equal to the weight of the fluid displaced;

$$B = V_{fluid\ \ displaced}\ * g * \rho_{fluid}$$

The blue line in figure 5.3 shows how the internal tension is described by the school of Archimedes. The essence is that the buoyancy force is a volumetric force, and is subtracted from the original tension in vacuum by

$$T2_{int_{Arc\ himedes}} = \rho_{steel}\ * g * A * \Delta L - \rho_{fluid}\ * g * A * \Delta L = g * A * \Delta L * (\rho_{steel}\ - \rho_{fluid}\ )$$

where ΔL is the length from the bottom of the bar to the point of interest.

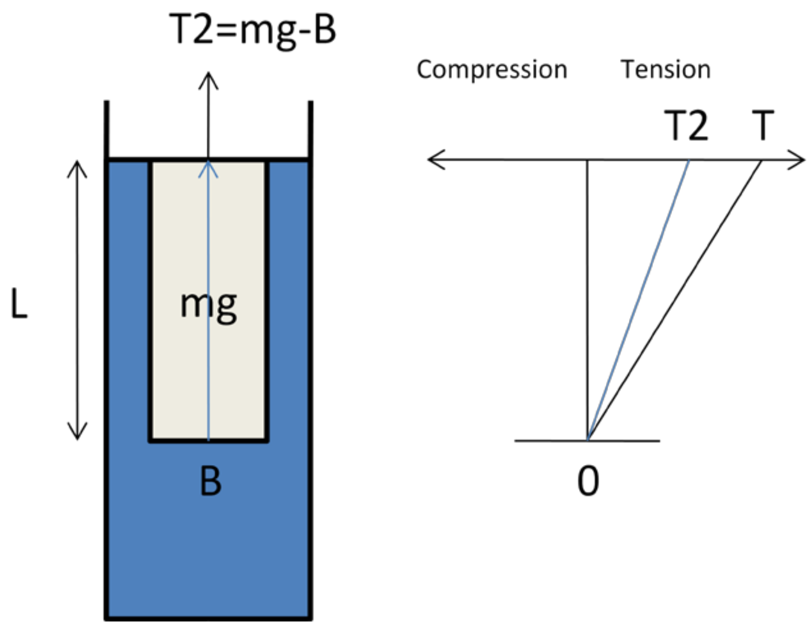


Figure 5.3 Internal forces acting in a solid bar immersed in fluid interpreted by the Archimedes School

This is the basics of the school of Archimedes.

5.3 Piston force school

The piston force school would agree on the first assessment in section 5.2, were the bar is hanging in vacuum and the internal forces acting as shown in figure 5.2.

When the bar is immersed in the fluid the piston force school applies the same mechanics; performing a vertical force balance of the rod (the horizontal forces created by the hydrostatic pressure acting on the vertical sides of the bar is neglected, as they are counteracting each other). The buoyancy force from the fluid will act on the lower end of the bar creating a vertical force directed upwards

$$B = p_{bottom} * A = \rho_{fluid} * g * L * A$$

The internal forces acting in the bar while immersed in fluid will then be

$$T2_{int_{piston}} = \rho_{fluid} * g * A * \Delta L - B$$

As shown in figure 5.4 this gives an external compressive force at the lower end of the bar and a top tension T which equals the weight mg of the bar minus the buoyancy force B. It is worth mentioning that if the bar was hanging in vacuum and a force equal to B was applied at the bottom of the bar, the internal force would be presented the same way.

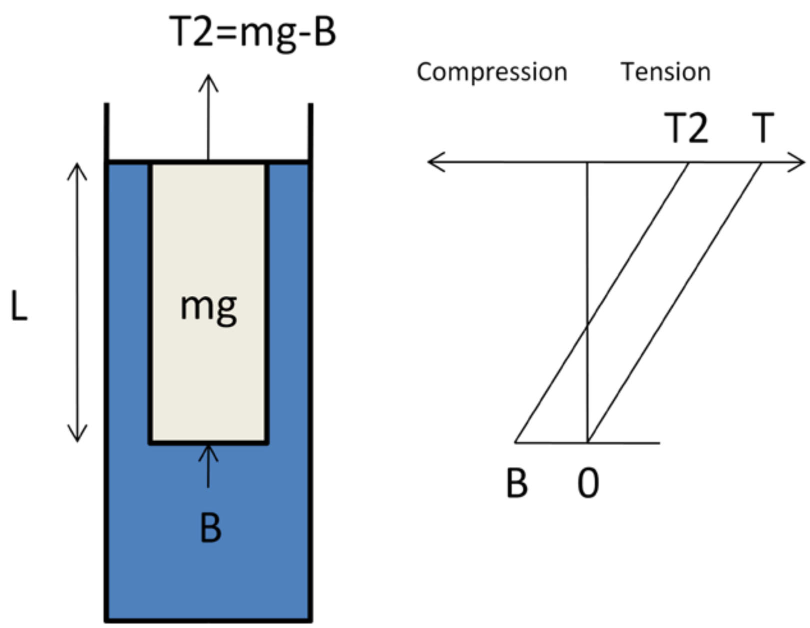


Figure 5.4: Internal forces acting in a solid bar immersed in fluid interpreted by the Piston Force School

5.4 Comparison

A closer look at figure 5.3 and figure 5.4 shows that in both cases the top tension is equal. But the internal forces are interpreted differently and the force on the lowest part of the bar is different. In figure 5.5 a comparison of the two schools interpretation of the internal forces are given. The blue line shows Archimedes' school and the black line shows the piston-force school.

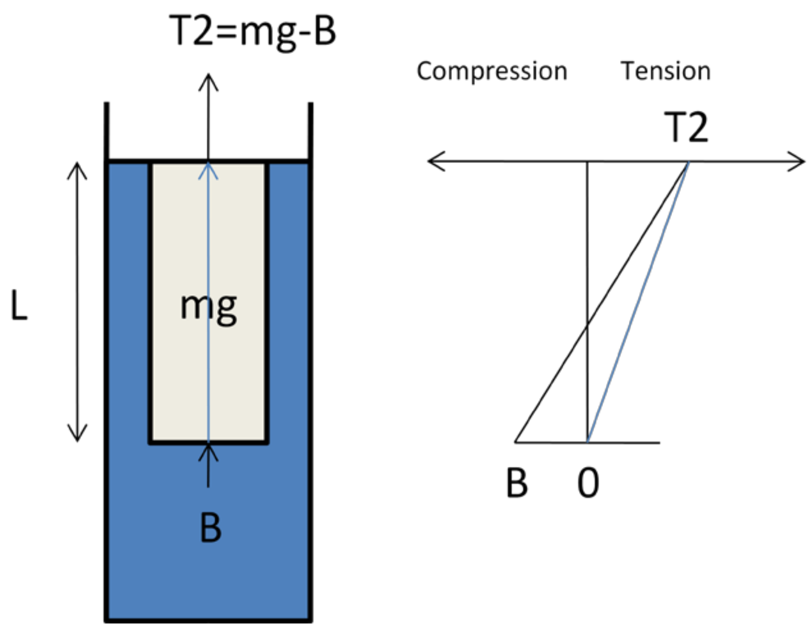


Figure 5.5: Comparison of both schools interpretation of internal forces

This shows that the piston-force school only calculates external axial forces (in one dimension), neglecting the horizontal force. If this approach is used on a free hanging bar lowered 2000 meters down into the sea, it would look like the bar would buckle because of the large lower end compression force. This would never happen in practice, as long as the bar has a density larger than the fluid it is displaced in, because of the horizontal pressure acting on the bar. When doing strength analysis, the piston-force method should never be used, as failure is not governed by the external axial force alone. [28]

We will now move the solid bar into a smaller beaker so that the bottom of the bar is in contact with the bottom of the beaker, as in figure 5.6. We will also assume that there will be no fluid between the beaker and the bar. If we now want to know how much top tension T is needed to prevent any compression at the bottom we will get two different answers.

The piston-force school will state that there is no pressure (force) acting on the lower end of the bar and that P will be equal to the weight of the bar, as in the vacuum case in the start of section 5.2. This is shown by the black line in figure 5.6.

The Archimedes' School will state that the bar is still displacing the same amount of fluid as in the two cases discussed above, thus the bar will be in tension at the lower end if T is equal to the weight of the bar as predicted by the piston force school. The result is shown by the blue line in figure 5.6.

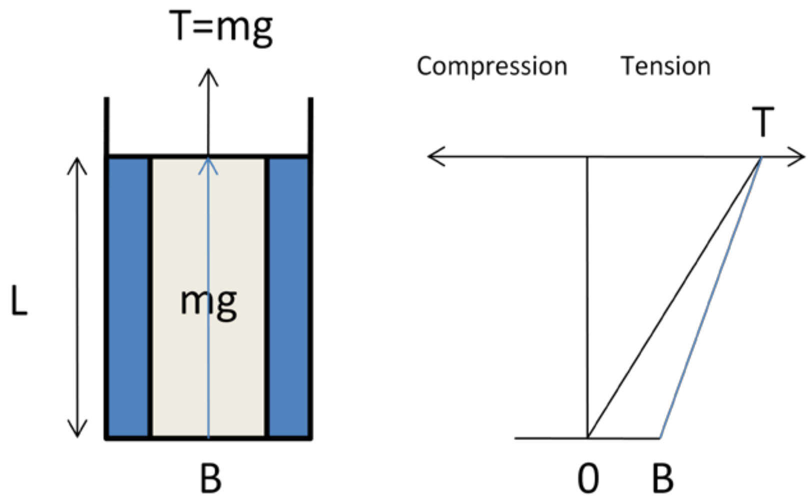


Figure 5.6: Solid bar standing on the bottom with no fluid

Recalling the effective tension equation derived in chapter three;

$T = w_a = w_t - w_e = w_t - p_e A_e$ (3.12)

Where

$T = Minimum\ top\ tension\ to\ prevent\ buckling$

$w_a = apparent\ weight\ of\ riser$

$w_t = weigth\ of\ riser\ in\ air$

$w_e = weight\ of\ displaced\ fluid$

$p_e = external\ pressure\ at\ lower\ end\ of\ riser$

$A_e = external\ cross - section\ at\ lower\ end\ of\ riser$

It is easy to show that the principle of Archimedes School method gives the same answer as the effective tension concept derived by Sparks in chapter 3.

It should also be noted that when using the principle of Archimedes School and the effective tension concept it is not the buoyancy force (defined in chapter 4) that is subtracted, but the weight of the fluid displaced by the bar.

This is the main reason for the confusion related to the term "buoyancy" when using the effective tension concept, as buoyancy is usually defined as "equal to the weight of the fluid displaced".

5.5 Relationship between the two Schools

As shown in section 5.4 there seems to be a relationship between both schools. In this section we will present this relationship.

Hubbert and Rubey wrote a paper in 1959 [36] on a geological study. In this paper they first derive mathematically the principle of Archimedes, as found in modern physic books. Then they transform it to a "generalized statement of the principle of Archimedes". In short this generalized statement of the principle of Archimedes shows the relationship between the "piston-force school" and the "Principle of Archimedes' school". [36]

Mathematical the transformation is shown in equation (5.1) [36]

$S = p + \sigma = \rho_l g z_2 + (\rho_s - \rho_l) g (z_2 - z_1)$ (5.1)

Where

$S = Total\ stress\ at\ depth\ z_2$

$p = fluid\ pressure\ at\ depth\ z_2$

$\sigma = effective\ stress\ in\ object\ at\ depth\ z_2$

$\rho_l = density\ of\ liquid$

$g = gravity$

$z_2 = length\ from\ fluid\ surface\ to\ desired\ point$

$\rho_s = density\ of\ solid$

$z_1 = length\ from\ fluid\ surface\ to\ top\ of\ solid$

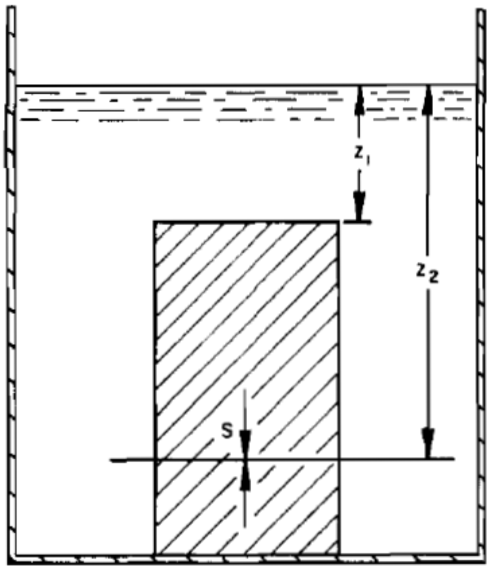


Figure 5.7: Illustration used in combination with equation (5.1) [36]

If equation (5.1) is used to calculate S at any given point of the body shown in figure 5.7, it will give the same result as calculating the internal tension using the piston force method. If S is subtracted with the fluid pressure at the same point, as if the solid body was not present, it will give the effective tension in the body as derived by Sparks [1], or the same answer as the school of Archimedes discussed in section 5.2.

In a paper from Hubbert & Rubey dated 1961 the authors describes how "one of the classical fallacies that has plagued engineers in the petroleum industry for several decades" can be explained by dividing the total stress S in to the sum of a hydrostatic stress component and an effective stress component.

5.5.1. Aadnoy

Aadnoy [28] describes both schools in detail in his book. He also shows that the total load in a submerged pipe can be divided in; [28]

$$Total\ load = Hydrostatic\ load + Deviatoric\ load$$

Where the deviatoric load component is the component calculated by the principle of Archimedes School.

5.6 Discussion

In chapter 5 we have been introduced to two different schools of understanding buoyancy and calculating internal tension in a submerged object. It is also shown that these schools are linked together. The piston force School only uses external axial forces to show the axial load in vertical direction neglecting the horizontal forces while the principle of Archimedes School shows the internal effective (or deviatoric) stress.

The main difference between the schools is that the piston-force school is a one dimensional view of the external axial forces while the principle of Archimedes school is a three dimensional view showing internal effective stresses along the entire axial length of the bar.

The term “deviatoric” seems to have replaced the term “effective” as used by Sparks [1] and Hubbert and Rubey [36][37][38]. Both Aadnoy [28] and Boresi [39] uses the term “deviatoric” which will add yet another term into all terms mentioned in chapter 1.

Hubbert & Rubey [36][37][38], Sparks [1] and Aadnoy [28] show how to calculate the effective stress in a bar using three different approaches which numerically gives the same answers.

Chapter 6. Bridgman experiment

In 1912 an experiment conducted by P. W. Bridgman showed that if a solid bar (Bridgman used different kinds of materials) is placed in a pressure vessel with both ends free, as illustrated in figure 6.1, and the pressure inside the vessel is in the numerical region of the maximum tensile strength of the solid's materiel, the bar will rupture as if in a tensile test. [40]

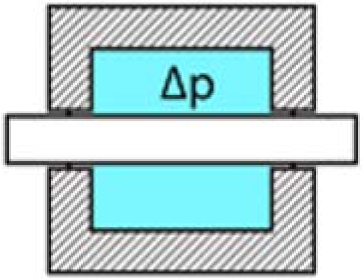


Figure 6.1: Setup for Bridgman experiment

At the time of the experiment this was considered as a paradox, as there were no axial pressure acting on the rod. Bridgman did try to explain this paradox, as shown in figure 6.2.

Application of axisymmetric pressure to a cylindrical specimen (or, similarly, biaxial stress applied applied to to a rectangular prism) is equivalent to a hydrostatic pressure applied to the specimen (triaxial stress) plus an applied axial tensile stress of the same value. [41] He further argued that since applying hydrostatic pressure does not essentially change the qualitative behavior of materials, this term in the "equation" disappears and the required result, that the axisymmetric pressure (or biaxial stress) is equivalent to applying an axial tension, remains" [42].

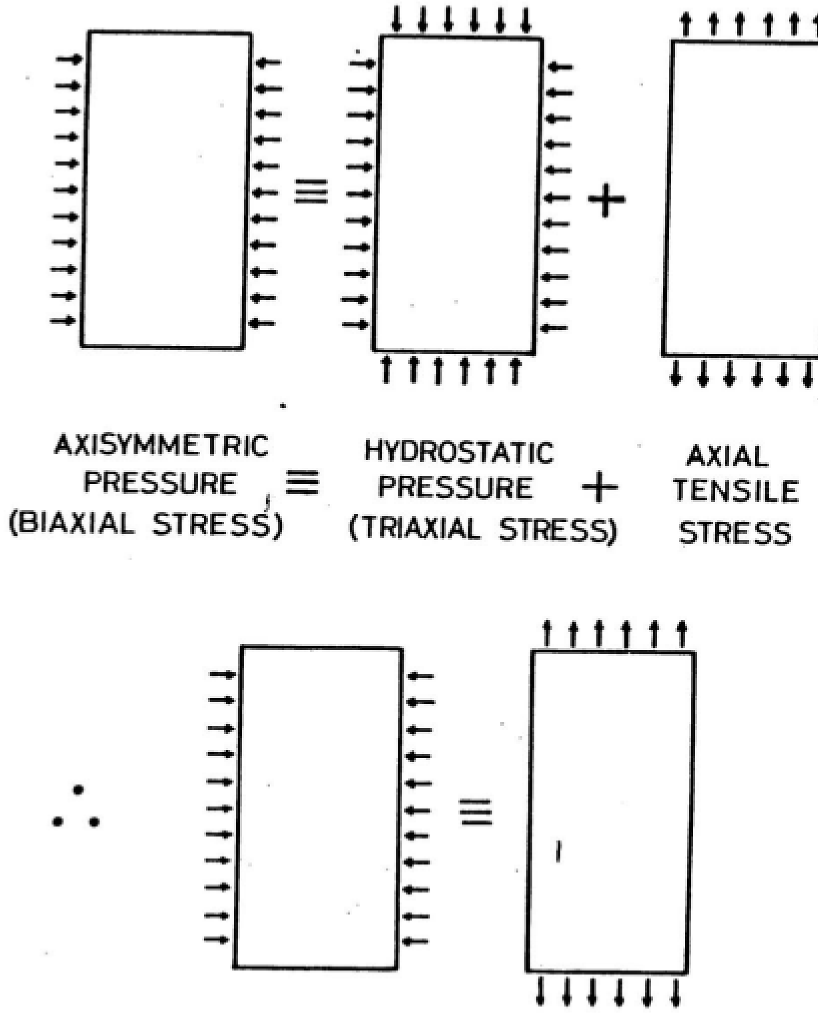


Figure 6.2: Bridgman's explanations of the "Pinching Off" effect. [42]

Hubbert and Rubey [38] did show that, by calculating the effective tension, one would derive to the conclusion that the axial stress would be equal to the radial pressure.

6.1 Introduction

The purpose of this experiment is to repeat P. W. Bridgman's experiment [40] showing that if a rod is exposed to hydrostatic pressure on its cylindrical surface (both ends are free) it will rupture as if in tension, if the hydrostatic pressure is close to or exceeds the tensile strength of the rod materiel. A sketch of the pressure vessel and rod is shown in figure 6.1.

By calculating the effective tension in the rods used in this experiment one would conclude that the rods would experience an effective tension equal to the pressure inside the pressure vessel. This would imply that the rod should rupture as if in a tension test. Furthermore, this will show that the effective tension is not just a "fictitious" stress, or force, as mentioned in chapter 1.

6.2 Procedure

6.2.1 Equipment

- High pressure unit including: pump, air pressure gauge, water pressure gauge, air regulator, on/off switch, open/close valve for high pressure side, bleed of valve, external air intake, external water intake and high pressure water outlet. (Figure 6.3 and 6.4)
- Pressure vessel. (Figure 6.5)
- High pressure hose to connect high pressure unit with the pressure vessel. (Figure 6.8)
- Rods made of Polytetrafluoroethylene (PTFE) (also known as Teflon). (Figure 6.6)
- Rods made of Poly(methyl methacrylate) (PMMA) (also known as acrylic glass or Plexi glass). (Figure 6.7)
- Water and air from external sources
- Wooden crate.

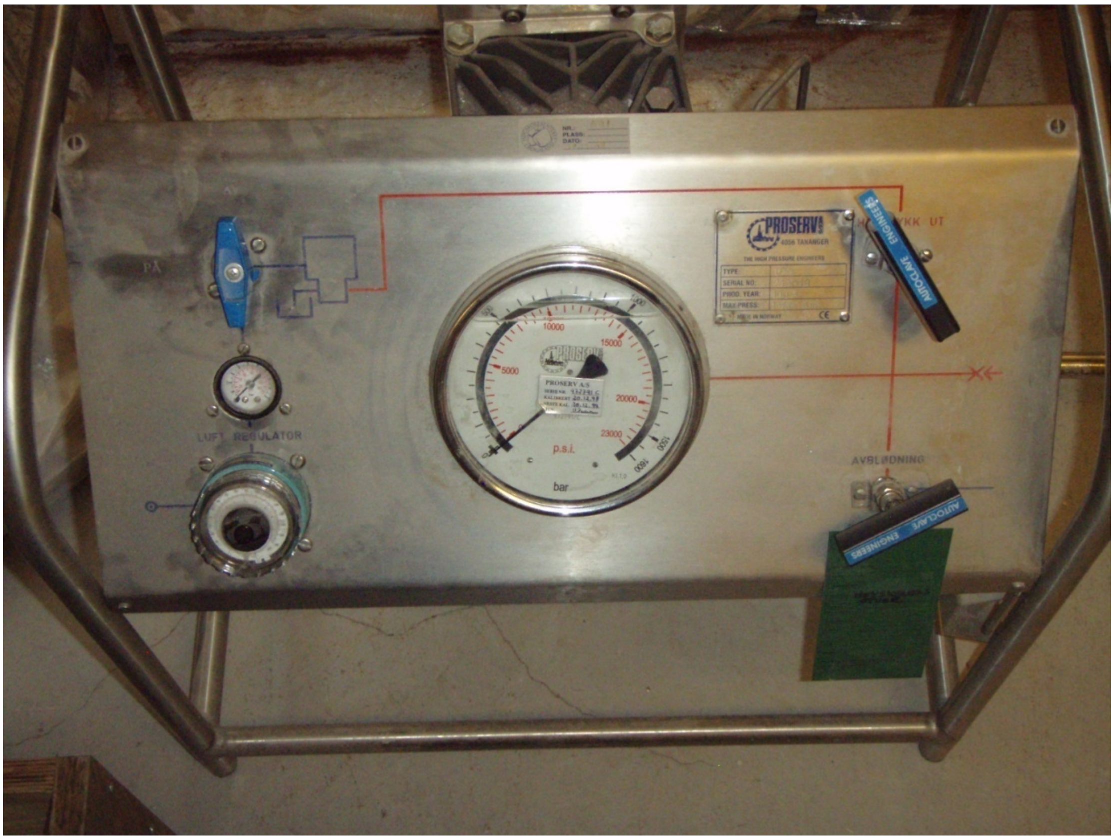


Figure 6.3: High pressure unit

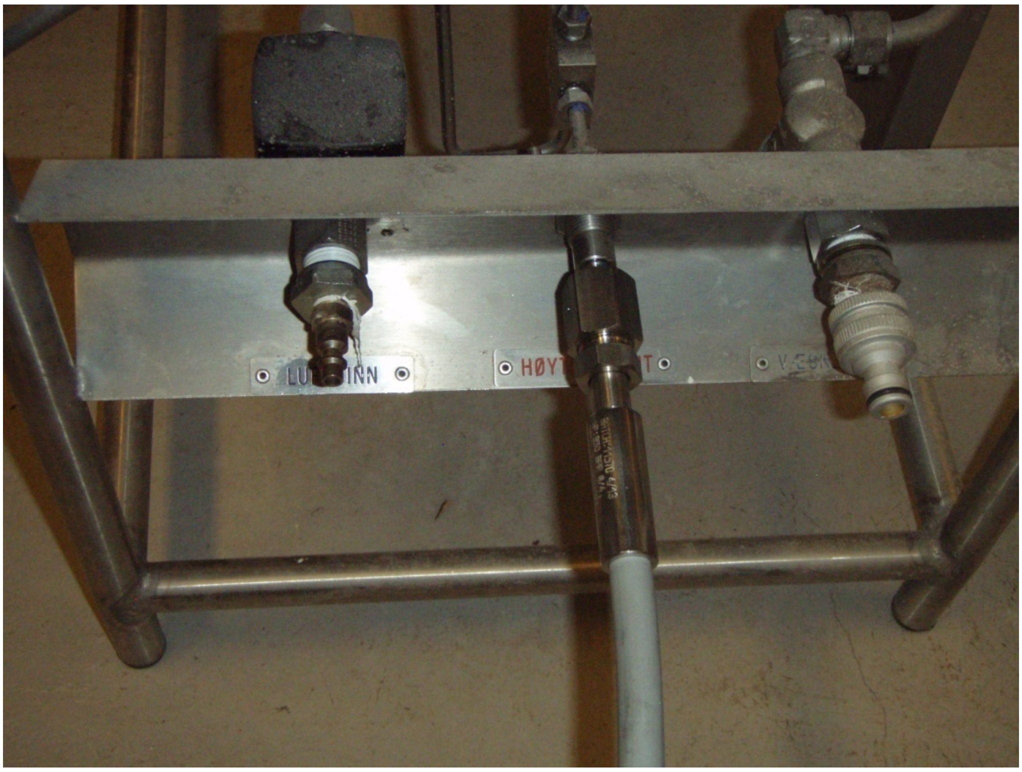


Figure 6.4: High pressure unit; External air and water intake and high pressure water outtake

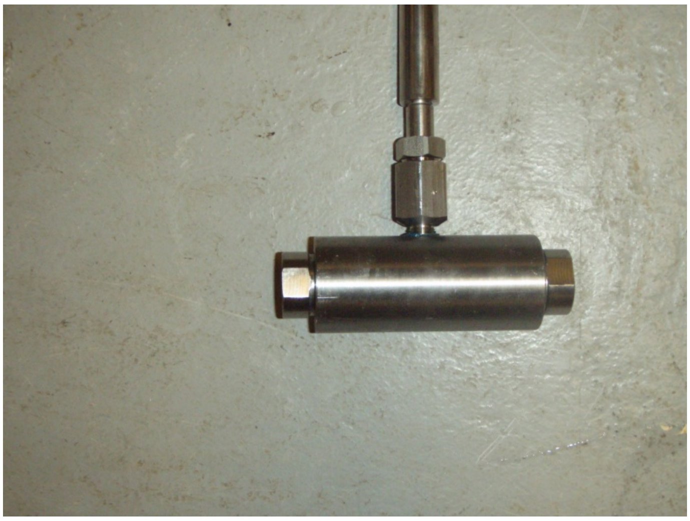


Figure 6.5: Pressure vessel (Bridgman's chamber) connected to high pressure hose

Figure 6.6: PTFE rod after test, a close look will show the initiated yielding and reduction of the outer diameter

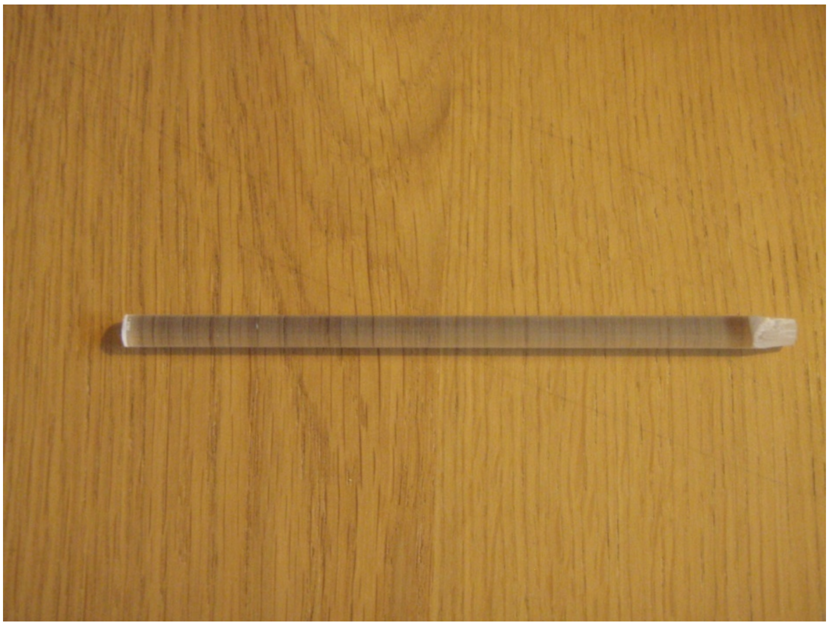


Figure 6.7: PMMA rod prior to testing, the cone in the end helps during introducing the rod through the seal rings

6.2.2 Setup

The high pressure unit is connected to water and air from external sources. The water is used as fluid inside the closed pressure system and the air is powering the pump to create the pressure needed. The high pressure unit has a maximum pressure rating of 1380 BAR (20000PSI).

Figure 6.8: High pressure unit connected to the pressure vessel

A high pressure hose forms the connection between the high pressure pump and the pressure vessel. The hose has a maximum pressure rating of 1425 bar (20668 PSI).

The pressure vessel is made up of a cylindrical body with a 9mm axial hole. In both ends there is a packing box with a high pressure seal assembly. The pressure vessel has a maximum pressure rating of 1034 bar (15000 PSI). For more details regarding the pressure vessel, see appendix B.

To prevent air inside the system water is circulated through the pressure vessel as the rod is introduced. To further prevent air in the system pressure is applied in small increments (<35 bar) after the rod is in place and then bled of. This is done several times to ensure that no air is left inside the system.

During testing the pressure vessel is placed in a wooden crate. (Figure 6.9) This is to absorb any potential impact from fractured rod elements.

Figure 6.9: Pressure vessel and ruptured rod parts inside a wooden crate

6.2.3 Execution

The experiment was conducted on rods made of PTFE and PMMA.

After the rod is introduced into the pressure vessel and air has been removed from the closed system, pressure is increased in increments of 35 to 50 bar each held constant for 30 to 60 seconds. This is done to let the pressure vessel and rod adapt to the increasing pressure. Pressure is raised in this manner until the pressure suddenly drops to zero. This indicates that the rod has eventually failed or there is a leakage in the system. The highest pressure reading is noted.

6.3 Material properties

Please note that the test rods specimens were delivered without specific material properties. Thus, general properties for the two types of rods have been used

6.3.1 PTFE

Typical material properties for PTFE[43]:

- Tensile strength: $\sigma_{TS_{PTFE}} = 20 - 30\ MPa$
- Elongation: 200 – 300%

A Typical stress – strain curve for PTFE is shown in figure 6.10.

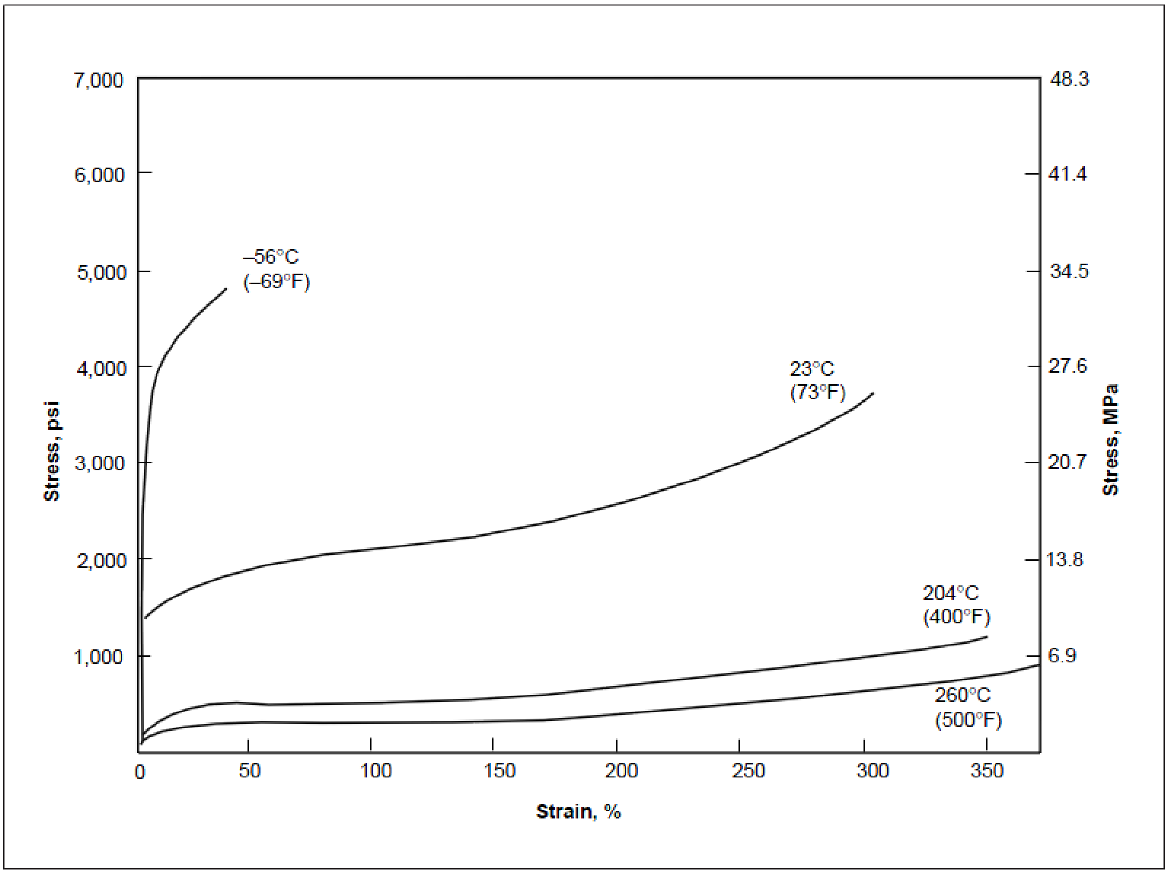


Figure 6.10: PTFE Stress vs. Strain in tension [44]

6.3.2PMMA

Typical material properties for PMMA[45]:

- Tensile strength: $\sigma_{TS_{PMMA}} = 48 - 76\ MPa$
- Elongation: 2 – 10%

A Typical stress – strain curve for PMMA is shown in figure 6.11.

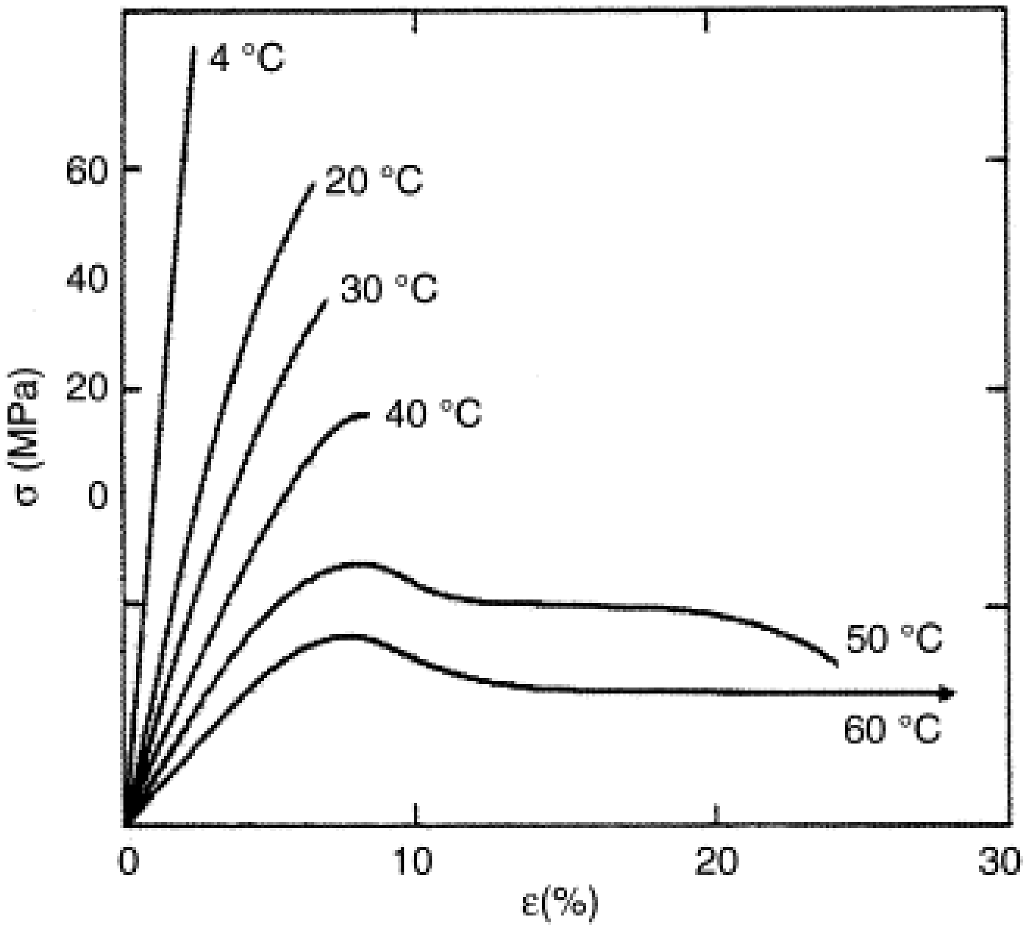


Figure 6.11: PMMA stress vs. strain in tension [46]

6.4 Calculations

This experiment is based on the assumption that the pressure needed inside the pressure vessel to rupture the rod is equal to the tensile strength of the material, thus we only need to consider the tensile strength of each material.

The basic assumption is that the pressure sets up an axial tension force, T;

$$T = \Delta p * A_{rod}$$

Where:

$$\Delta p = pressure\ differential, inside\ to\ outside\ of\ pressure\ vessel$$

$$A_{rod} = cross - sectional\ area\ of\ the\ rod$$

From this follows that the axial stress is:

$$\sigma_A = \frac{\Delta p * A_{rod}}{A_{rod}} = \Delta p$$

But as there are two seals in the pressure vessel setup we also need do determine the friction force between the seals and the rod as this will counteract any potential axial force from the hydrostatic pressure, Δp, see figure 6.12.

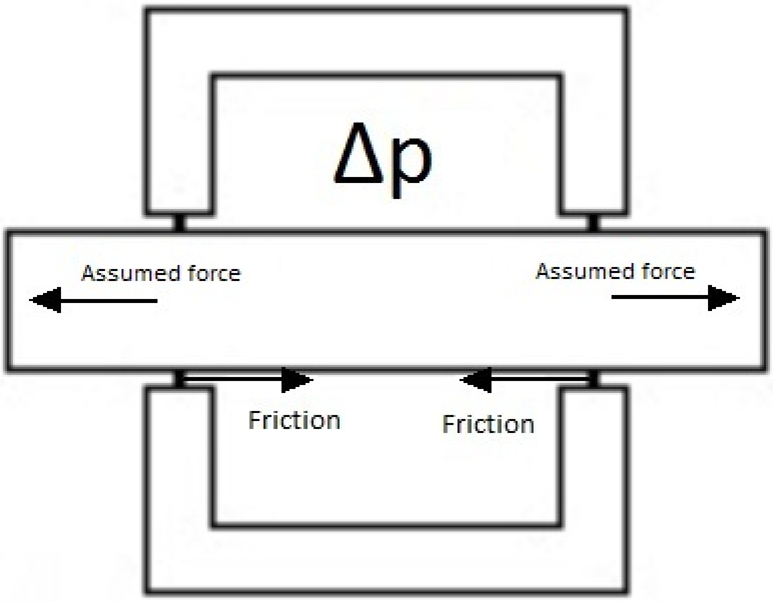


Figure 6.12: Friction forces from the sealing rings

Equation (6.1) shows the needed pressure inside the pressure vessel to rupture the rod.

$\Delta p = P_r = TS + \frac{F_{sf}}{A_{rod}}$ (6.1)

Where

$P_r = Pressure\ needed\ for\ rupture$

$TS = Tensile\ strength\ of\ the\ rod\ material$

$F_{sf} = Static\ frictoin\ force$

$A_{rod} = Area\ of\ rod$

When Δp is increased, F_{sf} also will increase. As a consequence of this, equation (6.1) should be iterated to give the most precise calculation. But as the tensile strength varies much we will neglect the iterations for the simplicity.

6.4.1 Friction

The friction force between the rod and the seals can be calculated according to equations found in the Parker Handbook [47], see appendix A for more detail. The results are shown in figure 6.13

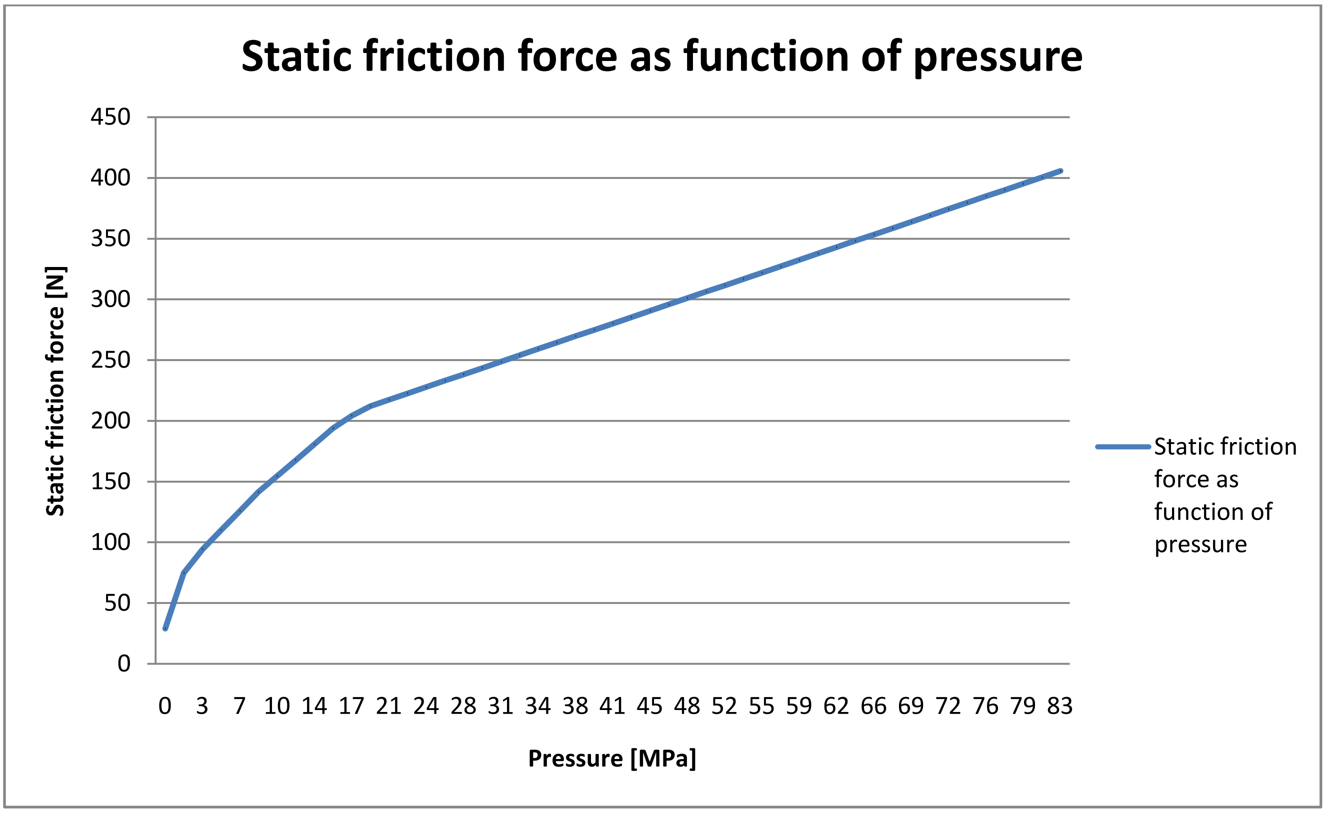


Figure 6.13: Static friction force as function of pressure

6.4.2 PTFE – calculated rupture pressure

Calculated pressure needed to rupture the PTFE rod is found in table 6.1. The difference between the two columns in table 6.1 is the calculated friction force.

Table 6.1: Calculated rupture pressure for the PTFE rod

Tensile Strength TS $[\mathrm{MPa}]$	Pressure P_r $[\mathrm{MPa}]$
20	24
22,5	27
25	30
27,5	32
30	35

The results show that the rod should rupture at a pressure in the range of 24 to 35 MPa depending on the tensile strength of the individual rod.

6.4.3 PMMA – rupture pressure

Calculated pressure needed to rupture the PMMA rod is found in table 6.2. The difference between the two columns in table 6.2 is the calculated friction force.

Table 6.2: Calculated rupture pressure for the PMMA rod

Tensile Strength TS $[\mathrm{MPa}]$	Pressure $\mathrm{P_r}$ $[\mathrm{MPa}]$
48	54
50	56
52	58
54	60
56	62
58	65
60	67
62	69
64	71
66	73
68	75
70	77
72	79
74	82
76	84

The results show that the rod should rupture at a pressure in the range of 54 to 84 MPa depending on the tensile strength of the individual rod.

6.5 Results

6.5.1 PTFE

Six rods where tested to a pressure of approximately 25 MPa. The pressure inside the pressure vessel dropped to zero after reaching this pressure and no rupture of the rod was observed. The outer diameter the rod that was inside the pressure vessel had signs of shrinkages (the material had gone into a plastic state). The rod had consequently elongated. Pressure readings and geometries before and after the test of the rods can be found in table 6.3.

Table 6.3: Pressure readings and geometries of PTFE rods before and after test

Rod no.	Maximum pressure [MPa]	Length before test [mm]	Length after test [mm]	Diameter before test [mm]	Diameter after test [mm]
1	24,7	187	194	8,19	7,57
2	25	207	213	8,18	7,7
3	24,6	201	208	8,18	7,67
4	25,1	210	217	8,18	7,65
5	25	210	217	8,18	7,73
6	24,9	176	185	8,17	7,5

6.5.2 PMMA

Ten rods made of PMMA was tested and all test specimens ruptured. The pressure at rupture varied from 65 MPa to 75 MPa. There is no noticeable reduction of the diameter or elongation of the rod. The rods had sign of small cracks along the length of the rod that was inside the pressure vessel.

Table 6.4:Pressure readings at rupture of PMMA rods

Rod no.	Max. Pressure [MPa]
1	67
2	69
3	73
4	73
5	75
6	71
7	70
8	69
9	74
10	72

6.6 Discussion of results

6.6.1 PTFE

The result from testing the PTFE rods shows substantial elongation in the axial direction. Following this the outer diameter of the rod is decreased and the fluid inside the pressure vessel leaks out through the seal rings (the pressure drops to zero). Looking at the material properties from section 6.4, especially the elongation before failure, it is concluded that PTFE is too ductile for this kind of rupture test.

6.6.2 PMMA

The result from testing the PMMA rods is as expected. The rods rupture when the pressure inside the pressure vessel is in the same range as the calculated values in section 6.5.3. The result will be discussed more thoroughly in the next chapter.

6.7 Sources of error

Readings from the pressure gauge: The pressure gauge is mechanical and does not record the pressures. All readings were done manually.

High pressure unit and pressure vessel size: The high pressure unit is quite large relative to the pressure vessel. It is probably intended to be used for larger fluid volumes than used in this experiment. This makes it very difficult to regulate the pressure in small increments.

Material properties: The material properties used are general ones. No test to find actual tensile strength of the material was attempted.

Chapter 7. Discussion of Bridgman experiment

In this chapter we will discuss the results from the Bridgman experiment.

7.1 Effective tension

Hubbert and Rubey did in 1961 derive that the effective tension stress in the length of the bar inside the pressure vessel would be equal to the radial pressure; an illustration of this is shown in figure 7.1. [38]

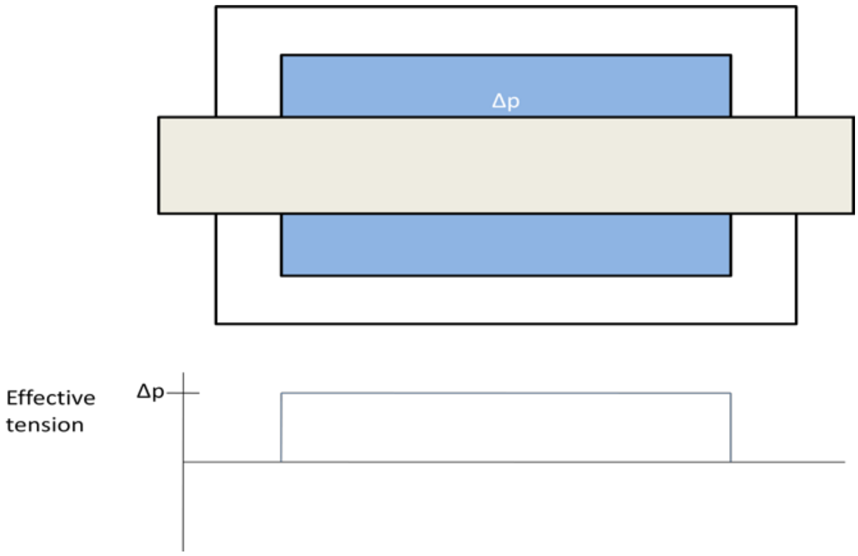


Figure 7.1: Effective tension in the bar derived by Hubbert and Rubey

This derivation can be interpreted as there is a pressure drop at both sealing rings equal to p,this will give an effective force equal to $F = p * A$ at both sealing rings. By using equation (3.11) which is derived from Sparks effective tension concept, this will yield the same answer. If the solid bar is cut precisely where the sealing rings are, this interpretation can easily be shown as in fig 7.2.

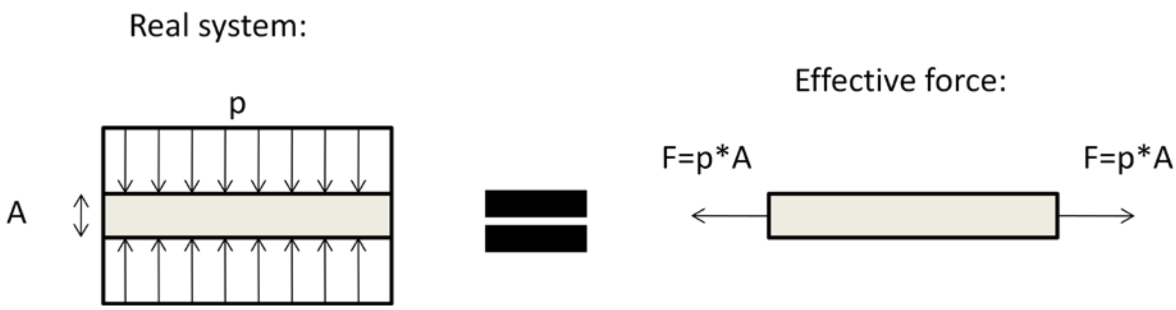


Figure 7.2: Interpretation of the effective tension

Another way of describing this interpretation is given by my supervisor Arnfinn Nergaard. Figure 7.3 shows three steps on how the Bridgman test can be understood, step 1 at the top, step 2 in the middle and step 3 at the bottom.[35]

Step 1:Faced with a seemingly trivial physical problem with a pressurized chamber with a bar protruding the chamber walls through sealing glands, most people opts for Solution 1; no resulting axial force; T= 0.

Step 2: Reversing the pressure there is 100% agreement that Solution 2 gives the right answer; a compression force corresponding to the overall piston effect.

Step 3: Most people agree that switching external pressure to negative (suction) gives the opposite of the Step 2 compression; a tension force with the same magnitude as the Step 2 compression.

And the question: Isn't Step 3 identical to Step 1?

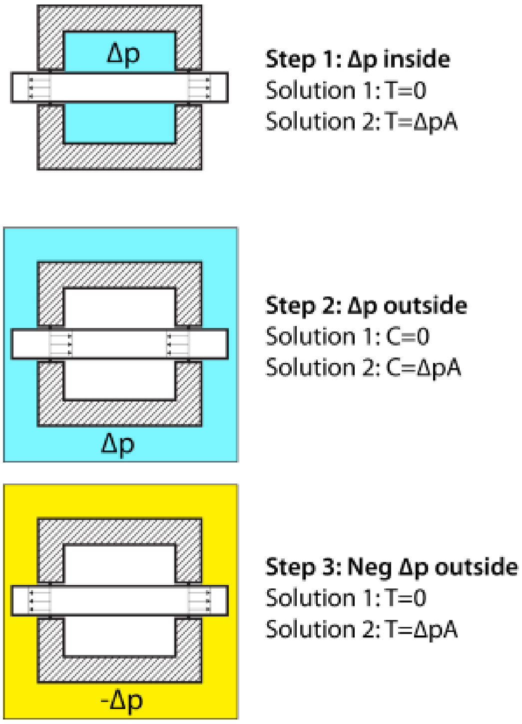


Figure 7.3: Nergaard's illustration of effective tension [35]

7.2 Numerical example using mechanics

In this section we will have a short introduction to stress tensors and then give a numerical example showing that with normal mechanics one would conclude that the bar in the Bridgman chamber would rupture as if in tension because of the radial pressure.

7.2.1 Introduction to stress tensors

The stress components at a point in a loaded body are shown in figure 7.4. [39]

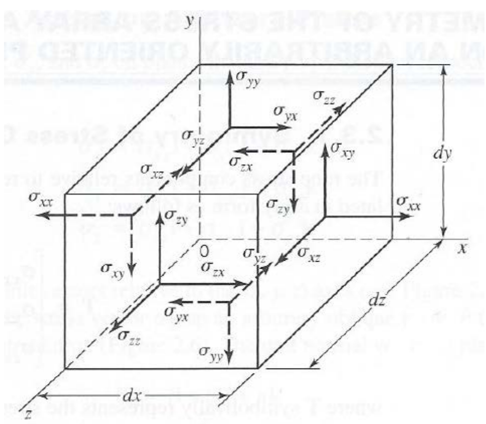


Figure 7.4: Stress components at a point in loaded body [39]

The nine stress components can be tabulated into a stress tensor as follows: [39]

$$T = \begin{bmatrix} \sigma_{xx} & \sigma_{xy} & \sigma_{xz} \\ \sigma_{xy} & \sigma_{yy} & \sigma_{yz} \\ \sigma_{xz} & \sigma_{zy} & \sigma_{zz} \end{bmatrix}$$

Experiments indicate that yielding and plastic deformations of ductile metals are essentially independent of the mean normal stress σ_m, where

$$\sigma_m = \frac{\sigma_{xx} + \sigma_{yy} + \sigma_{zz}}{3}$$

The mean normal stress can also be termed hydrostatic stress or neutral stress. [Aadnoy, Hubbert 59]

Most plasticity theories postulate that plastic behavior of materials is related primarily to that part of the stress tensor that is independent of σ_m. Therefore, the stress tensor is rewritten in the following form: [39]

$$T = T_m + T_d$$

Where T_m is

$$T_m = \begin{bmatrix} \sigma_m & 0 & 0 \\ 0 & \sigma_m & 0 \\ 0 & 0 & \sigma_m \end{bmatrix}$$

And T_d is

$$T_d = \begin{bmatrix} \sigma_{xx} - \sigma_m & \sigma_{xy} & \sigma_{xz} \\ \sigma_{xy} & \sigma_{yy} - \sigma_m & \sigma_{yz} \\ \sigma_{xz} & \sigma_{zy} & \sigma_{zz} - \sigma_m \end{bmatrix}$$

7.2.2 Tensile strength test

In the following we will assume that the bar in the Bridgman test had a tensile strength of 60 MPa, that the bar was rectangular and not cylindrical and that tensile stress is positive and compression stress is negative.

If a tensile test was conducted the bar would rupture when the applied axial stress exceeded 60 MPa as shown in figure 7.5.

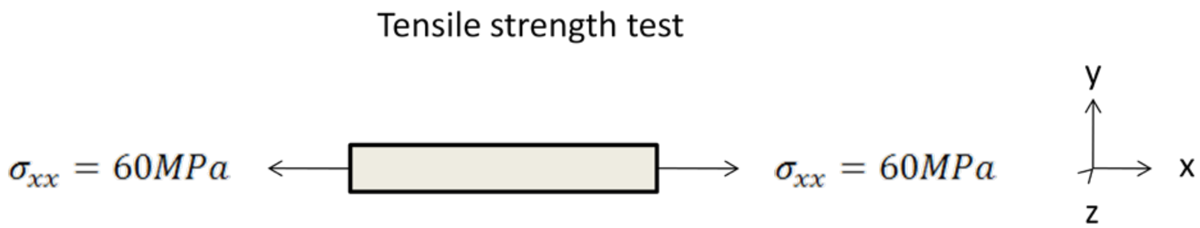


Figure 7.5: Tensile strength test

If we divide the stress tensor into a mean and a deviatoric stress tensor as shown in section 7.2.1 we will get;

$T = T_m + T_d$

$$\begin{bmatrix} 60 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} = \begin{bmatrix} 20 & 0 & 0 \\ 0 & 20 & 0 \\ 0 & 0 & 20 \end{bmatrix} + \begin{bmatrix} 40 & 0 & 0 \\ 0 & -20 & 0 \\ 0 & 0 & -20 \end{bmatrix}$$

By the statement made in section 7.2.1 it is the deviatoric stress tensor that initiates yielding and failure.

If we then look at the stress tensor given if we load the bar as done in the Bridgman experiment, as shown in fig 7.6, we get;

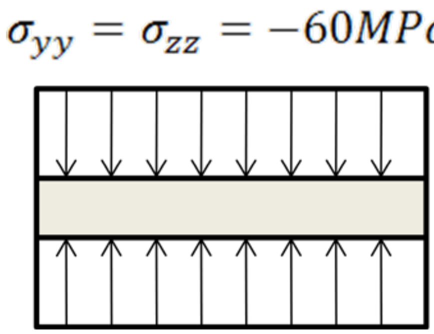


Figure 7.6: Pressure loading as in the Bridgman experiment

$$\begin{bmatrix} 0 & 0 & 0 \\ 0 & -60 & 0 \\ 0 & 0 & -60 \end{bmatrix} = \begin{bmatrix} -40 & 0 & 0 \\ 0 & -40 & 0 \\ 0 & 0 & -40 \end{bmatrix} + \begin{bmatrix} 40 & 0 & 0 \\ 0 & -20 & 0 \\ 0 & 0 & -20 \end{bmatrix}$$

We get the same deviatoric stress tensor as if the bar had been in a tensile test.

7.3 OpenFOAM results

Bjørn Hjertager, professor at the University of Stavanger, simulated the Bridgman experiment using computer program named OpenFOAM. The geometrical and mechanical properties of PMMA used were;

$Length\ of\ bar: L = 160mm$

$Diameter: D = 8mm$

$Length\ og\ bar\ inside\ pressure\ vessel: L_p = 60mm$

$Elastic\ modulus: E = 2450MPa$

$Poisson's\ ratio: v = 0,375$

$Density: \rho = 1185 \frac{kg}{m^3}$

The result from OpenFOAM is given as figures 7.7, 7.8, 7.9, 7.10, 7.11 and 7.12. Since the bar and pressure is symmetrical over the x and y plane only ¼ of the bar is shown in the figures. All results are in SI units.

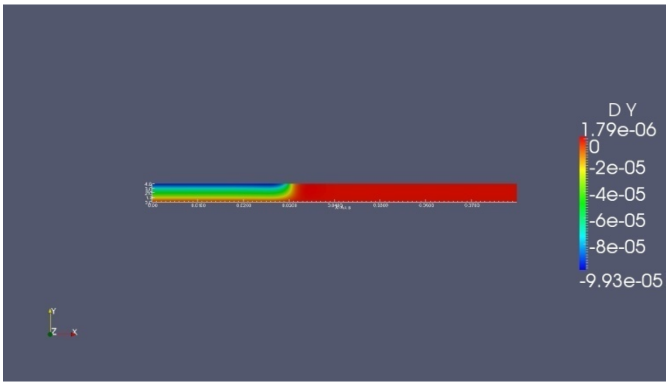


Figure 7.7: Elongation in y-direction [48]

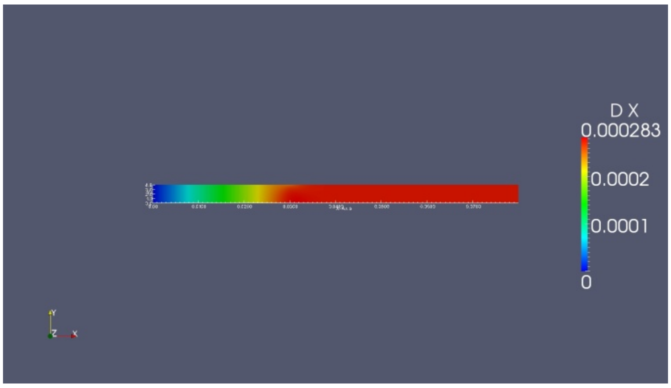


Figure 7.8: Elongation in x-direction [48]

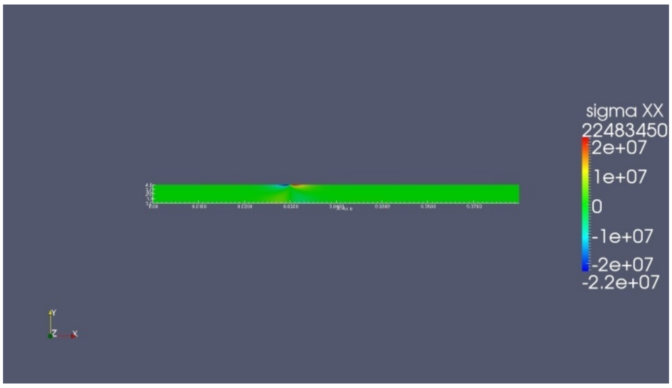


Figure 7.9: Stress in x-direction [48]

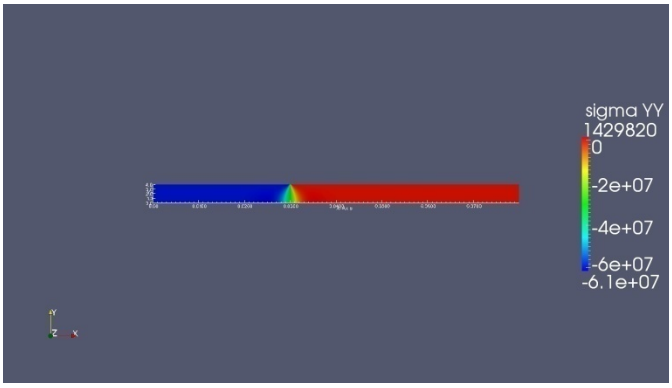


Figure 7.10: Stress in y-direction [48]

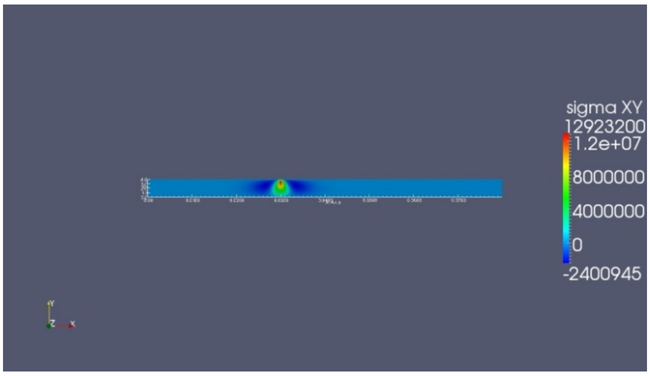


Figure 7.11: Shear stress [48]

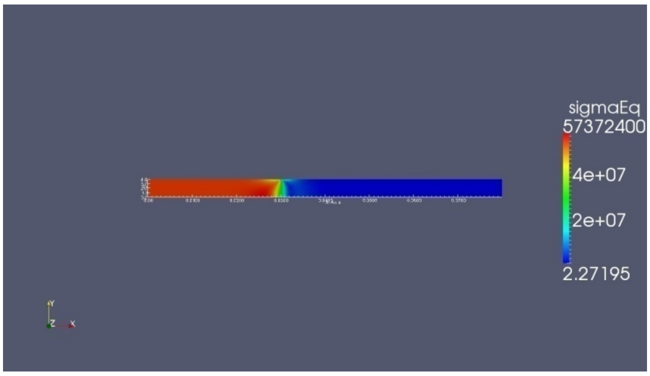


Figure 7.12: Von Mises stress [48]

As shown in figure 7.9 there is no stress in the x-direction but the Von Mises stress indicates that failure can occur along the length of the rod inside the pressure vessel because of the compression from the fluid pressure.

7.4 Comments

Using the results from the Bridgman experiment it has been shown how the effective tension can be interpreted or visualized as effective forces. As the bar do rupture as predicted by the effective tension concept it has been shown that the effective stress is not just an "fictitious force", but it does give a good correlation between the stress state in the material and the effective tension concept.

The result from section 7.2 indicates that the rods effective (or deviatoric) stress state is equal in a tensile test and in the Bridgman experiment. The effective tension is, as shown theoretically in this thesis, a description of the effective (or deviatoric) stress state in the rod. The results from the Bridgman experiment strengthen the theory presented.

Another observation is that the effective tension concept can be used on other calculations besides riser calculations (which, as in the Bridgman experiment, have no relationship to buoyancy forces).

Chapter 8. Conclusion and Recommendations

8.1 Conclusion

When being introduced to the effective tension concept one should have a solid fundamental understanding of what a buoyancy force is and how it is interpreted in physics. Further on it should be stressed that the effective tension concept is an interpretation of a three-dimensional stress state which gives a one dimensional answer (needed top tension). A buoyancy force is a reaction force directed in the opposite direction of gravity. It does not include the effects the horizontal forces would have on a riser. This does the effective tension concept account for.

Buoyancy should always be defined as external force acting on a body as a result of the pressure difference between the lower end and top end of an object submerged into fluid. When the object is resting on the bottom with no fluid beneath it this pressure difference disappears and thus there will be no buoyancy force acting on the object.

It seems that Sparks [1] has not defined properly what a buoyancy force is when deriving the effective tension concept. He jumps between the "two schools", as discussed in this thesis, creating misunderstandings and confusion regarding the effective tension concept.

Looking at figure 8.1 (figure 5.6 from chapter 5) we observe that the bar, when the tension at the top is equal to the weight of the bar, is in effective tension. Since the effective tension needs to be equal or higher than 0 of the entire riser length it is possible to lower the tension by an amount equal to the weight of the displaced fluid. One way of calculating this weight numerical is to use the term $p_e A_e$ as Sparks does in the effective tension concept; this is shown in figure 8.2.

This can lead to the interpretation that the riser is actually under influence of buoyancy despite that it is standing firmly on the bottom. As figure 8.1 and 8.2 shows, this is a wrong conception; there is no buoyancy force, as defined in chapter 4, acting externally on the bar.

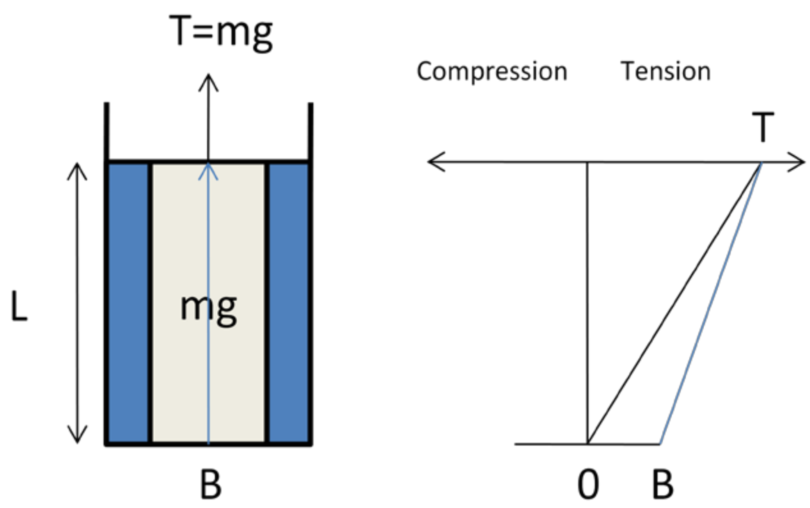


Figure 8.1: Solid bar standing on bottom

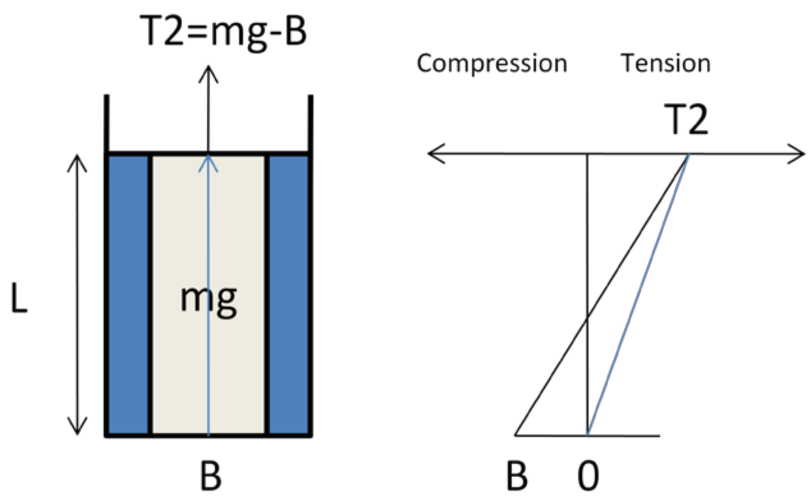


Figure 8.2: Solid bar standing on bottom. Effective tension equal to 0 at lower end

If we accept that the Bridgman experiment can be described using effective tension we can make an analogy, as shown in figure 8.3, to illustrate how effective tension can be interpreted in one dimension on the rod inside the pressure vessel.

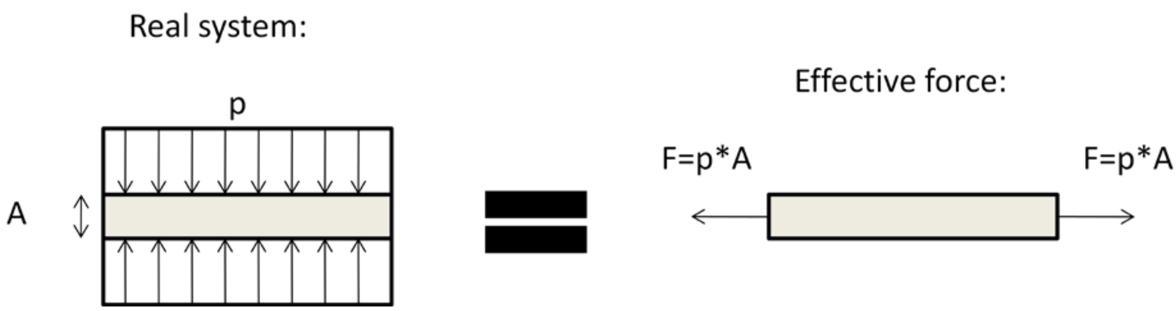


Figure 8.3: Interpretation of Bridgman experiment using effective tension

The experiment and the analogy proposed shows that the effective tension concept is not just some “fictitious” force, but it describes the internal stress state in the rod very accurate.

8.2 Recommendations for further work

8.2.1 Internal pressure term

Investigate the internal pressure term ($p_i A_i$) in the effective tension concept. An article [49] states that the internal pressure term is wrong and that it could potentially have cost the oil and gas over 500 million dollars.

8.2.2 Comparison of fracture surfaces

Conduct tensile strength test and Bridgman tests on different materials. Compare the fracture surfaces from both tests. It is also possible to map the difference in tensile strength and pressure needed to rupture the bar more accurate.

8.2.3 Study design software

Do an in-depth study on how design software (e.g. OpenFOAM, ANSYS) visualizes the results from the Bridgman experiment.

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Appendix A

Assumptions:

- Static friction is 3 times higher than the dynamic friction calculated
- Tolerances on rod OD and seal groove are neglected
- The outer diameter of the rods is assumed to be 8 mm exactly
- These calculations represents both materials

Friction force can be expressed by the following equation [50]

$F_f = N * \mu$ (A.1)

Where

$F_f = frictional\ force\ [N]$

$N = Normal\ force\ [N]$

$\mu = static\ (\mu_s) or\ kinetic\ (\mu_k) frictional\ coefficient$

The normal force between the sealing rings and the rod will increase as the hydrostatic pressure increases; this is shown in figure A.1. Due to this the friction force also will increase as shown by equation A.1.

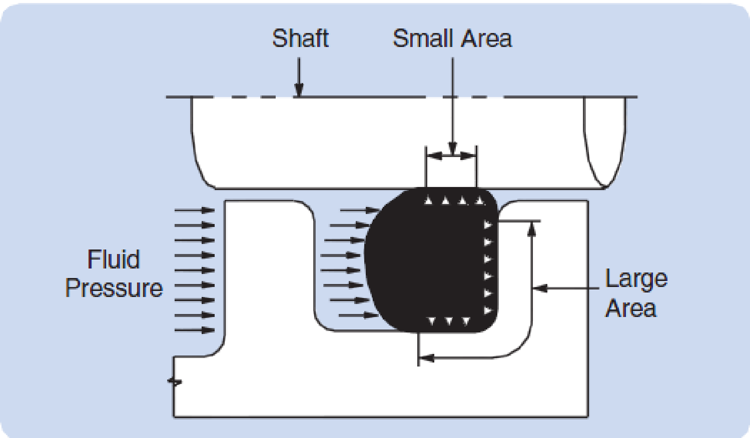


Figure A.1: Fluid pressure squeezes the sealing ring and increases the normal force between the rod and sealing ring [47]

To calculate the dynamic friction force between the rods and the sealing rings the following formulas will be used for a "Rod Groove" [47]:

$A_r = \left(\frac{\pi}{4}\right)\left[(A-1)^2{}_{max} - B_{min}^2\right]$ (A.2)

$L_r = \pi * B_{max}$ (A.3)

$A_r = \left(\frac{\pi}{4}\right)[A^2 - B^2]$ (A.4)

$F_c = f_c * L_r$ (A.5)

$F_h = f_h * A_r$ (A.6)

$F = F_c + F_H$ (A.7)

Where

- $A_r = Projected\ area\ of\ seal\ for\ rod\ groove\ applications\ [in\text{^}2]$
- $F = Total\ seal\ friction\ [lbf]$
- $F_c = Total\ friction\ due\ to\ seal\ compression\ [lbf]$
- $F_H = Total\ friction\ due\ to\ hydraulic\ pressure\ on\ the\ seal\ [lbf]$
- $f_c = Friction\ due\ to\ O-ring\ compression\ obtained\ from\ Figure\ B.3$
- $f_h = Friction\ due\ to\ fluid\ pressure\ obtained\ from\ Figure\ B.3$
- $L_r = Length\ of\ seal\ rubbing\ surface\ for\ rod\ groove\ applications\ [in]$
- $(A-1)^2_{max} = Rod\ gland\ groove\ inner\ diameter\ [in]$
- $B_{max} = Rod\ outer\ diameter\ [in]$

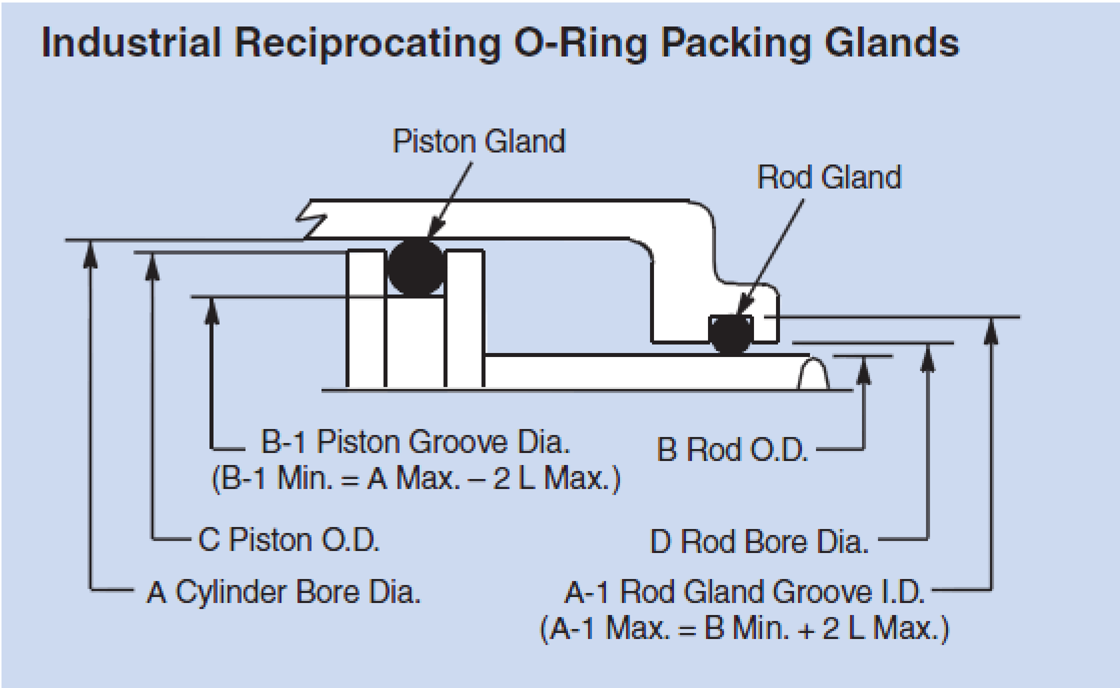


Figure A.2: Showing the different dimensions needed to do the calculations [47]

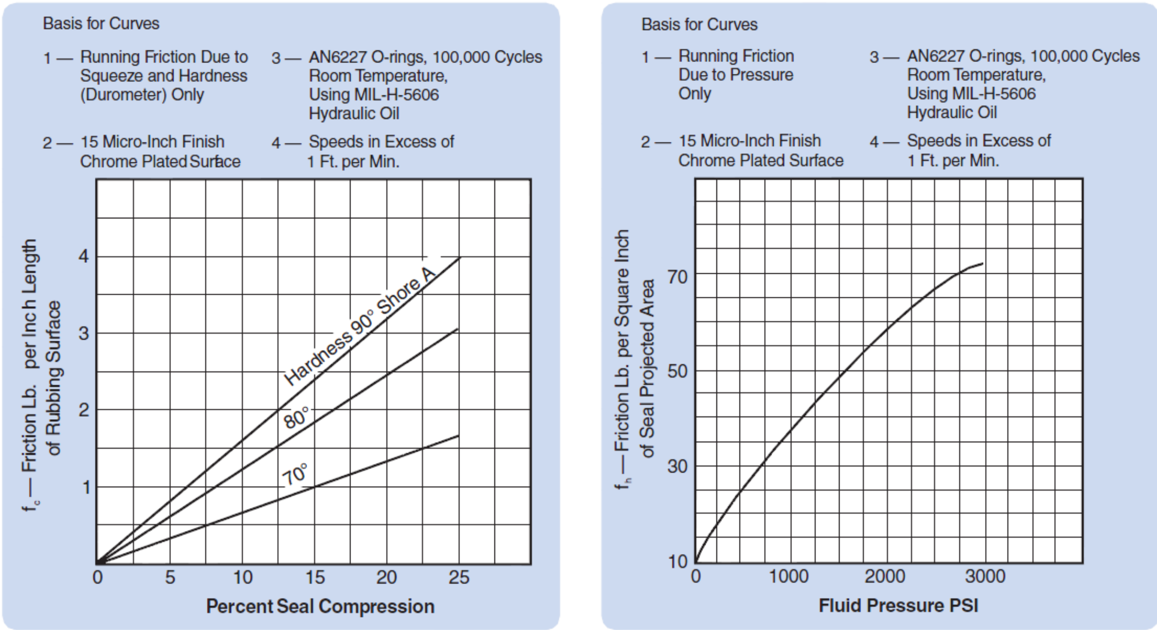


Figure A.3: Diagrams used to calculate the friction force [47]

Equation used to calculate percent seal compression:

$$\left(\frac{B}{ID_{seal}} - 1\right) * 100 \text{ (A.8)}$$

Numerical values is shown in table A.1.

Table A.1: Numerical values

$(A-1)_{max}$ [in]	B_{max} [in]	IDseal [in]	Hardness [Shore A]
0,591	0,315	0,272	85

Calculated friction due to compression is shown in table A.2.

Table A.2: Calculated friction due to compression

Compression [%]	f_c [lbf/in]	L_r [in]	F_c [lbf]
16	2,2	0,99	2,18

To be able to calculate the friction force due to hydraulic pressure on the seal in excess of 12000 PSI an estimated f_h-diagram have been made, see figure

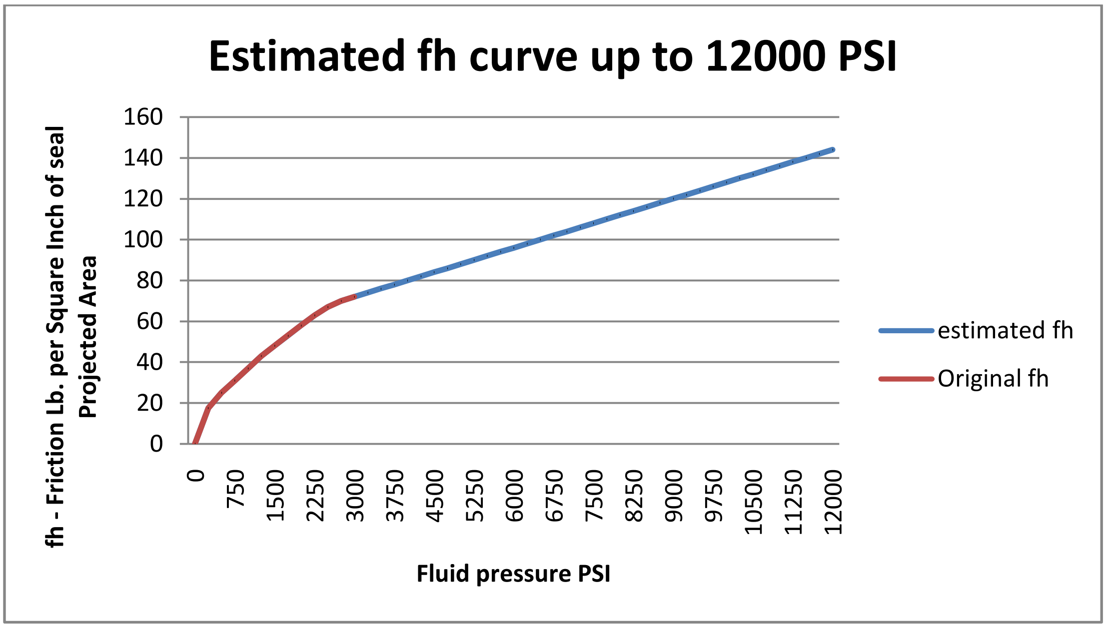


Figure A.4: Estimated fh curve

Calculated A_r value is shown in table :

Table A.3: Ar value

A_r $[in^2]$
0,20

Calculated F_h as function of pressure is shown in figure A.5.

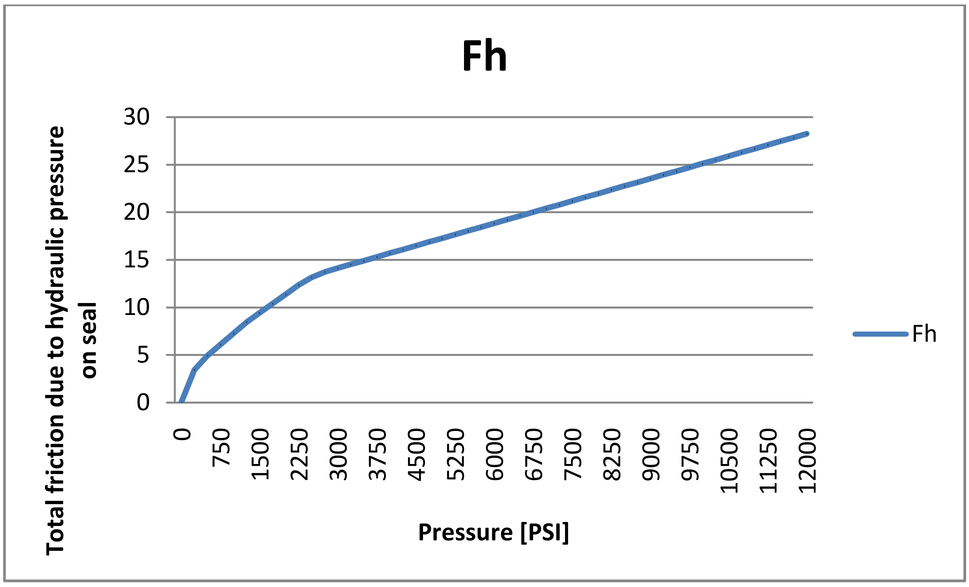


Figure A.5: Calculated Fh as function of pressure

Calculated dynamic friction force as function of pressure can be found in figure A.6.

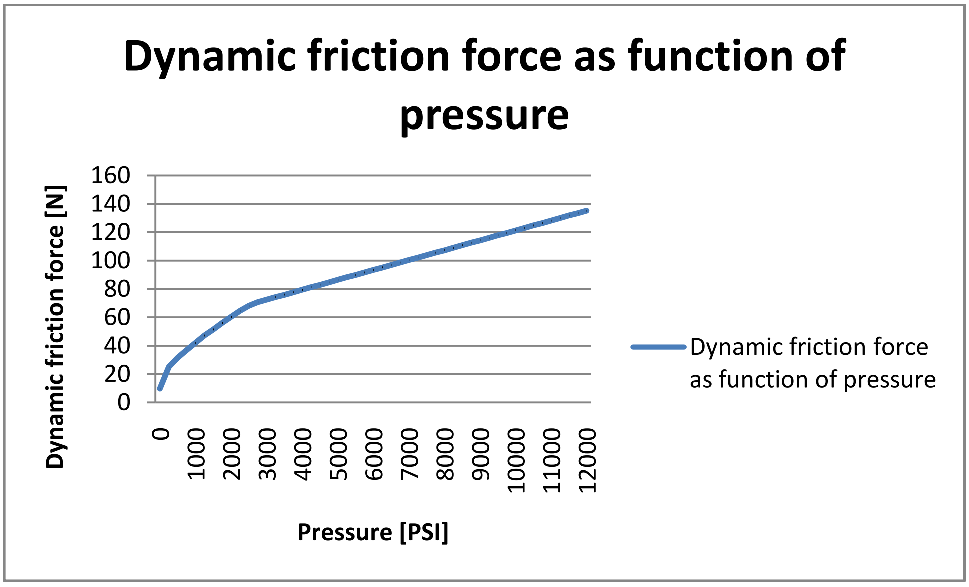


Figure A.6: Calculated dynamic friction force

Figure A.7 shows the static friction force. The pressure have been change to MPa in this figure.

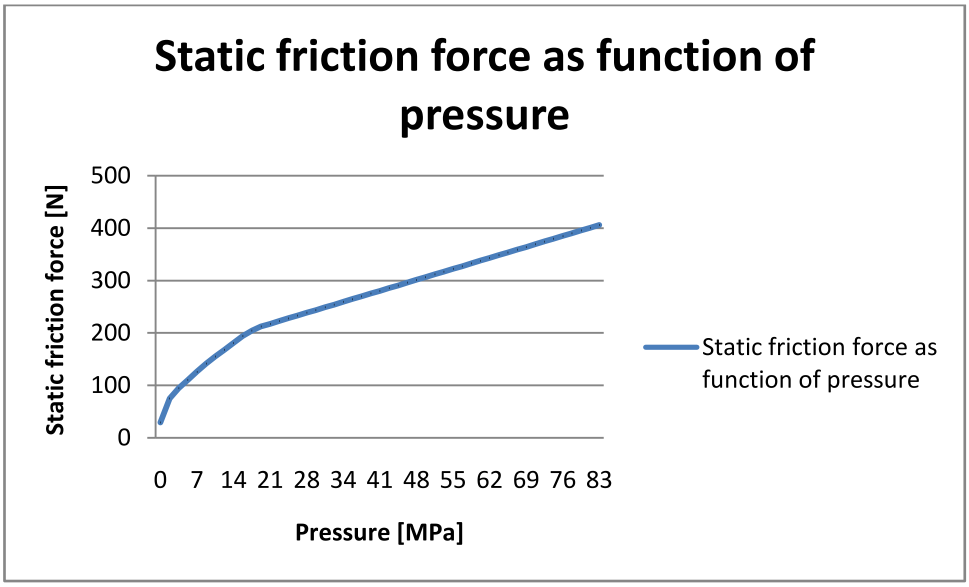
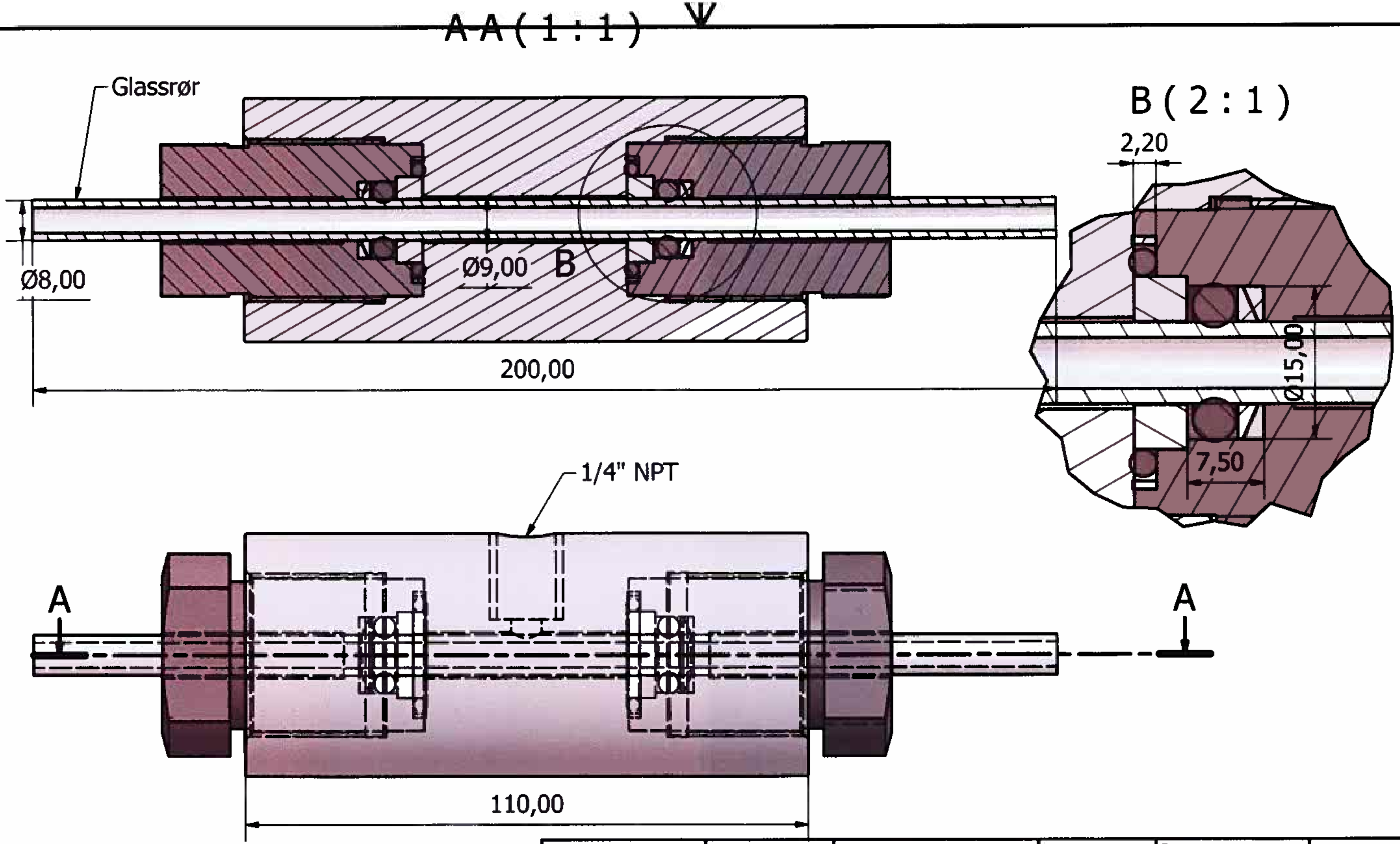


Figure A.7: Calculated static friction

Appendix B

Drawings of the Bridgman Chamber



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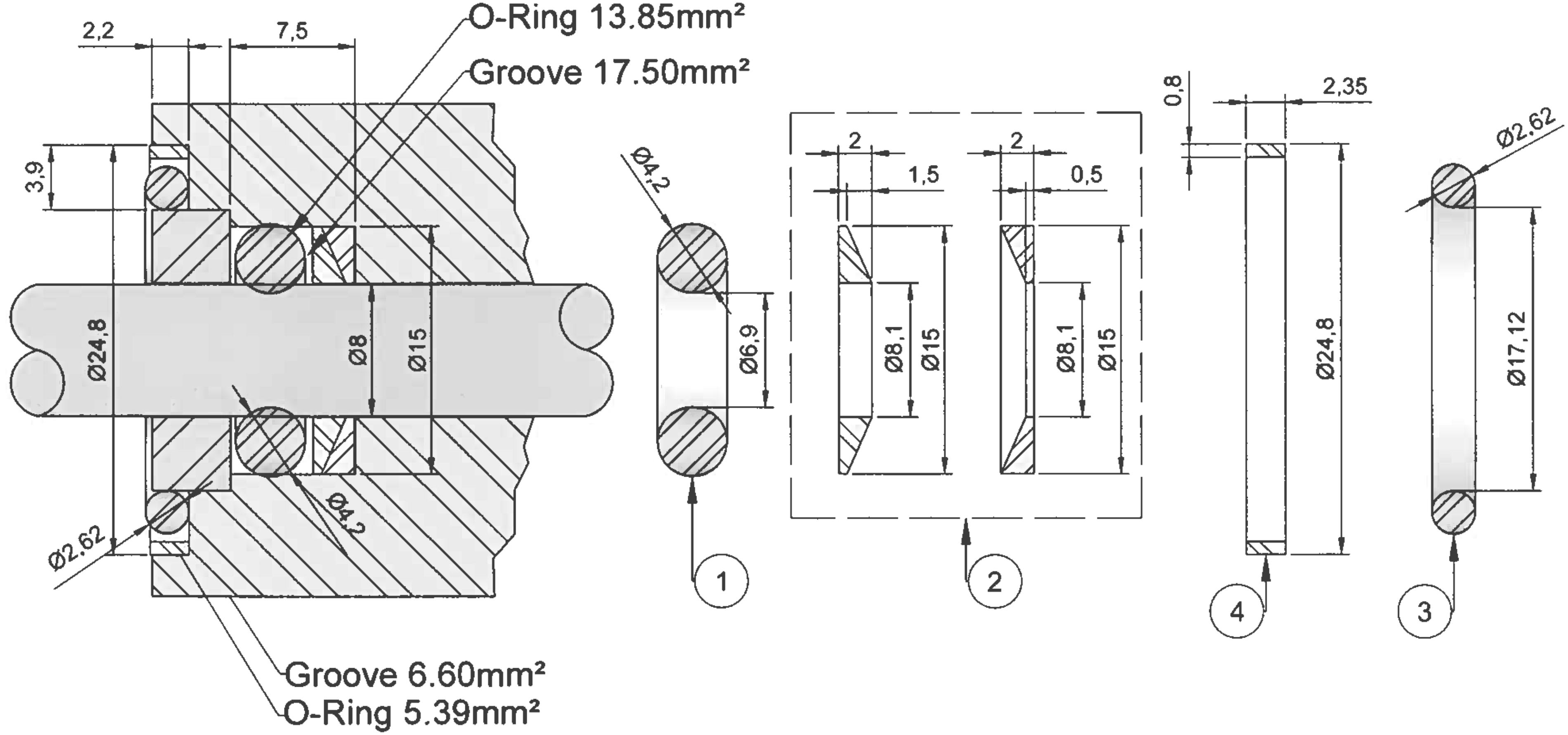

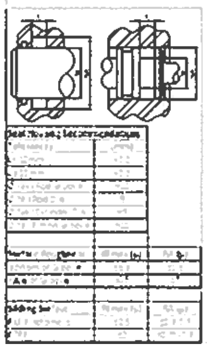

Itemref	Quantity	Title/Name Designation Material Dimensions etc	Article No./Reference
4	2	BU-Ring Ø24,8x23,2x2,35mm. POM-C/PA	_
3	2	O-Ring 17,12x2,62mm. NBR-90	10190001710262
2	2	BU-Ring Ø15x8,1x2mm. POM-C/PA	_
1	2	O-Ring 7,00x4,20mm. NBR-85	_

All Sealing Solutions

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Drawn by	PG	12 03 2013	Drawing no	Material: TBS
Checked by	N/A	N/A	99000100390000	Scale 4:1
Approved by			Customer IRIS	