Design and assembly of a standard compliant testing facility for locally manufactured industrial valves

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ABSTRACT

The greatest challenge facing the valve industry, is the access to affordable raw material. With the expected valve increase in South Africa, the industry identified the need for an increase of locally manufactured industrial valves that can reach the required quality at a competitive cost.

The quality of valves could only be ensured by the availability of required (necessary) testing facilities. Currently, these facilities are not generally available and the test procedures are not well known. The aim of this dissertation was to review international valve testing procedures, based on standards. Furthermore, the aim was the design and assembling of a first approach laboratory set-up valve testing facility, capable of utilising the above mentioned international valve testing procedures in order to test the quality and integrity of valves.

This testing facility was designed and manufactured to have the capacity of testing parallel flanged isolation valves with an outer flange diameter ranging between 165-350 mm and a pressure rating of up to ASME Class 300. To ensure that this facility conforms to the required specifications, a 2-Inch class 150 floating ball valve was subjected to standardised testing procedures (as per: API 598, ASME B16.34, ISO 5208, and MSS SP 61).

The results of the tests described above concluded that this test facility is capable of performing standardised hydrostatic testing procedures. Recommendations regarding the design and the system as a whole, are included in this dissertation.

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1 INTRODUCTION

An industrial valve is a device that regulates the flow of liquids, gasses or semi-solids through a pipeline or a system. In this chapter, the background regarding valves, together with the problem statement, aim and scope will be discussed.

1.1 Background

Looking at the valve market in South Africa, there are approximately 24 local manufacturers together with about 60 valve importers and resellers. The main clients include state owned enterprises, municipalities, and water boards (Merchantec Research, 2014).

Industrial valve demand with a forecast for 2017, is expected to rise approximately 5 % (The Freedonia Group, 2014). The greatest challenge the valve industry is experiencing, is the access to affordable raw material (Merchantec Research, 2014). Some imported valve products cost up to 60% less than locally manufactured products (Merchantec Research, 2014). Predominantly, the most valve manufacturing facilities are located in China and India (Tibbs, 2011).

With the expected increase in valve demand in South Africa, the industry identified the need for an increase of locally manufactured industrial valves that can reach the required quality at a competitive cost. With the increase in globally sourced products and much reduced domestic manufacturing, the demand for the testing of valves has risen (United Valve, 2011). A lead project funded by Technology & Human Resources for Industrial Programme (THRIP) has been initiated at the NWU Engineering Faculty in the Materials, Vibration, and Manufacturing Division, in collaboration with Ametex Pty Ltd. as an industrial partner. This project aims to investigate the possibility of the development of a valve that will meet the demand mentioned above.

In order to effectively test a valve, there must first be an established testing procedure, combined with accepted criteria or performance standards that a valve is expected to meet or exceed. If a valve doesn't comply with the requirements of these standards, the integrity thereof is questioned. There are a series of different valve standard generating organisations, of which the most relevant organisations for the investigation proposed by this study are: American Petroleum Institute (API), American Society of Mechanical Engineers (ASME), British Standard (BS), International Organisation for Standardisation (ISO), Manufacturers Standardisation Society (MSS) and South African National Standard (SANS).

The expected future increase in valve demand over the years and the critical importance of their quality necessitates the need for the design and assembling of a testing facility that is accessible to the local valve industry.

1.2 Problem Statement

The quality of industrial valves could only be ensured by the availability of required (necessary) testing facilities. Currently, these facilities are not generally available and the test procedures are not well known.

1.3 Aim

In order to find best practise configurations, procedures, and limits, a review of international valve testing procedures was conducted. Furthermore, the aim is to design and assemble a first approach laboratory set-up valve testing facility capable of utilising these existing valve testing procedures in order to test the quality and integrity of locally manufactured industrial valves.

2 LITERATURE REVIEW

In this chapter, valves and valve testing, with reference to applicable standards and required testing procedures, were studied. In addition, relevant industrial shut-off valve testing standards were investigated with the aim to identify the required testing procedures and instrumentation. The identified required instrumentations are then to be implemented in a locally designed and manufactured industrial valve testing facility, capable of performing required valve testing procedures.

2.1 Valve Components, Assemblies, and Ratings

Due to the diversity of valve applications, a series of different types of valves were developed, each having its own advantages and disadvantages. The most common types of valves used in the industry today are globe -, gate -, ball -, plug -, butterfly -, diaphragm -, check -, pinch -, and safety valves (Department of Energy, 1993). Each of these valves have been specially designed to meet specific needs (Department of Energy, 1993). Regardless of the type of valve, the basic components of all valves are the body, bonnet, trim, actuator and packing. Typically, the trim includes a disk, seat, stem, and sleeves needed to guide the stem (Department of Energy, 1993). Illustrated in the figure below are the mentioned components.

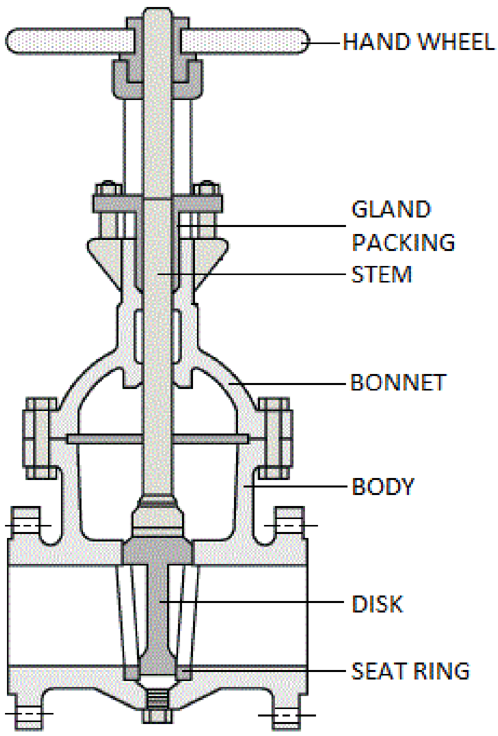


Figure 2.1 Section view representation of a gate valve, showing and identifying its basic components (Modified after Valve Products, 2010)

Each valve has a pressure temperature rating, which specifies the designed internal pressure that the valve is capable of withstanding at a certain temperature. This pressure temperature rating is based on the valve material and its operating temperature. This pressure temperature rating can either be given as Nominal Pressure (PN) or as ASME Class rating. Below are the ceiling pressures by class, in bar, at a temperature range between -29 to 38 degrees Celsius (ASME, 2013).

Table 2.1 – Pressure Temperature rating by class, in bar, at a temperature range between -29 to 38 degrees Celsius (ASME, 2013)

Class	150	300	600	900	1500	2500	4500
PN *	20	51.7	103.4	155.1	258.6	430.9	775.7

*Nominal Pressure

In addition to different types of valves, there are different types of end connections for these valves. These connections depend on the pressure and temperature of the working fluid as well as the maintenance and dismantling of the pipeline (Stoneleigh Engineering Services, 2012). The common types of end connections include the following (Stoneleigh Engineering Services, 2012):

- Screw Valve End Connections
- Flanged Valve End Connections
- Socket Weld Valve End Connections
- Butt Weld Valve End Connections
- Compression Valve End Connections
- Capillary Valve End Connections
- Socket Valve End Connections
- Spigot Valve End Connections

To ensure that the manufactured or repaired valves conform to the required standard(s), there are a series of companies that manufacture standardised valve testing facilities. In the following section, some of these companies are discussed.

2.2 Industrial Valve Testing Facility Manufacturing Companies

There are only a handful of companies that manufacture facilities that are capable of performing industrial valve tests. These facilities conform to industrial valve testing standards that ensure high-quality products. These companies are able to design, manufacture and assemble test rigs to meet the client's specific requirements.

Italcontrol (Pty) Ltd. is an Italian valve testing facility manufacturing company with valve testing experience dating back to the beginning of the 90's (Italcontrol, 2012). With the cooperation from their clients, this company offers a complete line of high-quality, high hydraulic and pneumatic pressures testing facilities (Italcontrol, 2012). These testing facilities are capable of testing in accordance with standards such as API, ANSI, BS, ISO, and EN (Italcontrol, 2012).

WIKA (Pty) Ltd. is a global market leader in the measuring of pressure and temperature, as well as level measuring technology (WIKA, 2017). Their philosophy is to constantly improve current devices and therefore, their products are of the highest standard. Approximately 600 million high-quality WIKA measuring devices are used around the globe (WIKA, 2017). In addition to measuring technology, they have the capacity to provide valve testing facilities capable of producing high pressures of both liquid and/or gas media to test the endurance of valves. These testing facilities generate a hydrostatic test pressure by means of using a filling pump together with a booster. For pneumatic tests, the required pressure rating is obtained using a compressor, pneumatic booster or gas supplied from external gas bottles (WIKA, 2014).

EFCO (Pty) Ltd. is an internationally recognised company with branches in China, India, Moscow, France, USA, and Germany. Their project range include, among many others, portable and stationary machining and repairing machinery for valves, flanges and pipelines, surface grinding and lapping machines, valve test benches, and mobile or stationary workshops (EFCO, 2015). Valve testing facilities produced by this company consist of clamping units, control panels, and measuring equipment (EFCO, 2015). These testing facilities accommodate testing procedures for hollow bodies, such as pipes, valves, and flanges (EFCO, 2011). The testing procedures commonly describe tests such as shell tests, closure tests, as well as set pressures for safety relieve valves in particular (EFCO, 2015).

Ventil (Pty) Ltd. is a company that produces test units for shut-off valves, safety valves, and control valves. Their experience exceeds 50 years of design and they manufacture high-quality industrial valve repairing and testing facilities (Ventil, 2012). Today, these facilities are used by valve manufacturing companies to ensure high-quality products. The Shut-Off valve testing facility is capable of testing both the integrity and the performance of any Gate -, Globe -, Plug -, Ball -, Check -, Slide - and Butterfly valve, with end connections such as Flange -, Thread - or Weld connections in the range DN6 – 3000 mm (Ventil, 2012). In addition to this, the company is also fully equipped with manual or automated test systems for Body and Seat leakage testing procedures (Ventil, 2012). These facilities are capable of generating a liquid test pressure of 0-2000 bar and a gas test pressure of 0-1000 bar.

Drawing from the expertise of the above mentioned companies, the optimum circumstances under which valve testing could take place would require the design of a testing facility capable of performing and conforming to testing procedures. The layout of the facility will depend on the size and the type of valve to be tested. In the following section, different valve testing facilities with regard to their designed configuration will be discussed.

2.3 Valve Testing Configurations

Research shows that there are a series of different valve testing facility configurations. Relying on a leader in the field, such as Italcontrol, these configurations include the following (Italcontrol, 2012):

- Horizontal test bench series (HTS)
- Vertical test bench series (VTS)
- Tilting test bench series (TTS)
- Multi-station test bench series (MTS)
- Safety relieve valve test bench series (SVA)
- Special test bench series (STS)
- Actuator test bench series (ATS)

Different configurations have different advantages and disadvantages. The table below contains a brief discussion of some of these advantages and disadvantages.

Table 2.2 – Table discussing some advantages and disadvantages of different test bench configurations (Italcontrol, 2012)

Configuration	Advantages	Disadvantages
Horizontal test benches (HTS)	Easier to place the test valve into position. Easier to assemble some particulars such as bellows, actuators, steering's etc. (Italcontrol, 2012).	Venting in this position is difficult, however, some HTS are equipped with one or two screws, in symmetrical or inclined positions (30° or 45°), which can help with the venting of the test specimen (Italcontrol, 2012).
Vertical test benches (VTS)	Air is easily vented when filling the test specimen with liquid.	Handling and placing the test specimen into position may be difficult.
Tilting test benches (TTS)	Can perform tests in the horizontal, vertical and at inclined planes.	
Multi-stations test benches (MTS)	Can test a series of valves simultaneously.	

Some configurations, such as the STS, are designed and manufactured based on customer specifications. These specifications may address or eliminate a possible disadvantage of a test bench configuration. The SVA and ATS test bench configurations are beyond the scope of this study and are not included in Table 2.2 above.

2.4 **Valve leakage and system failures**

In terms of valve leakages, two types commonly occur, namely: fugitive emissions, and leakage through the valve (Contractor Unlimited, 2017). According to Contractor Unlimited, “Valves are considered to be the major contributors to fugitive emission losses” (Contractor Unlimited, 2017). Depending on the type of leak and the leakage rate thereof (if a leak is present), the valve has failed.

Most valve failures occur due to a high-pressure drop. According to Skousen, this phenomenon holds complications such as cavitation, flashing, choked flow, high velocities, high noise levels, and vibration (Skousen, 2004). These complications can cause immediate damage to the valve, such as erosion or cavitation damage to the body and trim.

The primary mode of a shut-down valve failure is through leakage (Meland *et al.*, 2011). According to Meland *et al*. and Contractor Unlimited, the principal causes of through leakage include the following:

- The valve or its seals become jammed due to the presence of foreign objects or a deposited material;
- corrosion or erosion of the valve and its components;
- the seal of the valve is blown out of position;
- the seals within the valve are damaged;
- the valve seats are damaged;
- insufficient closure of the valve, due to the failure of the valve's actuator;
- external leakages;
- incorrect closure speed; and
- insufficient actuator travel.

A small leakage rate is permissible in the valve industry. However, if the leakage rate is greater than the permissible rate, the valve has failed. This permissible leakage rate only applies when considering through leakage. Leakage through the pressure boundary is not permitted. This pressure boundary includes the body, the bonnet, or cover and all gasketed joints (ASME, 2013). However, when testing, leakage through the valve stem or stem packing will not be cause for rejection.

The leakage rate of a valve is commonly measured in mL/s of bubbles/s, depending on the testing medium. If the testing medium is liquid, the leakage rate will be in mL/s (cc/s), with 1 mL being equivalent to 16 drops (API, 2009; MSS, 2009). If the testing medium is gas, the leakage rate will be in bubbles per second.

There are a series of different leak detection techniques used in the industry today. Each having its advantages and disadvantages. According to Nayyar, some common leak detection techniques include (arranged from less sensitive to most sensitive) (Nayyar, 2000):

- Hydrostatic leak detection testing;
- Pneumatic or gaseous-fluid leak detection testing;
- A combination of pneumatic and hydraulic leak detection testing;
- Initial service leak detection testing;
- Vacuum leak detection testing;
- Static head leak detection testing;
- Halogen and helium leak detection testing.

The most prevalent and favoured leak detection technique in the industry today is hydrostatic leak detection (Nayyar, 2000). Compared to other pneumatic or gaseous-fluid leak detection techniques, the hydrostatic leak detection technique is far safer, this is due to the fact that the test medium is incompressible. Depending on the system application and the system material, the preferred hydrostatic leak detection medium is water (Nayyar, 2000).

The test fluid temperature, with reference to liquid, ranges from approximately 0 to 85 degrees Celsius. Some standards such as the SANS 664-1 are more specific than other, which states that the test fluid should be in the range of 23±2 degrees Celsius (SANS, 2011a), while other such as the ASME B16.34 states the test fluid temperature should not exceed 50 degrees Celsius (ASME, 2013). Moreover, API 598 states that test fluid temperature should be in the range of 5 to 50 degrees Celsius (API, 2009).

The easiest leak detection method was found to be through mass or volume balance methods (Ostapkowicz, 2016). A disadvantage of this technique is the lack of the precise location of the leak (if a leak is present). Companies such as WIKA manufacturers valve testing facilities equipped with leak detection and bubble/dropper counter systems (WIKA, 2014).

When considering a hydrostatic leak detection technique, it is of great importance that all the air is vented from the system before a pressure test commences. According to Nayyar, trapped air within a pressure system may result in complications such as (Nayyar, 2000):

- a greater safety hazard;
- a water leak may be hindered if air is trapped at a leak location; and
- the pressurising time of the system will increase.

When considering pressure generating systems, the first components to fail are generally the pump or the pipe system. Some hydraulic system failures occur due to inadequate use of components, incorrect installation of components, hydraulic transients, and seal perishing or bursting, among others.

Hydraulic transient occurs whenever the kinetic energy carried by the fluid is converted into strain energy in the pipe walls (Boulos *et al.*, 2005). In practice, hydraulic transients are usually more severe in control valves, pump stations, high-elevated areas, low static pressure locations, and in isolated areas that are distanced from overhead storage (Boulos *et al.*, 2005). Some effects of a hydraulic transient include (Grobbelaar, 2015):

- Change of valve opening;
- Starting or stopping of pumps;
- The opening of check valves, air release valves, pressure reducing valves, and pressure relieve valves;
- Pipe leak or – rupture;
- Valve failure;
- Improper filling, flushing, or removal of air from pipelines;
- Trapped air pockets;
- Flow component damage;
- Change in power demand of hydraulic turbines.

Due to the incompressibility of water, the effect of a hydraulic transient on the internal pressure of a hydraulic system can be 10 times higher than the normal operating pressure (Taljaard, 2012). Techniques used to mitigate transient conditions include (Boulos *et al.*, 2005):

- Installing of surge protection devices;
- Improving valve and pump operating procedures;
- Alter physical pipeline characteristics;
- Design special facilities for filling, flushing, and releasing air from pipelines;
- Increase the pressure class of the pipeline.

According to Mays and Boulos *et al*.,in practice, a piping system's pressure rating should accommodate a predicted surge incidents' pressure (Boulos *et al.*, 2005; Mays, 2000). In terms of pressure vessels and the welding thereof, the use of the correct material is of great importance. If an insufficient material is used, the heat affected zone (HAZ) can become susceptible to brittle cracking, which evidentially entail hazardous circumstances.

2.5 Standard complying tests and descriptions

As stated in Section 1.1, most valve manufacturing facilities are located in China and India. Most of these facilities are not firmly rooted in western quality prospects (Tibbs, 2011). According to Greg Johnson, president of Houston-based United Valves, during a presentation at VMA's Technical Seminar on the third of March 2011 (Tibbs, 2011): "Chinese NDE personnel, techniques, and equipment are still not universally equal to western standards, nor do they fully understand western NDE requirements and specifications." To ensure that manufactured valves conform to the required quality, standardised tests and inspections need to be conducted (Bradbury, 2017). This is also to be considered the aim of this project.

According to WIKA’s valve test benches catalogue: “… 50 to 60 % of all the emissions in industrial processes are caused by leaking valves” (WIKA, 2014). This alone is significant enough to ensure that products being tested, conform to the requirements specified in valve standards. If a product delivered by a company or an organisation doesn’t conform to the necessary standards, product commercialising should not commence.

There are a series of different standard generating organisations such as API, ASME, ANSI, ISO, MSS, AWWA, SANS, and others. These generated standards are to be revised every couple of years. The three most commonly used standards in the oil, gas and petrochemical industry today are the API 598, ANSI FCI 70-2 and MSS-SP-61 (Contractor Unlimited, 2017).

Different types of valves are commonly tested to a variety of standards. According to David Bayreuther, Vice-President of engineering at Metso, the reason for the great multitude of valve testing standards is due to confusion within standard generating organisations (Bayreuther, 2016). This statement, however, proves that gaps in testing procedures can still be present, which in turn highlights the need for this study. As stated earlier in this chapter, there are a series of different standard generating organisations. The ASME Boiler and Pressure Vessel Code has provided the basis for these standards, with reference to the PVF (Pipe, Valve, Flange) industry (Valve Magazine, 1996). The following table indicates common valve types and the related testing standards (Global Suport Line, 2013).

Table 2.3 - Common valve types and their related testing standards (Global Suport Line, 2013)

Valve type	Common test standard
Steel ball, gate, globe and check valves	API 598
Steel ball, gate, globe and check valves	BS 6755*, ISO 5208 (EN 12266-1)
Cast Iron gate valves	API 598, MSS SP – 70
Bronze gate, globe and check valves large than NPS 24"	MSS SP – 80
Pressure seal gate, globe and check valves	ASME B16.34
Pipeline valves	ASME B16.34
Cast iron checks	API 6D, ISO 5208
Cast iron globes	API 598, MSS SP - 71
Cast iron plugs	API 598, MSS SP - 85
Cast iron plugs	API 598, MSS SP - 78
Steel ball valves	API 598
Steel butterfly valves	API 598
Cryogenic valves	API 598, BS 6364
Control valves	FCI 70-2, ISA-S75
Pressure relief valves	API 527, ASME PTC 25

*ISO 5208 (EN 12266-1) supersedes BS 6755

Research shows that the most common shut-off valve tests are the Shell –, Closure -, and Backseat tests. Upon further study, it was found that some additional valve tests included:

- Gate strength test,
- Torque Test,
- Endurance test.

Discussed in the following subsections, are brief descriptions of the above mentioned tests. Attention was given to the procedures, and the apparatus required in performing these tests.

2.5.1 Shell Test

The aim of this test is to ensure that the valve's body is capable of withstanding the required generated internal pressure rating. During this test, the valve's obturator needs to be in a partially open position with one end connected to the pressure source, and the other blanked off. The valve is then to be pressurised and held for a certain time duration (specified in related standards). The test fluid of this test is normally liquid, but gas testing is sometimes requested by clients. When considering a liquid test, venting of all the air within the valve should be done prior to a hydrostatic pressure test.

If the test fluid is gas, the leak detection is commonly done by means of submerging the pressurised valve as a whole in a body of water. If bubbles form, then there is a leak somewhere on the body of the valve. This method, however, doesn't indicate the precise location of the present leak. According to ASME, visually detectable leakage through the pressure boundary is not permitted (ASME, 2013).

With reference to the ASME B32.16 (Valves- Flanged, Threaded, and Welding End), each valve shall be tested at a gauge pressure no less than 1.5 times the pressure rating, rounded off to the next higher 1 bar (ASME, 2013).

2.5.2 Closure Test or Seat Tightness Test

The aim of this test is to ensure that the valve’s obturator is capable of withstanding the required internal pressure rating. During this test, the test valve is partially filled with the testing fluid (which is normally water), with one end of the valve connected to the pressure source and the other end free to atmospheric conditions. Once half of the valve is filled with the testing medium, the valves obturator is then brought to the fully closed position. With the valve in the fully closed position, the pressure within the valve is raised to the required pressure rating and held constant for a specified time duration. According to ASME, the required hydrostatic gauge pressure should be no less than 110% (or 1.1 times) of the designed valve pressure rating (ASME, 2013).

The pressurised valve is then examined for any leakages. If a through leakage is present, the leakage rate should then be determined. Following the determination of the leakage rate (if a leak is present), the valve’s opposite end should be tested by means of the above procedure.

According to ASME, this test is to commence after the shell test, except for valves NPS 4 and smaller with rating Class 1500 and lower. The closure test may precede the shell test when a gas closure test is used (ASME, 2013).

2.5.3 Gate Strength Test

The aim of this test is to test the strength of the obturator. The procedure of this test is similar to the test described above. The main difference between this and the above-mentioned test is the generated pressure rating. During this test, the generated gauge pressure rating should be at least 1.5 times the working pressure of the valve (SANS, 2011b).

2.5.4 Stem Backseat Testing

According to Kate Kunkel, senior editor of VALVE Magazine, more than 75% of valve leakages come from the valve's stem (Kunkel, 2014). Therefore, the significance of this test.

The aim of this test is to ensure that the valve submitted to the test complies to the required backseat pressure. This type of testing is only for valves with a stem backseat feature. This test is typically done prior to the shell test. During this test, the valve obturator is in a partially open position, with one end connected to the pressure source, and the other blanked off. The valve body is then pressurised and held for a certain time duration (specified in related standard). Commonly, the generated gauge pressure is 1.1 times the working pressure of the valve.

The valve is observed for any leaks, especially around the valve stem. The test fluid of this test is normally liquid but gas is sometimes requested by clients. If the test fluid is gas, the leak detection procedure is similar to the procedure described in the section regarding the shell test.

2.5.5 Torque Test

The aim is to ensure that the valve subjected to this test, complies to the required operating torque. For the purposes of this test, it is of great importance that the valve subjected to testing, is firmly fixed to the testing facility. This will ensure test accuracy. As per SANS 664-1 specification, there are two types of torque tests to be performed on a valve. These tests are the Maximum Operating Torque Test (MOT), and the Maximum Strength Torque Test (MST). These tests are discussed in more detail in the sections following. The amount of torque to be applied to the stem of the valve can be found in associated standards.

MOT Test

Valves that have mechanically operated obturators should be able to successfully open and close, and be made leak proof by means of applying a specified amount of torque to the valve's stem. By partially opening the valve obturator and by the blanking of one end of the valve, using a blank flange containing a vent valve, and connecting the other end of the valve to a hydrostatic pressure source, the valve can be filled with water while ensuring no air is trapped within the valve. Once the valve body is filled with liquid, the valve obturator can be closed by means of applying specified torque.

With the valve in a fully closed position, the internal hydrostatic pressure is raised to the required pressure rating, and held for a specified time duration (see associated standards).

Subsequently, the valve (still under pressure) is then opened and closed by means of applying a specified amount of torque. This torque is not to exceed the MOT. This procedure is then to be applied to the opposite end of the valve.

MST Test

The aim of this test is to ensure that the valve subjected to the test (in the fully open, as well as the fully closed position), can withstand the maximum torque without the impairment of any functional capabilities. Furthermore, when the MST is applied to the valve spindle (with the valve in the fully open position), the magnitude and time duration of the torque applied should be equal to that specified in associated standards.

When the MST is applied to the valve spindle (with the valve in the fully closed position), the principal is similar to that of the closure test. The valve must be filled with water and fully closed by means of applying the MST to the spindle. When filling the valve with water, it has to be ensured that no air is trapped in the valve cavities. Once all the air is vented from the valve, the hydrostatic pressure of the valve end connected to the hydrostatic pressure source, is raised to the required pressure rating and held constant for a specific time duration. The inspection of the valve is done subsequent to the alleviation of internal hydrostatic pressure.

2.5.6 Endurance Test

The aim of this test is to demonstrate the performance of the valve (SANS, 2011b). The procedure regarding this type of testing is similar to the procedure described in the section regarding the Closure Test or Seat Tightness Test. The valve undergoes a series of cycles during this test and each cycle consists of the valve being closed, pressurised and opened.

The valve is to be placed in the horizontal plane with the obturator in the fully open position. One end of the valve must be connected to the pressure source with a gooseneck pipe connected to the other end, as per SANS standards (SANS, 2011b). It is then filled with the test fluid and all excess air is vented from the valve. Once all the air is vented, the valves obturator is brought to the fully closed position by means of applying torque to the spindle of the valve. This applied torque should not exceed the MOT (SANS, 2011b).

Once the obturator is in the fully closed position, the internal hydrostatic pressure is raised to the required pressure rating and held for a specified time duration. Once the required pressure and the pressurised duration are reached, the valve is to be opened and the next cycle commences.

The instrumentation needed in performing these tests and testing procedures include a pressure source, a leakage rate calculator, a torque instrument (as described in Section 2.5.5), blank flanges, a timer and a cycle counter. Pressure and temperature gauges are also essential.

2.6 Summary and scope

After careful consideration of the literature review, practical configurations and safety precautions, the choice of test bench configuration is a vertical configuration. With safety in the forefront, the relevant testing procedures are done using water as testing medium.

The scope of this study is to design and manufacture a standard compliant hydrostatic industrial valve testing facility capable of performing standardised testing procedures. This facility will be designed to have the capacity of testing valves with an outer flange diameter ranging between 165-350 mm, and a pressure rating of up to ASME Class 300. To ensure that this facility conforms to these specifications, a 2-Inch Class 150 floating ball valve was subjected to standardised testing procedures (as per standards API 598, ASME B16.34, ISO 5208 and MSS SP 61).

3 DESIGN

This chapter will comprise of designs, calculations, and simulations regarding the critical components of the designed hydrostatic pressure testing facility. These simulations were done using SolidWorks Simulation software. All detailed drawings and calculations can be viewed in Appendix A and B, respectively.

As in the scope of this study, isolation valves with an ASME pressure Class rating of up to 300 should be tested, using water as the testing medium. Below is an isometric model of the designed facility, capable of conforming to the specifications discussed in the scope of this study.

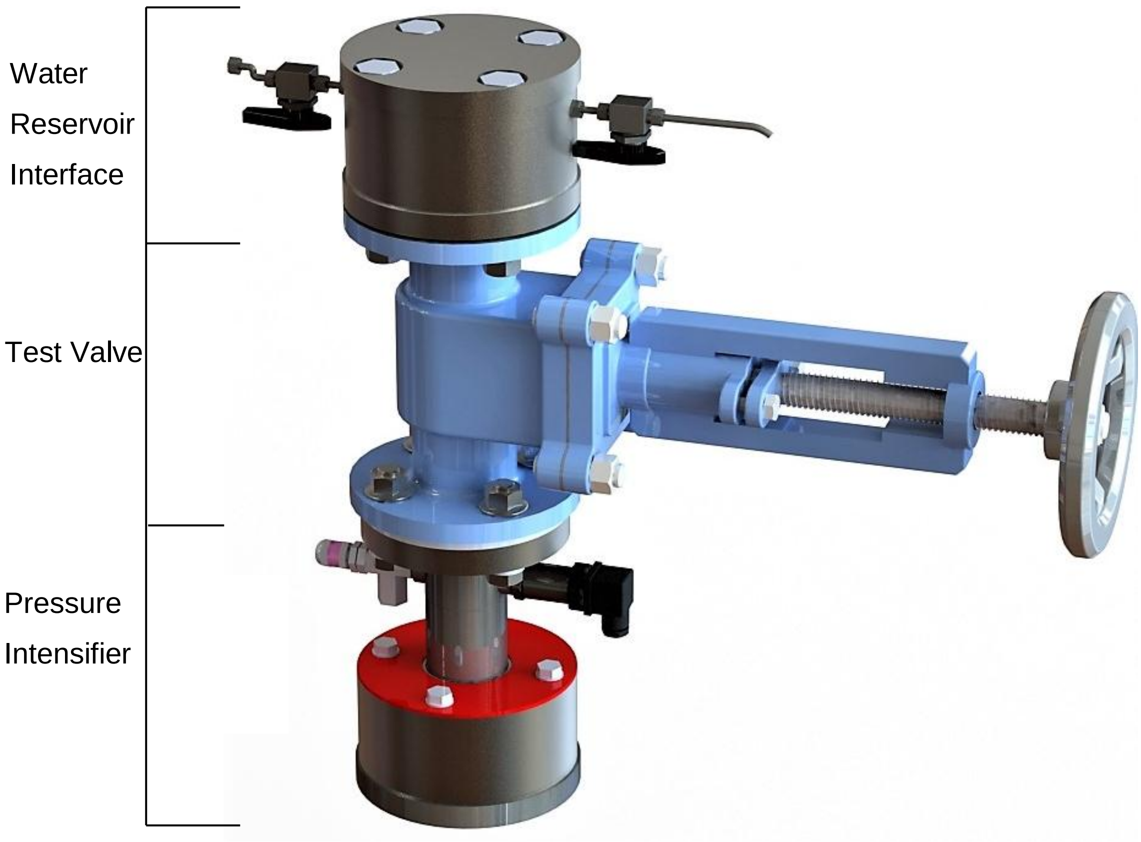
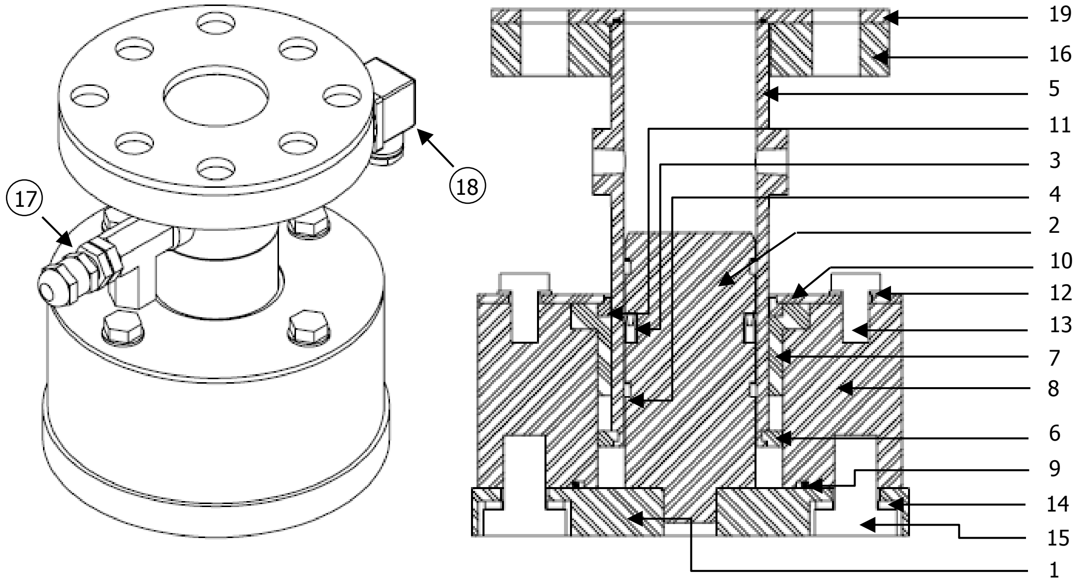


Figure 3.1 – 3D model of the designed Pressure Intensifier and the Water Reservoir Interface mountings, connected to a test valve

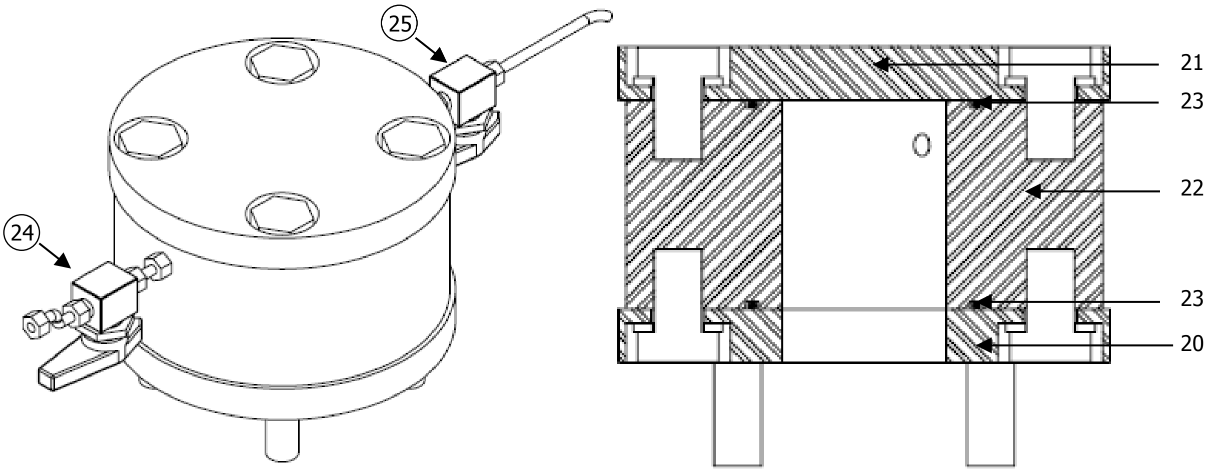
As seen in Figure 3.1 above, the designed system comprises of two subassemblies - the Pressure Intensifier and the Water Reservoir Interface. These subassemblies are placed on opposite ends of the test valve. Figure 3.2 shows a section view of the designed pressure intensifier, in conjunction with the index numbers.



Pressure Intensifier					
Component	**No**	**Component**	**No**	**Component**	**No**
Piston Base	1	Piston Sleeve Housing	8	M16 Bolt	15
Piston	2	Piston Sleeve Housing O-Ring	9	Connection Flange	16
Piston Seal	3	Perspex Lid	10	Safety Relieve Valve	17
Bearing	4	Wiper Seal	11	Pressure Transmitter	18
Piston Sleeve	5	M10 Washer	12	O-Ring Flange	19
Stopping Ring	6	M10 Bolt	13		
Guide Bush	7	M16 Washer	14		

Figure 3.2 - Section view of the designed pressure intensifier, together with index numbers allocating this subassembly's components

When an axial load is applied to this mounting, the Piston Sleeve (component 5) slides over the Piston (component 2). This sliding feature enables this mounting to generate the required pressure rating. For safety purposes, a Safety Relieve Valve (component 17) is connected to the pressure intensifier. The generated pressure is monitored using a Pressure Transmitter (component 18). Below is a section view of the designed water reservoir interface, in conjunction with the index numbers.



Water Reservoir Interface					
Component	**No**	**Component**	**No**	**Component**	**No**
Bidirectional Flange	20	Water Reservoir Interface Housing	22	Water Inlet Ball Valve	24
Base Flange	21	Water Reservoir Interface O-Ring	23	Water Outlet Ball Valve	25

Figure 3.3 - Section view of the designed water reservoir interface, together with index numbers allocating this subassembly's components

Depending on the type of test to be performed by the user, the ball valves (components 24 and 25) will be either closed or open. When considering a hydrostatic shell test, both ball valves are closed (once all the air is vented from the system). If a hydrostatic closure test is performed, both these valves should be in the open position. If through leakage is present, drops will form at the end of the pipe connected to the ball valve (index number 25).

Calculations and simulations regarding the critical components such as the Piston and the Piston Sleeve were done. In the following sections, these results are presented and discussed. Additionally, components such as the Guide Bush, Seals, Bearings, Gaskets, Safety Relieve Valve, Swagelok Ball Valves, Data Acquisition Models, Pressure Transmitter, and Thermocouple are also discussed.

3.1 Piston

The minimum designed pressure rating that this system should be able to generate and withstand equals 1.5 times the ASME Class 300 rating. Using CES Edupack Material Selection software, heat treated K110 Tool steel was indicated as the best suited material for this application.

During a pressure test, the Piston is subjected to high compression loads and for this reason, the Rockwell C hardness (HRC) needs to be in a required hardness range. Commonly, material used in a similar application is heat-treated carbon steel that is ground, hard-chromed and polished to obtain the required hardness and surface finish. Hard chrome deposits have a hardness range of 56-74 HRC, depending on the bath type used (Svernson, 2006).

The figure below shows the Rockwell C hardness of this piston, prior to stress relieving. This component was hardened using oil as quench medium. A maximum surface hardness of 57 Rockwell C hardness (HRC) was obtained from this oil quench (prior to stress relieving). The final hardness of the piston, after stress relieving, is 52-53 HRC.

Figure 3.4 – Image indicating the Rockwell C hardness of the Piston prior to stress relieving

The design of this Piston allows it to accommodate two bearing strips and a Piston Seal. Detail regarding the bearing strips and the Piston Seal can be viewed in Section 3.6. To avoid damage to these components, lead-in chamfers and rounded edges was introduced to the design. As specified by Hallite, the seal manufacturers, the maximum surface finish of the groove that accommodates the Piston Seal, is limited to 3.2 μmRa (Hallite, 2015).

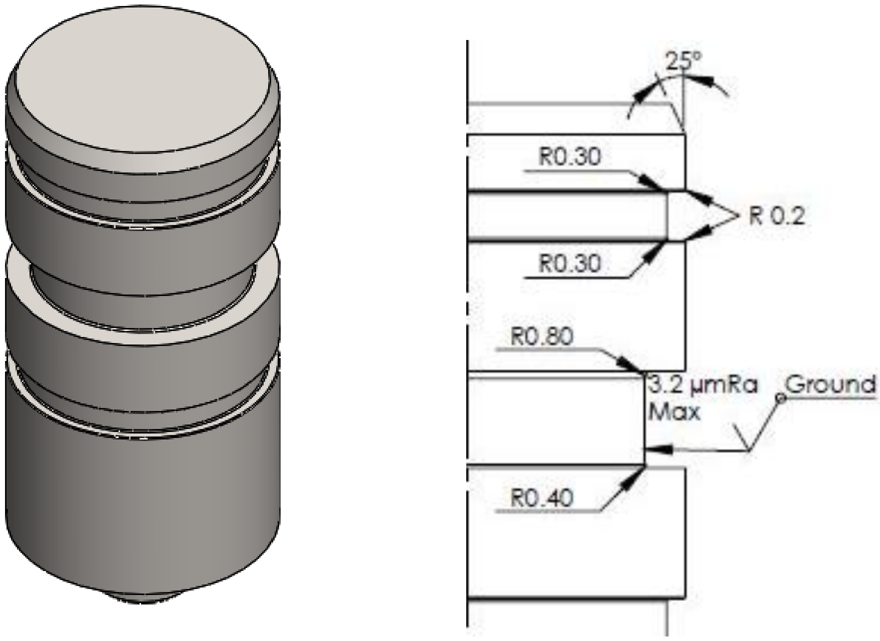


Figure 3.5 – Isometric representation of the designed Piston, with some dimensions

The Piston is attached to the Piston Base Plate by means of threading which, sequentially, is a blank flange mounted to the Systems Sleeve Housing. The clearance between the piston and the piston sleeve is 0.35 mm.

To ensure that this component does not deform as the required internal pressure rating is generated, calculations regarding the axial and radial displacement were done. The maximum radial deformation is limited to the clearance between the Piston and the Piston Sleeve. For calculation purposes, this Piston was considered a solid structure without any grooves.

3.1.1 Axial and Radial Displacements - Calculated

The axial displacement (δ) of a point, at whatever time an axial force F is exerted on a cross section area A of an object with original length L, can be calculated by means of using *Equations 1* and *2*, with E representing Young's Modulus (Hibbeler, 2010). To obtain the strains in both the axial and radial direction, *Equation 3* is used (Hibbeler, 2010):

$$\delta = \frac{FL}{AE} \qquad \textit{Equation 1}$$

$$\sigma = \frac{F}{A} \qquad \textit{Equation 2}$$

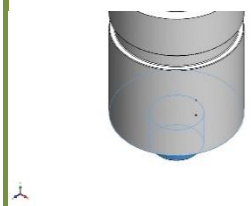
$$\varepsilon_{Axial} = \frac{\delta}{L} \text{ and } \varepsilon_{Rad} = \frac{\delta'}{r} \qquad \textit{Equation 3}$$

By using *Equation 1 and 3*, the maximum axial and outward radial displacement results of this component, at whatever time a compressive load of 10 MPa is subjected to the component, are 4.825e-3 mm and 3.698e-4 mm respectively.

3.1.2 Axial and Radial Displacements - FEA

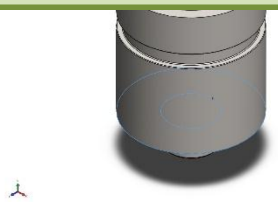
The table below illustrates and describes the fixtures and parameters used in obtaining the simulated axial and the radial displacement, given a 10 MPa pressure load is applied to the Piston. The Piston is fixed to the Piston Base Plate and this restrains displacement in the vertical direction.

Table 3.1 – Table showing and discussing the Piston’s fixtures

Fixture Name	Fixture Image	Fixture Details
Baseplate Fixed		Entities: 1 face (s) Type: Fixed geometry

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force (N)	-0.933237	2465.65	-0.274899	2465.65
Reaction Moment (N.m)	0	0	0	0

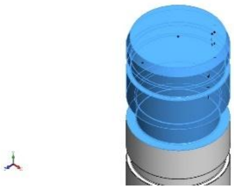
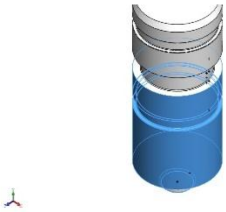
Fixture Name	Fixture Image	Fixture Details
Baseplate Slide		Entities: 1 face (s) Type: Roller/Slider

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force (N)	-3.04728	17658	3.90541	17658
Reaction Moment (N.m)	0	0	0	0

The pressure loads applied to selected surfaces is illustrated and discussed in Table 3.2 below. One section of this component is subjected to the maximum generated pressure rating, whereas the other section is being subjected to atmospheric pressure (due to the seal located in the middle groove).

Table 3.2 – Table showing and discussing the Piston's applied pressure loads

Pressure Name	Section Under Pressure	Pressure Details	
Internal Pressure		Entities:	16 face (s)
		Type:	Normal to selected face
		Value:	1e+007
		Units:	N/m^2
		Phase Angle:	0
		Units:	Deg
Atmospheric Pressure		Entities:	9 face(s)
		Type:	Normal to selected face
		Value:	100000
		Units:	N/m^2
		Phase Angle:	0
		Units:	Deg

Presented in the following figure (Figure 3.6), are the results of the radial and axial displacement of the Piston, using the above tabulated simulation parameters. See Table 3.3 for a results comparison between the calculated and the simulated results.

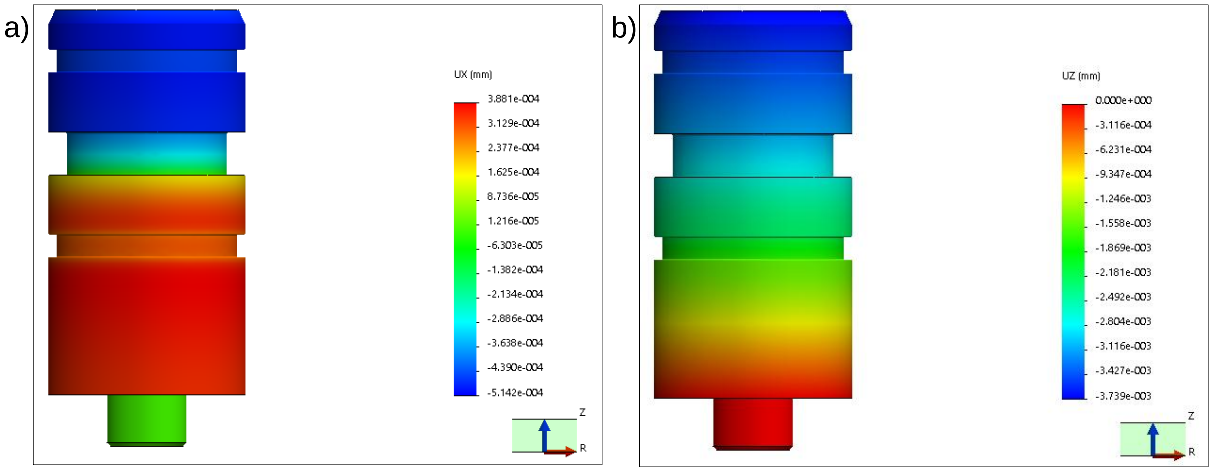


Figure 3.6 – Simulated SolidWorks Results of the Piston's a) Axial and b) Radial Displacement given a 10 MPa hydrostatic pressure load

Presented in the table below is a comparison between the calculated and simulated results. The deviation between these results is due to the assumption that the Piston is a solid bar (without any grooves), contrary to that of the simulated model.

Table 3.3 – Table showing the calculated and simulated axial and radial displacement results, together with the deviation between these results, of the Piston when a hydrostatic pressure of 10 MPa is applied thereon

	Calculated Results	FEA Results	Deviation
Axial Displacement	4.825e-3 mm	3.739e-3 mm	1.086 e-3 mm
Radial Outward Displacement	3.698e-4 mm	3.881e-4 mm	0.183 e-4 mm

The clearance between the Piston and the Piston Sleeve is 0.35 mm. From the results tabulated above, it is clear that there are no complications when the described pressure load is applied to this component.

3.2 Piston Sleeve

For safety purposes, the Piston Sleeve was designed to withstand an internal pressure rating of 15 MPa, contrary to that of other designed components which were designed to withstand an internal pressure of 10 MPa. As previously mentioned, the testing medium is water. Using CES Edupack Material Selection software, the material best suitable for this application is Stainless steel 304 L. A low grade carbon stainless steel is less likely to sensitise during welding.

To ensure that the designed testing facility is capable of testing a variety of valve sizes, the Piston Sleeve is threaded on the outer diameter at one end (to accommodate variable sized machined flanges - see Section 3.3) and with grooves machined on the other outer end. These grooves accommodate the Stopping Ring as will be described in Section 3.4.

According to the ASME Boiler and Pressure Code, paragraph PG-39 (ASME, 2015), there is a series of different methods of attachment of pipe and nozzle necks to vessels. These connection methods are as follows: welded -, studded -, threaded - and expanded connections. Of these methods, the best suited type, given the application, is threaded connections.

When it comes to threaded connections, there are a minimum number of threads that need to be engaged. Depending on a pressure vessel wall thickness, a built-up pad or fitting may be used to provide the adequate material thickness (ASME, 2015). The minimum number of threads required per connection for a pressure greater than 2 MPa, for a pipe connection (DN) of 15 to 20 the minimum plate thickness, are 11 mm with the number of threads to be engaged, being 6 (ASME, 2015).

The Piston Sleeve is a 50NB schedule 80 stainless steel 304L seamless pipe with an outer diameter of 60.33 and a wall thickness of 5.54 mm that was machined to accommodate the design of this component, *see* Appendix A for detailed drawings. To ensure this component complies with the above mentioned ASME specifications, built-up pads were added to the outer wall of this component. Below is an isometric view of the designed Piston Sleeve, together with some dimensions.

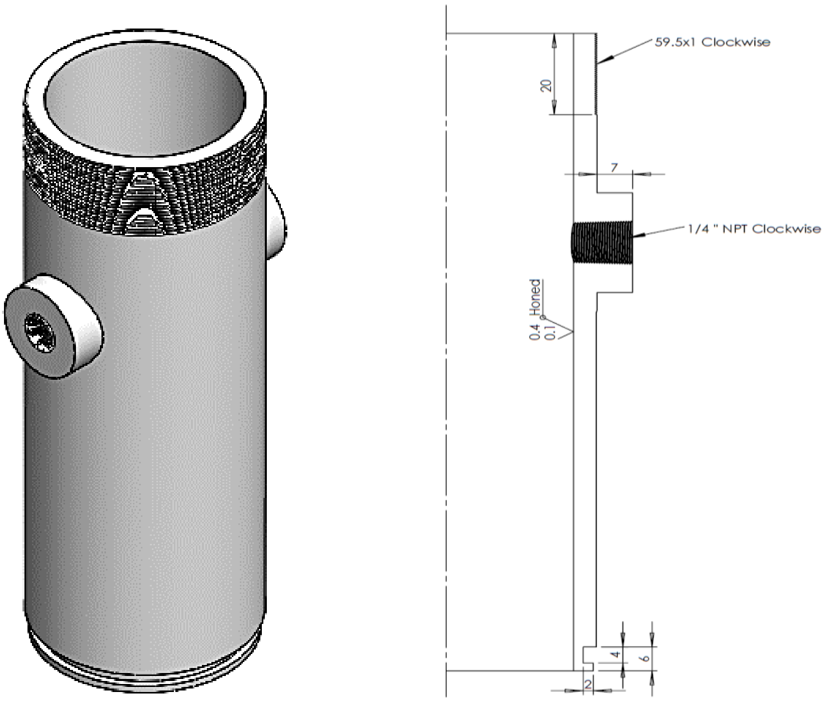


Figure 3.7 –Isometric representation of the designed piston sleeve, with some dimensions

The internal surface finish of this Piston Sleeve, as specified by Hallite, ranges between 0.1-0.4 μmRa (Hallite, 2015). The outer surface finish of this component did not receive any special attention.

To ensure that the Piston Sleeve (under internal hydrostatic pressure) does not cause problems with the rest of the system, a series of calculations were done. These calculations include minimum wall thickness -, internal stresses -, and radial displacement calculations. The parameters of the internal and external pressure loads are 15 MPa and 100 KPa (atmospheric pressure), respectively.

3.2.1 Minimum Wall Thickness - Calculated

Stresses caused by internal pressure is considered safe when the wall thickness of the component conforms to the minimum required wall thickness, including mechanical, corrosion, and erosion allowances (ASME, 2012). According to ASME, the minimum wall thickness (t_m) can be calculated by means of adding together the wall thickness (t) with the mechanical allowance (c) of the component (*Equation 4)*.

$$t_m = t + c$$ *Equation 4*

$$t = \frac{PD}{2(SEW + PY)} \quad \text{or} \quad t = \frac{P(d+2c)}{2[SEW - P(1-Y)]}$$ *Equation 5*

The material stress value (S) can be obtained in Table A-1 of the ASME B31.3 code. The material quality factor (E) and weld joint strength reduction factor (W), can also be obtained in the ASME B31.3 code. According to this code, the value of coefficient Y can be obtained from Table 304.1.1 if the wall thickness is less than the outer diameter (D)/6. If the wall thickness is greater or equal to that of D/6, then the following equation should be used in calculating the Y coefficient:

$$Y = \frac{d + 2c}{D + d + 2c} \qquad \textit{Equation 6}$$

With (d) being the internal diameter of the pressure vessel. The magnitude of the designed pressure rating (P) within this component is 15 MPa. The safety relieve valve in the system ensures that the generated pressure within this component doesn't reach critical levels.

By means of using the equations above, the minimal wall thickness of this component was calculated as 4.135 mm. This component is considered to a be a thick wall pressure vessel (Hibbeler, 2010). The wall thickness used is 4.6 mm.

3.2.2 Directional Stresses - Calculated

To ensure that this component does not yield when subjected to the required internal hydrostatic pressure rating, calculations regarding the stresses, in each direction, were done. The following equations (*Equation 7-9)* were used in obtaining the stress in the Axial, Circumferential and Radial directions.

$$\sigma_a = \frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2} \qquad \textit{Equation 7}$$

$$\sigma_c = \frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2} - \frac{r_i^2 r_o^2 (p_o - p_i)}{r(r_o^2 - r_i^2)} \qquad \textit{Equation 8}$$

$$\sigma_r = \frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2} + \frac{r_i^2 r_o^2 (p_o - p_i)}{r(r_o^2 - r_i^2)} \qquad \textit{Equation 9}$$

The calculated Axial, Circumferential and Radial stresses are 36.98 MPa, 74.06 MPa, and 100 KPa, respectively. The yield strength of the selected material (Stainless steel 304 L) is 241 MPa (AK Steel, 2007).

3.2.3 Radial Displacement - Calculated

As in previous sections, the clearance between opposing components plays a major role. The radial displacement at any point on a cylinder is given by the following equation:

$$u = A_1 r + \frac{A_2}{r} \qquad \textit{Equation 10}$$

The constant values of A_1 and A_2 can be determined by boundary values on the cylinder and was calculated using the equations below (*Equations 11-12)*, with Young's Modulus (E) and poison ratio (v) being material properties.

$$A_1 = \frac{(1-\nu)}{E}\left[\frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2}\right] \qquad \textit{Equation 11}$$

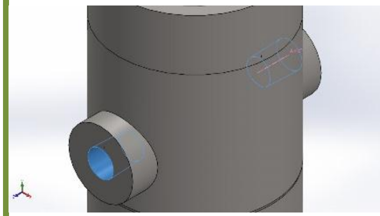
$$A_2 = \frac{(1+\nu)}{E}\frac{(p_i - p_o) r_i^2 r_o^2}{r_o^2 - r_i^2} \qquad \textit{Equation 12}$$

The inner (r_i) and outer radius (r_o) of the cylinder were calculated with reference to the wall thickness of the cylinder. The internal and external radiuses being 25 mm and 29.6 mm, respectively. By using *Equation 10-12*, the outward radial displacement of the system subjected to the described pressure loads was calculated as 0.01154 mm.

3.2.4 Radial Displacements - FEA

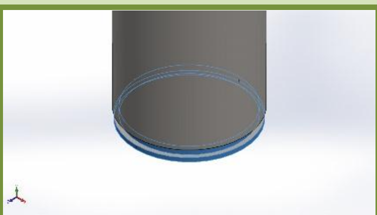
The tables below illustrate the fixtures and parameters used in the simulation of this component when the specified pressure load is applied. The results of this simulation are compared to that of the calculated results in Table 3.6.

Table 3.4 - Table showing and discussing the Piston Sleeve's fixtures

Fixture Name	Fixture Image	Fixture Details
1/4" NPT fitting		Entities: 2 face (s) Reference: Axis2 Type: Use reference geometry Translation: ---, 0 rad., 0 Units: mm

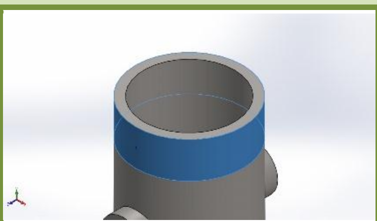
Resultant Forces

Components	X	Y	Z	Resultant
Reaction force (N)	0.113792	-2163.32	0.59634	2163.32
Reaction Moment (N.m)	0	0	0	0

Fixture Name	Fixture Image	Fixture Details
Guide Ring		Entities: 2 face (s) Type: On Cylindrical Faces Translation: 0, ---, --- Units: mm

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force (N)	-0.066622	0	-0.410699	0.416068
Reaction Moment (N.m)	0	0	0	0

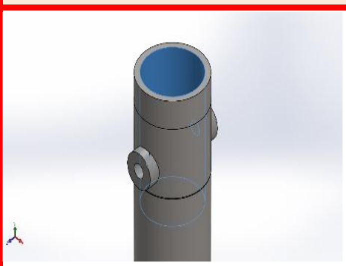
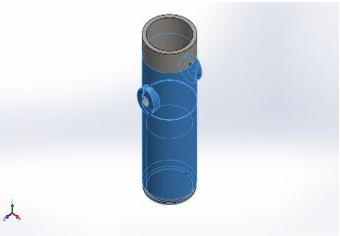
Fixture Name	Fixture Image	Fixture Details
Flange		Entities: 1 face (s) Type: On Cylindrical Faces Translation: 0, ---, 0 Units: mm

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force (N)	-1.78839	2163.57	0.55439	2163.57
Reaction Moment (N.m)	0	0	0	0

The loads applied to this system are both the internal and atmospheric pressures. As stated previously, the system must be able to generate and withstand an internal pressure rating of 15 MPa.

Table 3.5 - Table showing and discussing the Piston Sleeve's applied pressure loads

Pressure name	Section under pressure	Pressure details
Internal Pressure		Entities: 1 face (s) Type: Normal to selected face Value: 1.5e+007 Units: N/m^2 Phase Angle: 0 Units: Deg
Atmospheric Pressure		Entities: 11 face(s) Type: Normal to selected face Value: 100000 Units: N/m^2 Phase Angle: 0 Units deg

The maximum strain will occur on the outer diameter of the Piston Sleeve. The figures below show the simulated radial displacement in both the X and Z planes, respectively.

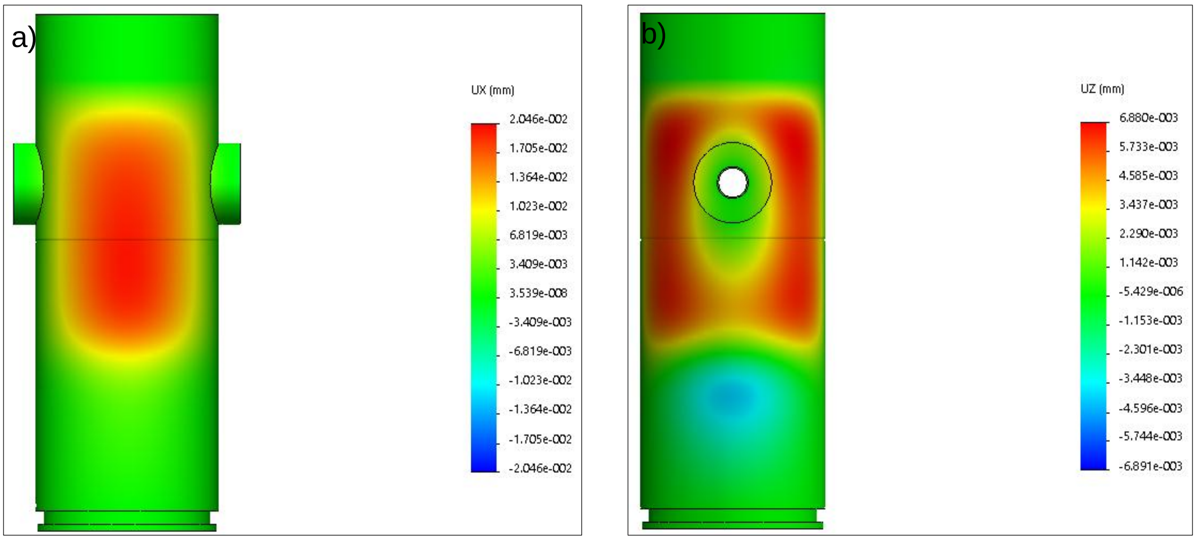


Figure 3.8 - Simulated Results of the Piston Sleeve's Radial displacements in different planes

From the figure above, the maximum radial displacement in the X and Z planes are 2.046e-2 mm and 6.880e-3 mm, respectively. Tabulated below are a comparison between the calculated and simulated radial displacement results of the Piston Sleeve, when subjected to the described loading conditions.

Table 3.6 - Comparison between the calculated and simulated radial displacement of the Piston Sleeve, when subjected to an internal pressure of 15 MPa.

	Maximum Outward Radial Displacement
Calculated Results	11.54 e-3 mm
FEA Results	2.046 e-2 mm
Deviation	8.92 e-3 mm

From these results, it is clear that no complications will occur when this Piston Sleeve is subjected to the described loading conditions. The maximum radial displacement is less than the clearance between this component and the opposing component (Guide Bush). The clearance between these components is 0.04 mm.

3.3 Flanges

In the designed subassemblies, there are a series of different machined flanges, each specifically designed for a specific purpose. Below is an isometric view of these flanges.

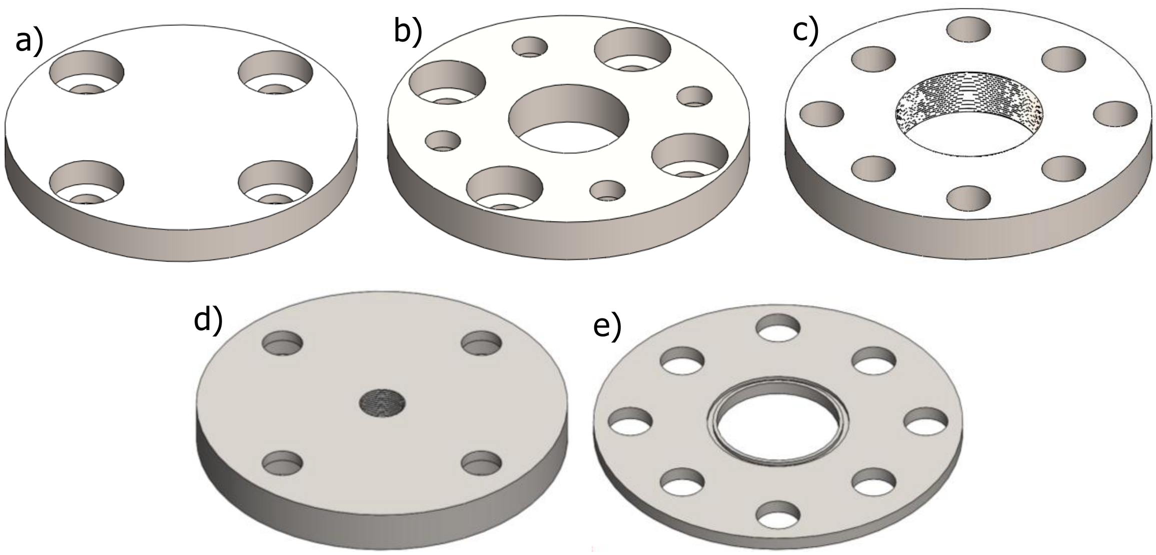


Figure 3.9 – 3D models of the designed flanges

The upper and lower faces of the designed mountings (water reservoir interface and pressure intensifier) are perpendicular to the working direction. To ensure level faces, pockets were machined to sink the heads of the bolts used in the system assembly.

The flange illustrated in Figure 3.9(b) is specially designed to accommodate a bidirectional fixture, attaching the testing specimen to the water reservoir interface. A hex pocket at the bottom of the flange was machined to tighten the test specimen to this flange. To ensure that the Piston used in the system stays fixed and parallel to the working direction, the thread machined in flange 3.9(d), accommodates those machined on the Pistons.

Due to the threaded connection between the connection flange and the Piston Sleeve, leakage through the thread might occur under high hydrostatic pressures. This being the motivation for the design of flange E. It is machined in this manner to accommodate an O-ring with an outer diameter not greater than the connection flange's inner diameter.

3.4 Stopping Ring

In the case of a sliding component, it is beneficial to support the sliding component at multiple positions and, therefore, the motivation for the Stopping Ring. This component was designed to accommodate the grooves machined in the outer bottom end of the Piston Sleeve. Presented in the figure below, is a representation of the assembling of the Stopping Rings of the Piston Sleeve.

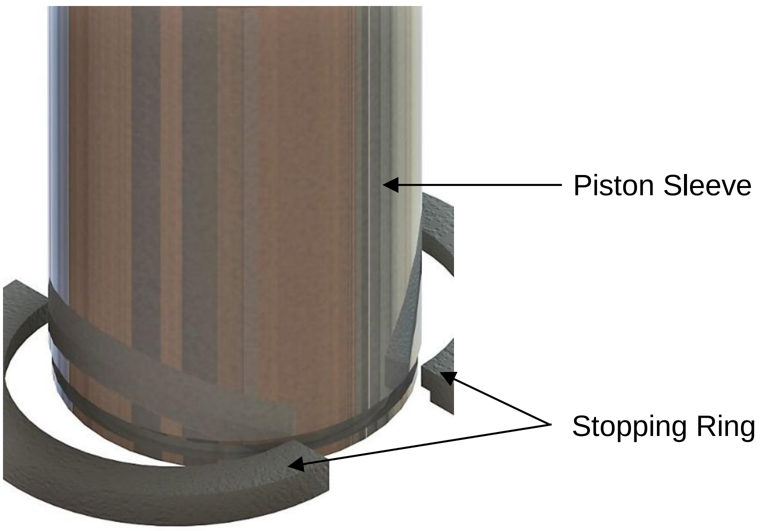


Figure 3.10 - Partial assembly view of the Stopping Ring and Piston Sleeve

For assembly purposes, the Stopping Ring was machined into two half sections. The gap between these sections allows lubrication to pass from one side to the other, as the system as a whole slides up and down. The material used for both the Stopping Ring and the Guide

Bush is Vesconite. The reason for the use of this material is to prevent galling. The clearance between the outer face of the Stopping Ring and the sleeve housing the inner face is 0.04 mm.

3.5 Guide Bush

As stated in the previous section, it is beneficial for a sliding component to be supported at multiple positions. The Guide Bush serves as one of these supports. To ensure that the system stays free from external particles, the Guide Bush was designed in such a manner as to accommodate a wiper seal. Below is a figure representing the assembling of this Guide Bush and surrounding components.

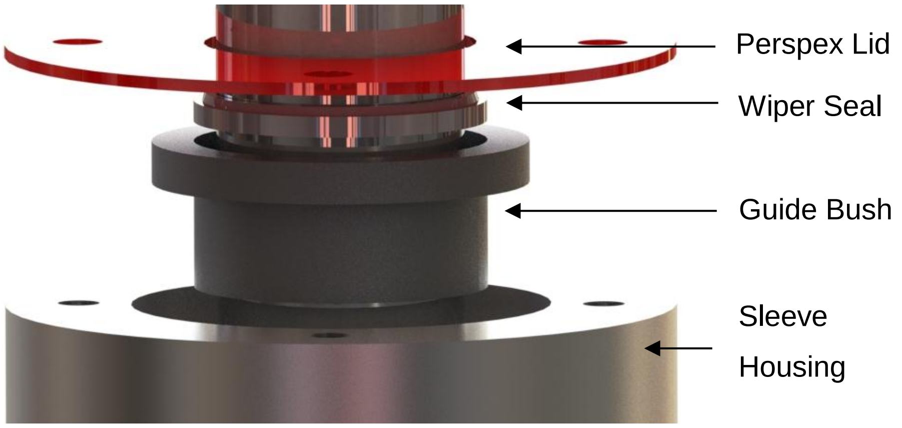


Figure 3.11 – Partial assembly view of the Guide Bush and surrounding components

As previously mentioned in Section 3.4, the material of the Guide Bush is Vesconite. As a safety precaution, the components manufactured from this material will fail first whenever the system misaligns. As mentioned in Section 3.2, the clearance between the Guide Bush and the sliding Piston Sleeve is 0.04 mm.

3.6 Seals, Bearings, and Gaskets

The seal used in the system is a Hallite 601 high-performance general purpose piston seal. According to Hallite specifications, the maximum permissible extrusion gap for this seal, with a pressure rating of 160 -, 250 - and 400 bar, are 0.6 -, 0.5 - and 0.4 mm, respectively. To ensure the maximum lifetime of a seal, the surface opposing the seal needs to be as smooth as possible. Thus, the surface finish of the Piston Sleeve's inner surface, as specified by Hallite, is in the range of 0.1-0.4 µmRa.

In hydraulic systems, the most commonly used guides are guide rings (SKF, 2017), commonly known as bearings. These bearings not only guide the Piston in the sleeve but also accommodate radial loads acting on the system (SKF, 2017). In addition to this, these guides prevent metal-to-metal contact between opposing sliding materials, which in turn reduce the potential of galling. The type of wear strip used in this design is a PTFE Diamond ruster wear strip.

Figure 3.12 – Figure showing a 3D model and a picture of a) a high-performance general purpose piston seal and b) Klinger Graphite PSM/AS gasket respectively

Lodged between the pressure intensifier, test valve and water reservoir interface, are gaskets. These gaskets ensure a leak tight system. The size of these gaskets depends on the size of the test valve. The type of gasket used in this system is the Klinger Graphite PSM/AS gasket. This graphite based gasket material has a high compressibility that causes good sealing properties. These gasket materials also have an anti-stick finish that contributes to the ease of system maintenance and reuse thereof. The operating temperature of these gaskets ranges from -200°C to 460°C (Klinger, 2010).

3.7 Safety Relieve Valve

To prevent the system from over pressurisation, a Safety Relieve Valve (SRV) was installed. The type of relieve valve used in this system is an R-series Swagelok relieve valve. This series of relieve valves open as the pressure within the system reaches the set pressure and closes as the system's internal pressure falls below this set pressure. Below is a figure representing a 3D model of this component.

Figure 3.13 – 3D model of a Safety Relieve Valve (SRV)

The parameters of the set pressure are determined by the type of spring installed in the SRV. The type of spring installed in the SRV used in this designed system has a purple spring, which means that the pressure ranges thereof is 51.7 to 103 Bar.

3.8 Swagelok ball valves

In this study, two Swagelok ball valves were used. Each placed on opposite sides of the water reservoir interface. One at the water inlet, and the other at the water outlet/venting point. These valves are manufactured from stainless steel 316. Below is a representation of one of these valves.

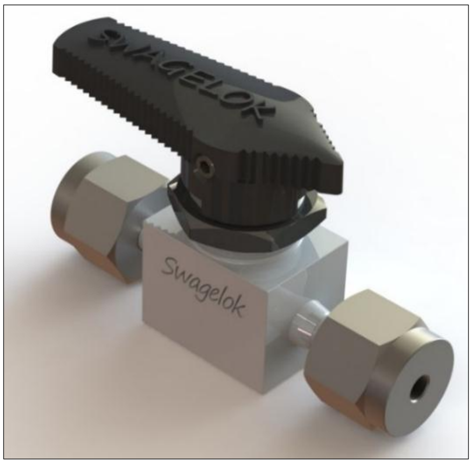


Figure 3.14 – 3D model of the ball valves used in the system

Depending on the type of test to be performed on the test valve, these Swagelok ball valves will either be opened or closed. The maximum pressure rating of these ball valves at a temperature range of 37 °C, is 206 Bar.

3.9 Data Acquisition Modules

The relevant data was obtained using Advantech Data Acquisition Modules. These modules were integrated with software programmed in Visual Studio. The type of Modules used in this study, are:

- ADAM – 4017 8-Channel Analog Input Module;
- ADAM – 4018 8-Channel Thermocouple Input Module;
- ADAM – 4060 4-Channel Relay Output Module;
- ADAM – 4520 RS-232 to RS-422/RS-485 Converter.

The above mentioned modules accept any power unit that supplies power within the range of +10 to +30 VDC (Advantech, 1994). The operating voltage of the DB Board is 24 VDC.

3.10 Pressure Transmitter and Thermocouple

The pressure transmitter used in the designed system is a VegaBar14. This component delivers a current of 4 to 20 mA according to the pressure rating within the system. This specific pressure transmitter has a pressure range of 0 to 40 bar. Thus, for safety purposes, the maximum generated internal pressure within the system is limited to this pressure transmitter's range. The operating power of this component ranges between 12 to 30V DC. Below is a 3D isometric model of a pressure transmitter.

Figure 3.15 – 3D model of a VEGABAR 14 Pressure Transmitter

The correlation between the mA signal (x) sent to the ADAM 4017 Data Acquisition Module and the measured pressure (y), are linear. *Equitation 13 and 14* were used in obtaining the value of the generated internal water pressure rating, in MPa.

$$y = mx + c \qquad \textit{Equation 13}$$

$$y = (2.5x - 10)/10 \qquad \textit{Equation 14}$$

The temperature of the testing medium was measured using a type T thermocouple, integrated with an ADAM 4018 Data Acquisition Module. The values of the measured pressure and temperature ratings are displayed on a Graphical User Interface (GUI), which is presented and discussed in the following chapter.

4 SOFTWARE

As previously elaborated on in Section 3.9, the specific ADAM modules were integrated with the system by means of software specifically developed to obtain the required data from these modules. The software was developed with Visual Basic and the figure below demonstrates the Graphical User Interface (GUI).

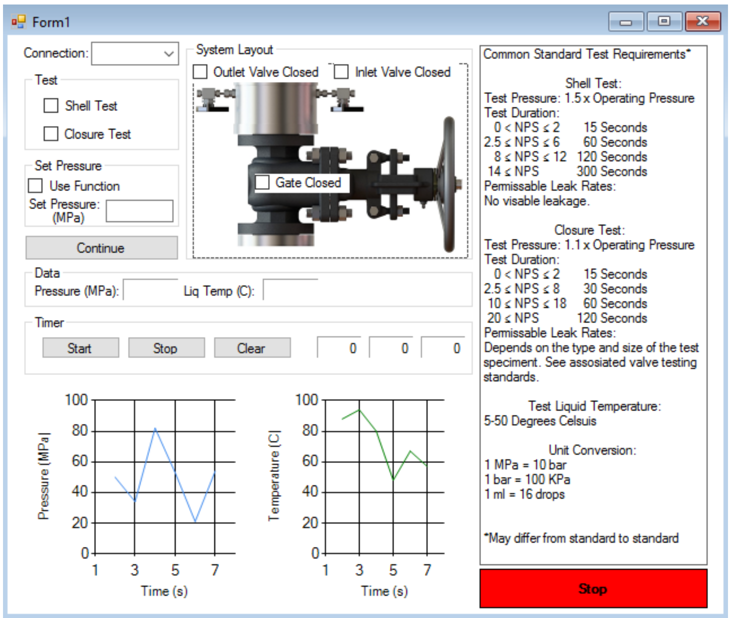


Figure 4.1 - Graphical User Interface (GUI) of the software

The software was developed to accommodate the valve testing procedures discussed in Section 2.5 of this dissertation. Below is a functional block diagram of the developed software.

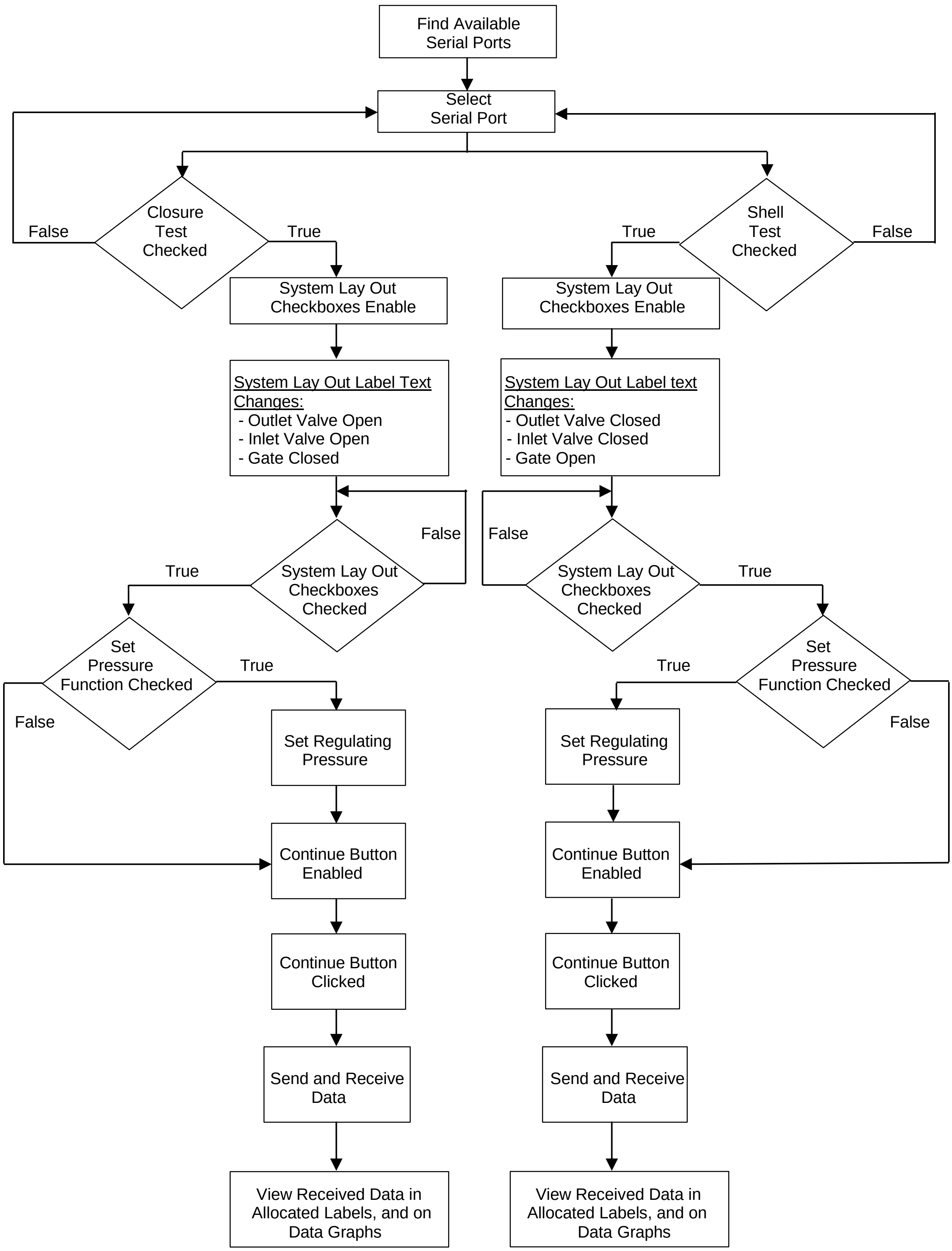


Figure 4.2 - Functional Block Diagram of the developed software

The operating power of the DB board is 24 V DC. The ADAM Data Acquisition Modules discussed in the previous chapter, operates at a voltage range between 10-30 V DC (Advantech, 1994). Based upon research, it was found that a cheaper alternative in obtaining this required voltage range would be to use a 12 V power supply, connected to a step-down converter (buck converter) which evidently supplies 24V DC. The layout of the DB board can be viewed in Appendix D.

5 LEAK DETECTION AND VERIFICATION OF DESIGN

This chapter is divided into two sections, the first of which focuses on the leak tightness of the designed subassemblies, whereas the second section focuses on complete standard compliant hydrostatic shell- and closure tests of a Class 150, 2-inch floating ball valve (using these designed subassemblies to facilitate in the pressure generating procedures). The results of these test runs are presented and discussed. The raw data of these tests can be obtained on the attached CD.

The axial force applied to the system to generate the required hydrostatic pressure rating, was applied by means of a Tinius Olsen press. The configuration of this press is of such a nature that the anvils thereof applies loads in the vertical direction. Figure 5.1 shows an image of this press.

Figure 5.1 - Figure illustrating the press that was used in the system

The designed subassemblies, together with a valve being tested are placed between the anvils of the press. The figure below shows a schematic representation of the system assembly.

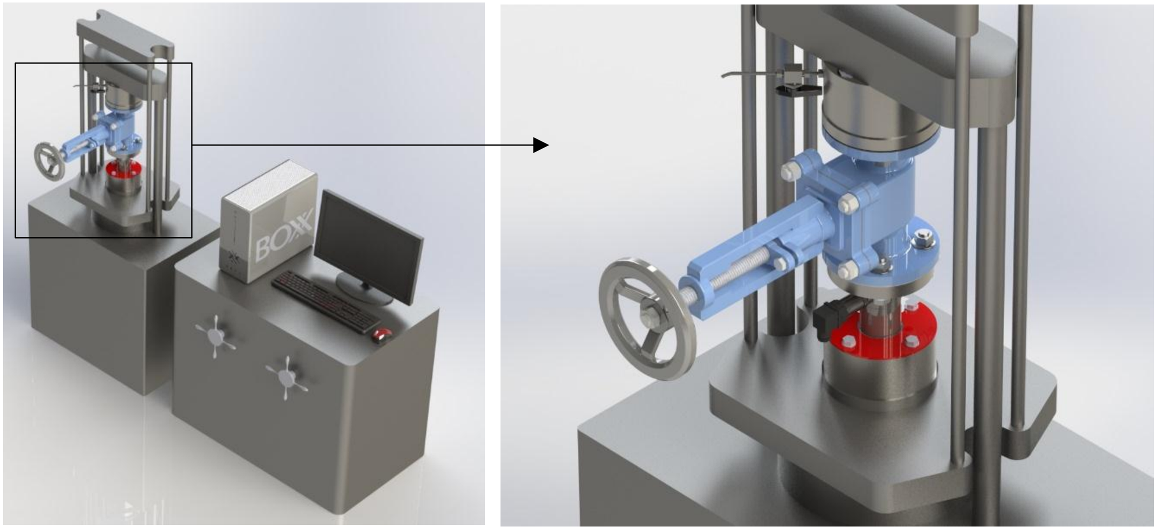


Figure 5.2 - Figure representing the designed subassemblies, together with a test valve lodged between these subassemblies

Once the assembly (designed subassemblies together with a test valve) is placed into position, with the body of the valve filled with water, a test run can commence. A test run consists of an internal water pressure generation and a pressure regulation (of the generated pressure) cycle. The temperature of the water is measured as it enters the above mentioned assembly. The duration of the pressure regulation depends on the size and type of valve to be tested.

5.1 Designed System Leak Tightness

To reduce any potential leaks located in the system (press and the designed system), a test valve was first excluded for a preliminary test run. As seen in Figure 5.3, the water reservoir interface is connected to the pressure intensifier. Following this figure, is a brief description of the test run procedure.

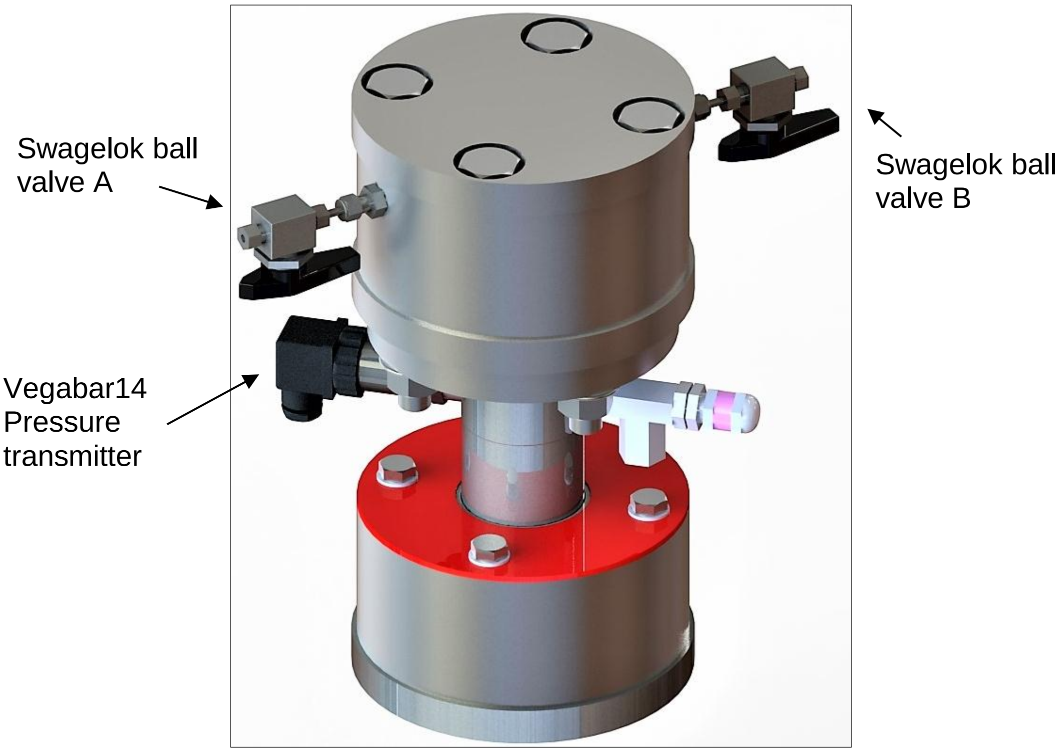


Figure 5.3 – 3D model of the assembly used in the preliminary test run

With this assembly (represented in Figure 5.3) in place, water enters through Swagelok ball valve A, whereas air vents through Swagelok ball valve B (as the assembly fills with water). Once all the air is vented from this assembly, and the Swagelok ball valves are closed, an axial load, caused by the described press, is applied. The hydrostatic pressure within the assembly is measured using a Vegabar14 pressure transmitter (see Section 3.10).

The test runs conducted on the above assembly were done with an internal hydrostatic pressure rating of 3.7 MPa, due to the fact that the available (used) pressure transmitter has a pressure range of 0-4 MPa. The reason for the selected pressure rating of 3.7 MPa, and not 4 MPa, is due to monitor pressure overshoot.

5.1.1 Preliminary test run

Following the procedure described above, a preliminary test run was conducted. The internal hydrostatic pressure was raised to the previously discussed 3.7 MPa. An increase between the water inlet temperature (21.4°C) and outlet temperature, before and after this hydrostatic test, was about 0.5°C. The following figure depicts the results of the internal generated pressure of this test run.

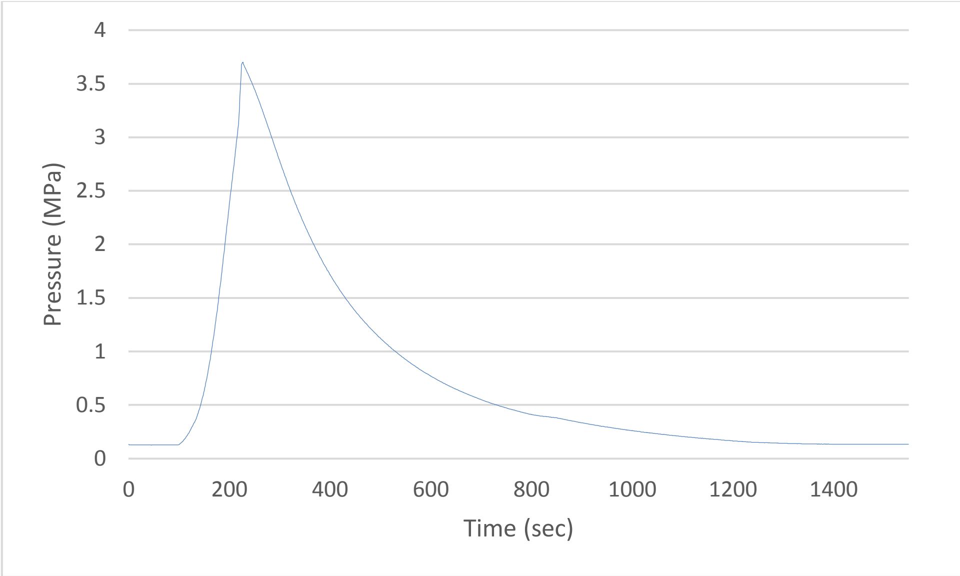


Figure 5.4 – Results of the 3.7 MPa internal water pressure rating preliminary test run

From these results (depicted in the figure above), it is clear that the system loses its generated hydrostatic pressure. This can be due to the following reasons:

1. Leakage of the designed subassemblies;

2. Leakage of the hydraulic piping within the press.

To determine which of the above mentioned reasons are responsible for the mentioned internal hydrostatic pressure drop, a dead weight test was conducted on the designed system. These weights were placed on the described assembly to mimic the internal pressure that should have been generated by the press, thus eliminating the potentially faulty press (see Figure 5.5). The available weights are capable of generating an internal hydrostatic pressure rating of about 0.5 MPa. In addition to the monitoring of the internal hydrostatic pressure, the compression caused by these weights (on the assembly), was monitored, given a constant load. The axial displacement was measured using a clock gauge that was placed beneath the connection flange.

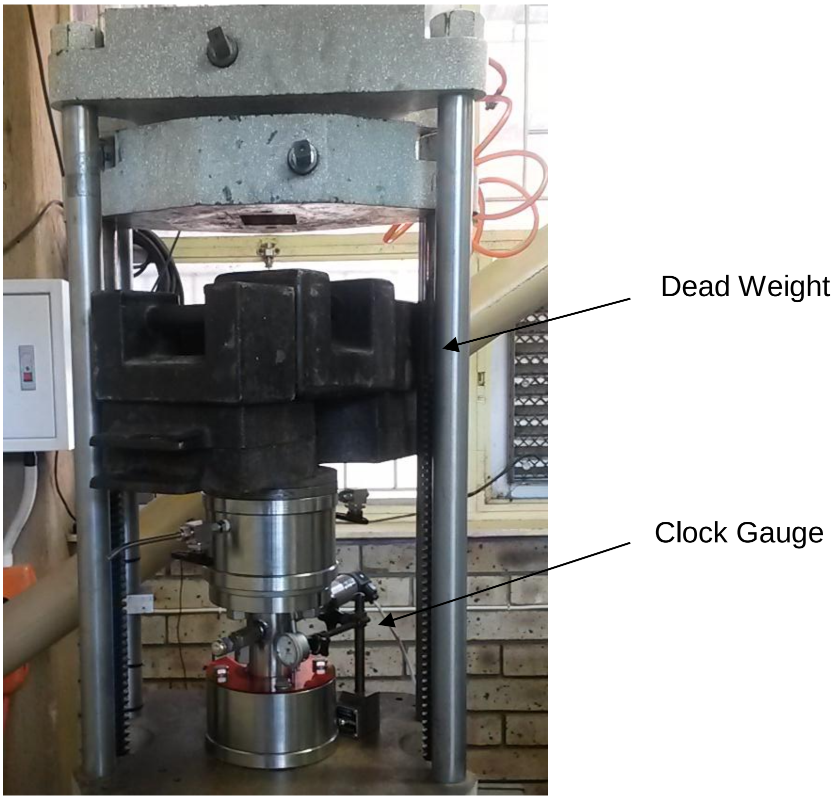


Figure 5.5 – Figure showing weights placed upon the assembly (with test valve excluded)

If the generated internal water pressure stays constant, with a zero-compression displacement on the clock gauge, then it can be concluded that the press used in the system, is faulty. Measuring the hydrostatic pressure within this assembly, the pressure ratings below were obtained.

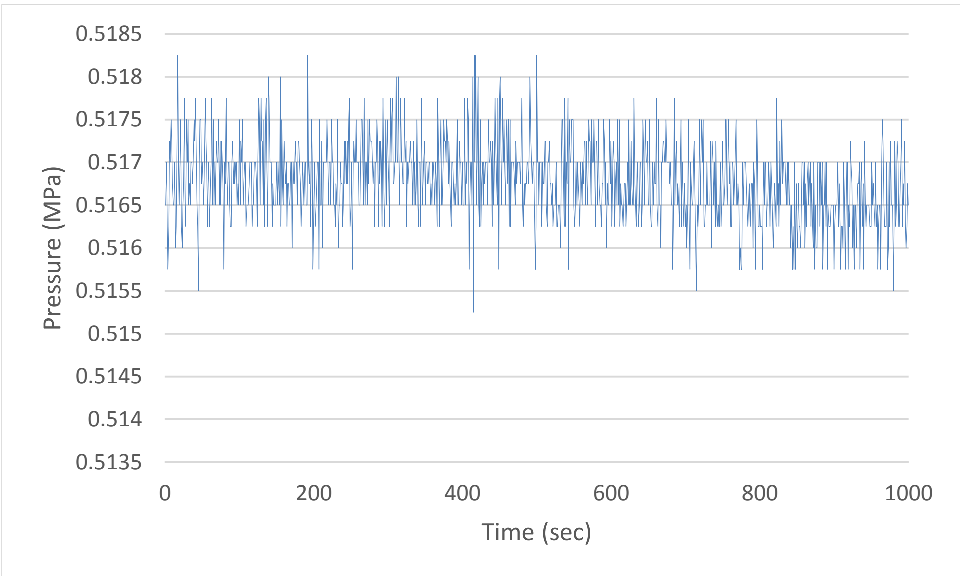


Figure 5.6 – Line graph illustrating the results of the internal water pressure with a dead weight placed upon the assembly, see figure 5.5

In the results presented above, it is clear that hydrostatic pressure generated by the weights placed on the assembly, stayed constant. The figures below show the total axial compression displacement of this assembly, under constant loading conditions, for a time period of 1000 seconds.

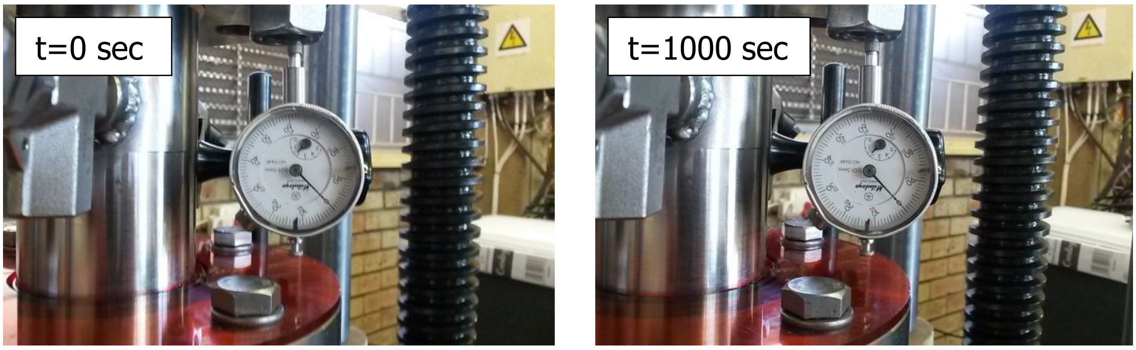


Figure 5.7 – Figures showing the compression displacement, at certain time interval, of the system under a constant load of about 1100 N.

From the results presented in Figure 5.6 and Figure 5.7, it can be concluded that the designed system did not produce any leakage. This proves that the press used in the system, is faulty. This problem was eliminated by means of using a relay which was programmed to start and stop the press compression at a specified internal pressure rating in order to compensate for the pressure drop.

5.1.2 Automated Relay Test Run

By using this faulty press, as well as the above described relay, an internal hydrostatic pressure rating of 3.7 MPa was generated and regulated. The results of this test run are depicted in the figure below.

.

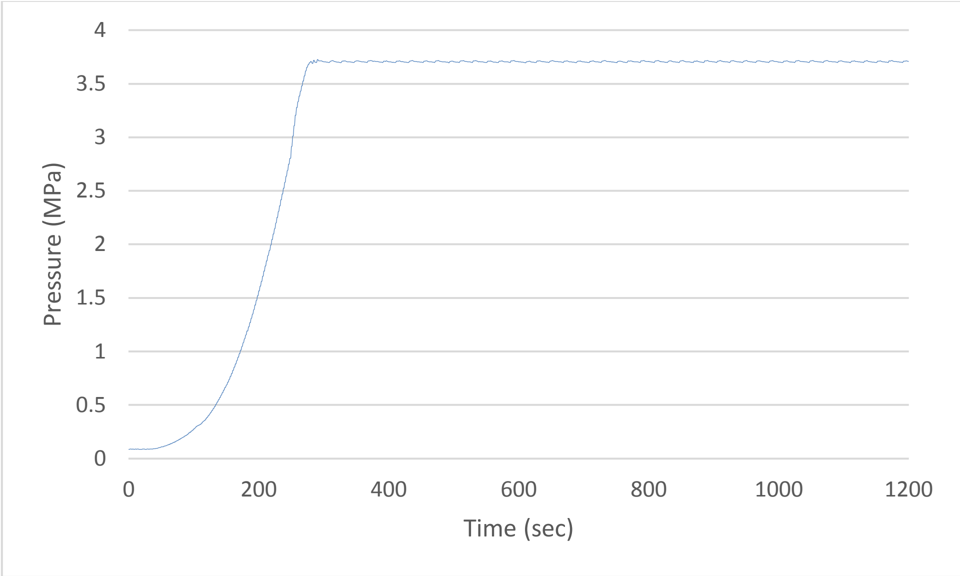


Figure 5.8 – Results of the regulated internal hydrostatic pressure test

From these results, it is clear that the faulty press is capable of generating the set internal hydrostatic pressure rating as well as the relay capable of regulating this generated hydrostatic pressure. Thus, the system is able to perform standardised hydrostatic pressure tests on isolation valves.

5.2 Standard Compliant Hydrostatic Valve Test

In this section, results of a standard complying hydrostatic shell, and closure tests of the Class 150, 2-inch floating ball valve is presented and discussed. This valve was tested according to the following standards:

- API 598
- ASME B16.34
- ISO 5208
- MSS SP 61

Hydrostatic Shell Test

According to ASME B16.34, ISO 5208 and MSS SP 61 the generated hydrostatic pressure rating of a shell test should be conducted with a gauge pressure of no less than 1.5 times the operating pressure rating of the valve being tested. As per API 598, this pressure rating should be no less than 26 bar gauge, given the specifications of this valve. For this specific valve, the minimum duration of this test (as specified by the above standards) is 15 seconds. The timer starts once the required pressure rating is reached. The figure below shows the results of this hydrostatic shell test.

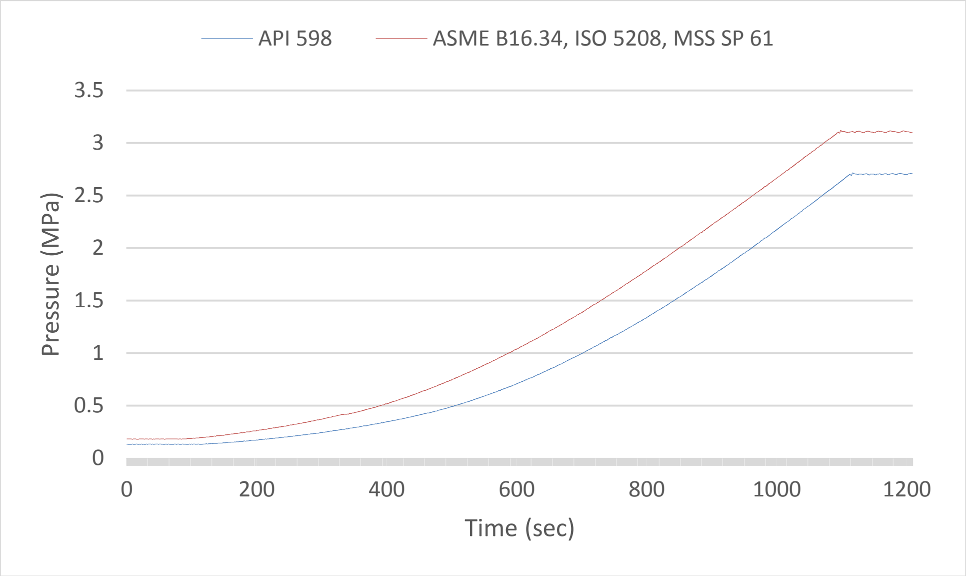


Figure 5.9 – Absolute hydrostatic pressure test results of a standard compliant Hydrostatic Shell test conducted on the Class 150 floating ball valve

The graphs presented in Figure 5.9 depict the hydrostatic pressure within the valve. The red line represents the generated and regulated hydrostatic pressure rating in the valve, according to standards ASME B16.34, ISO 5208 and MSS SP 61. The blue line represents the generated and the regulated hydrostatic pressure rating within the valve as per API 598. These test runs exceeded the minimum testing duration.

As stated earlier in Section 2.4, leakage through the pressure boundary is not permissible. This valve's body was capable of withstanding the required pressure rating, without showing any signs of leakage.

Hydrostatic Closure Test

The generated hydrostatic pressure rating of a closure test (given the previously mentioned standards) was done with a gauge pressure no less than 1.1 times the operating pressure rating. The minimum duration of this type of test, as specified by these standards, for the specific valve at hand, is 15 seconds. The figure below shows the results of this hydrostatic closure test (for one end of the valve).

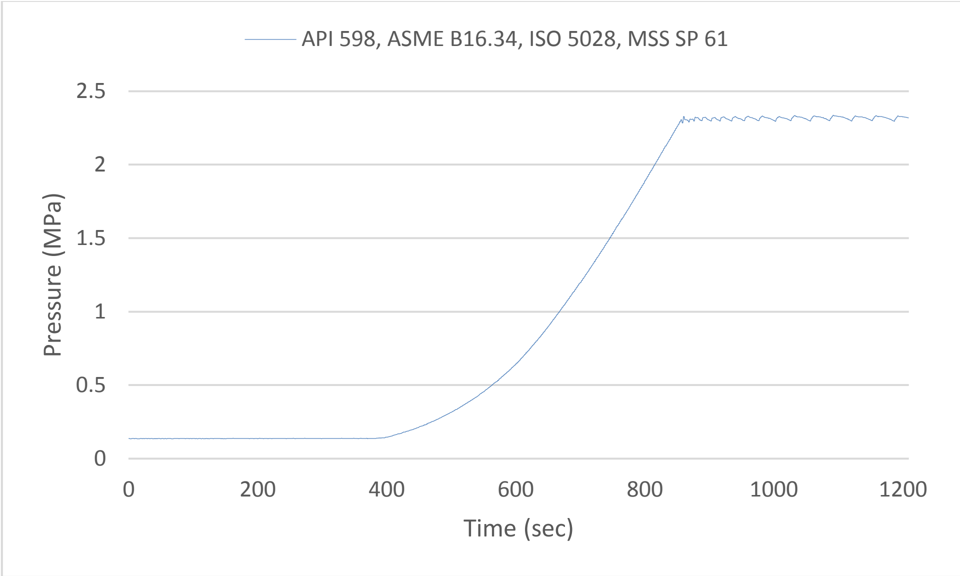


Figure 5.10 – Absolute hydrostatic pressure test results of a standard compliant hydrostatic Closure test on the Class 150 floating ball valve

The graph presented in Figure 5.10 depict the hydrostatic pressure within the valve. The blue line represents the generated and regulated internal hydrostatic pressure ratings according to standards API 598, ASME B16.34, ISO 5208 and MSS SP 61, which is 22 bar gauge. There were no drops formed at the assembly's venting point (*see* Figure 3.3, component 25), thus concluding that this end of the valve subjected to this test, is capable of withstanding the required internal hydrostatic pressure rating.

6 SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

This chapter includes a summary, conclusions and recommendations of the designed subassemblies and the system as a whole. Moreover, recommendations regarding some of the auxiliaries used in the system, are discussed.

6.1 Summary and Conclusions

In Chapter 1 (Section 1.2), a problem statement is presented. From this statement, an aim was identified. This aim is divided into two main divisions, the first of which was to review international valve testing procedures and to find best practise configurations based upon procedures and limits. The second part of this aim includes the design and assembly of a first approach laboratory set-up valve testing facility, capable of utilising the above international valve testing procedures.

The literature study in Chapter 2 concluded that the best test bench configuration is a vertical configuration, meaning the assembly as a whole (designed subassemblies together with a valve to be tested) is placed in a press with the anvils thereof capable of displacing in the vertical direction. The main reasons for this type of configuration is due to practicality and safety precautions. This designed and manufactured testing facility uses water as a testing medium. Furthermore, the literature described a series of different tests to be conducted on a valve. From these test descriptions, a series of apparatus were identified. These apparatuses were then brought into consideration when designing the valve testing facility.

The designed facility consists of two subassemblies namely, the pressure intensifier and the water reservoir interface. The components within these subassemblies were designed and manufactured in accordance with original equipment manufacturers' (OEM) standards and specifications. These subassemblies are designed to be attached on opposite ends of a parallel faced flange isolation valve. Which is placed distinctly in the vertical press.

In a preliminary test run, the subassemblies were fixed to one another and placed in the described press. The results of this test run indicated that a major leak within the system was present. To identify the leak location, an additional test run was conducted. In this test run the top anvil of the press was detached from the assembly and replaced with dead weights.

This dead weight test run's results showed that there was no leakage in and from the designed subassemblies. Thus, the hydraulic pipes or pipe connections of the press are faulty (the press is able to generate the required pressure rating, but is unable to continuously apply loading to the assembly). This problem was resolved by means of using a relay that was programmed to automatically start or stop a compression cycle, once a set pressure is reached.

With the above mentioned alterations, standardised hydrostatic shell and closure tests were performed on a 2-Inch parallel faced flange isolation Class 150 floating ball valve. From these test runs, the following conclusions were made:

1. The designed system is able to generate, monitor and regulate the internal hydrostatic pressure within a valve, up to 4 MPa. This constrain is due to the available pressure transmitter pressure range. However, the designed system is capable of generating and regulating an internal pressure rating of up to 10 MPa.

2. The valve was able to withstand the required internal hydrostatic pressure rating. The testing procedures comply to those described in the following standards: API 598, ASME B16.34, ISO 5208 and MSS-SP-61.

6.2 Recommendations

Recommendations regarding this study are divided into two sections namely, the system design recommendations and the auxiliary recommendations and will be discussed in more detail.

Design Recommendations

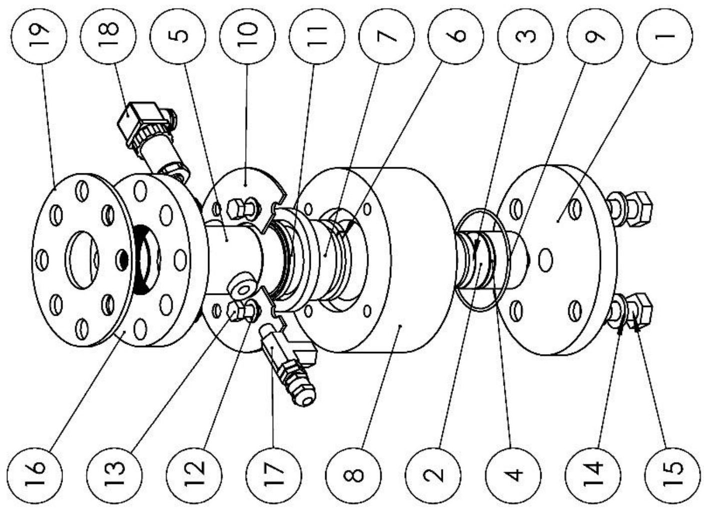
The designed configuration entailed some limitations. The weight of the designed hydrostatic testing subassemblies together with a test valve (weighs about 45 kg), made the handling of the assembly quite challenging. Manoeuvring and tilting this assembly in order to fully vent the air from the system, was also quite difficult. As a design recommendation, the water reservoir interface subassembly’s design should be modified in order to improve the flow of air venting.

Auxiliary Recommendations

The pressure transmitter which was available, has a pressure rating of 0 to 40 bar. This limited the range of valves for testing. To ensure higher pressure rating tests, a pressure transmitter with a greater pressure range, should be used.

Some standards, such as those generated by SANS, dictate that the temperature range of the test fluid should be 23±2 °C. This range is more specific than those presented in other standards, such as API, ASME and ISO, among others. To enable more accurate testing, a test fluid temperature regulator can be installed into the system. Standards, such as SANS, require the testing of torque, endurance and fire testing. This facility can be expanded further to test these parameters.

7 APPENDIX A – Detail Drawings



ITEM NO.	PART NAME	QTY.
1	Piston Base	1
2	Piston	1
3	Piston Seal	1
4	Bearing	2
5	Piston Sleeve	1
6	Stopping Ring	1
7	Guide Bush	1
8	Piston Sleeve Housing	1
9	Piston Sleeve Housing O-Ring	1
10	Perspex Lid	1
11	Wiper Seal	1
12	M10 Washer	4
13	M10 Bolt	4
14	M16 Washer	16
15	M16 Bolt	12
16	Connection Flange	1
17	Safety Relieve Valve	1
18	Pressure Transmitter	1
19	O-Ring Flange	1

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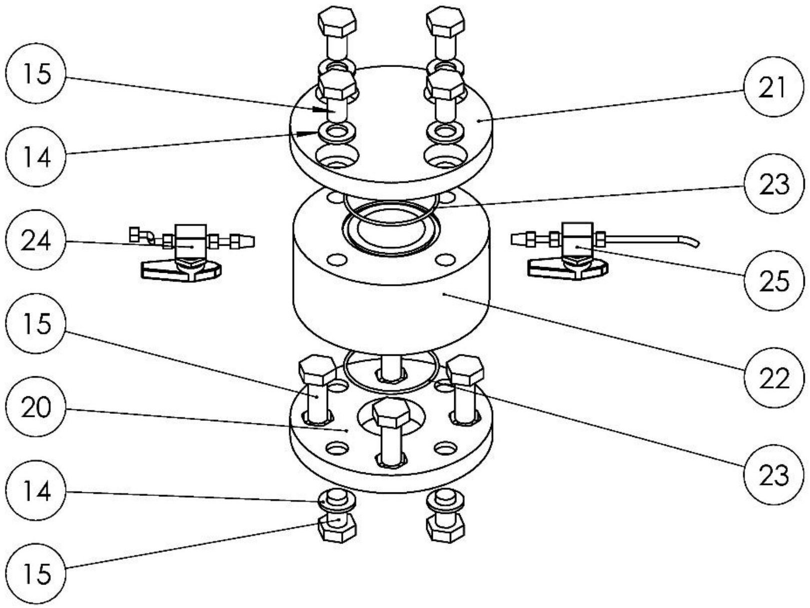
REVIEWER NAME: PROF J MARKGRAAFF	QUANTITY: -		
REVIEW DATE: 04/09/2017	SCALE: 1:5		SHEET 1 OF 1



ITEM NO.	PART NAME	QTY.
20	Bidirectional Flange	1
21	Base Flange	1
22	Water Supplier Housing	1
23	Water Supplier Housing O-Rings	2
24	Water Inlet Ball Valve	1
25	Water Outlet Ball Valve	1

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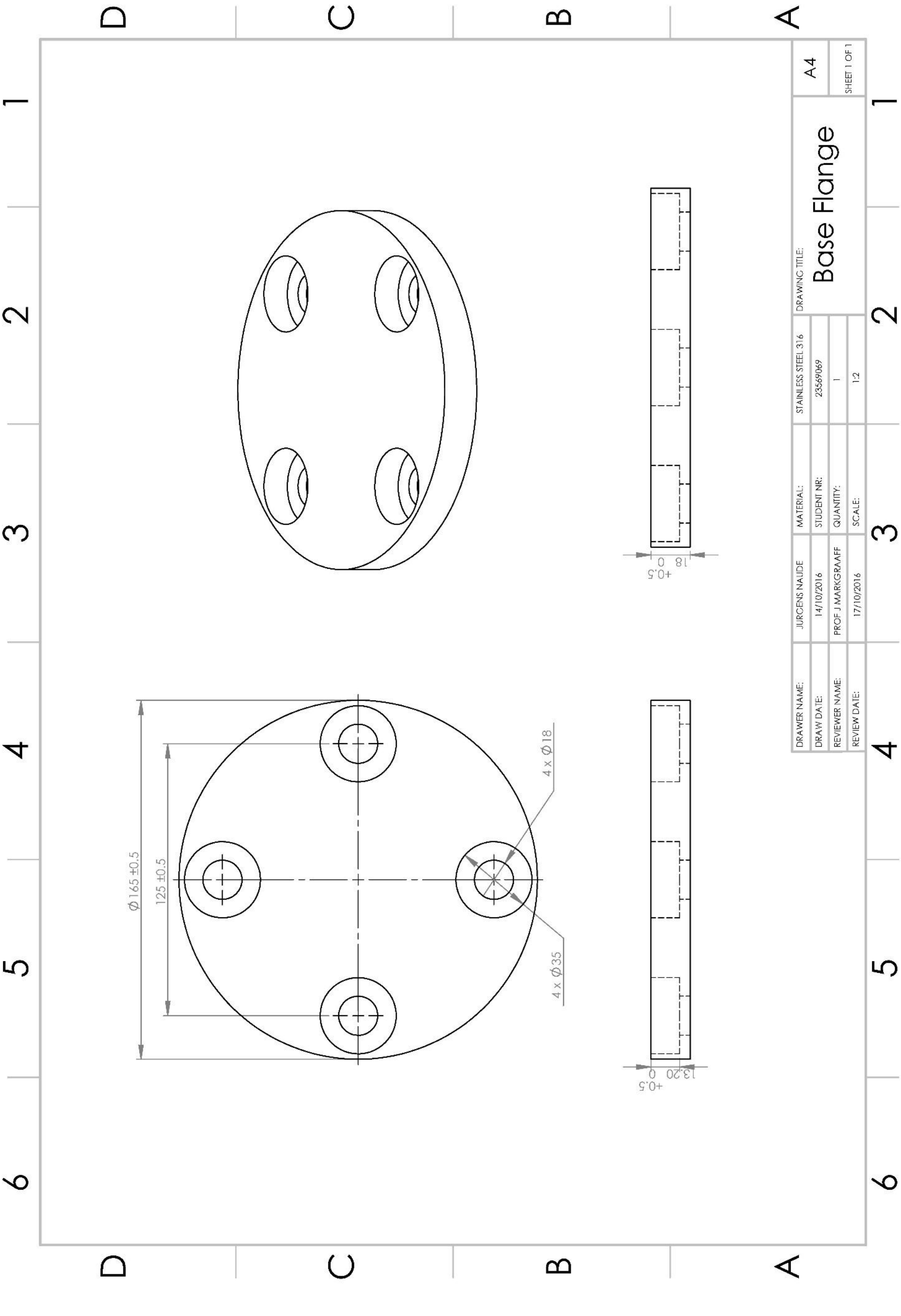
Ø165 ±0.5
125 ±0.5
4 x Ø18
4 x Ø35
18 0 +0.5
13.20 0 +0.5
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DRAW DATE: 14/10/2016
REVIEWER NAME: PROF J MARKGRAAFF
REVIEW DATE: 17/10/2016
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STUDENT NR: 23569069
QUANTITY: 1
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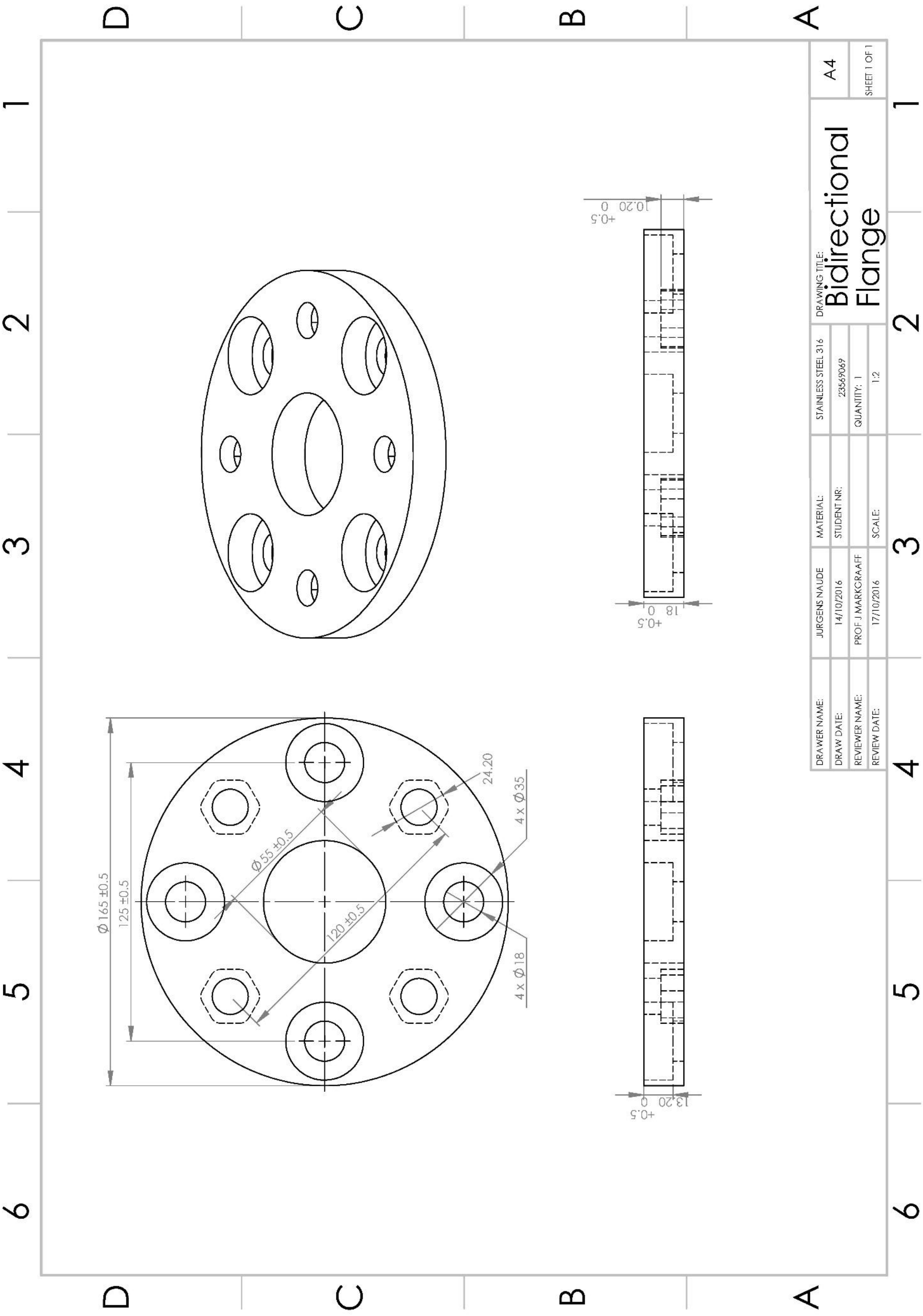
DRAWING TITLE:
Base Flange
A4
SHEET 1 OF 1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REVIEW DATE: 17/10/2016
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STUDENT NR: 23569069
QUANTITY: 1
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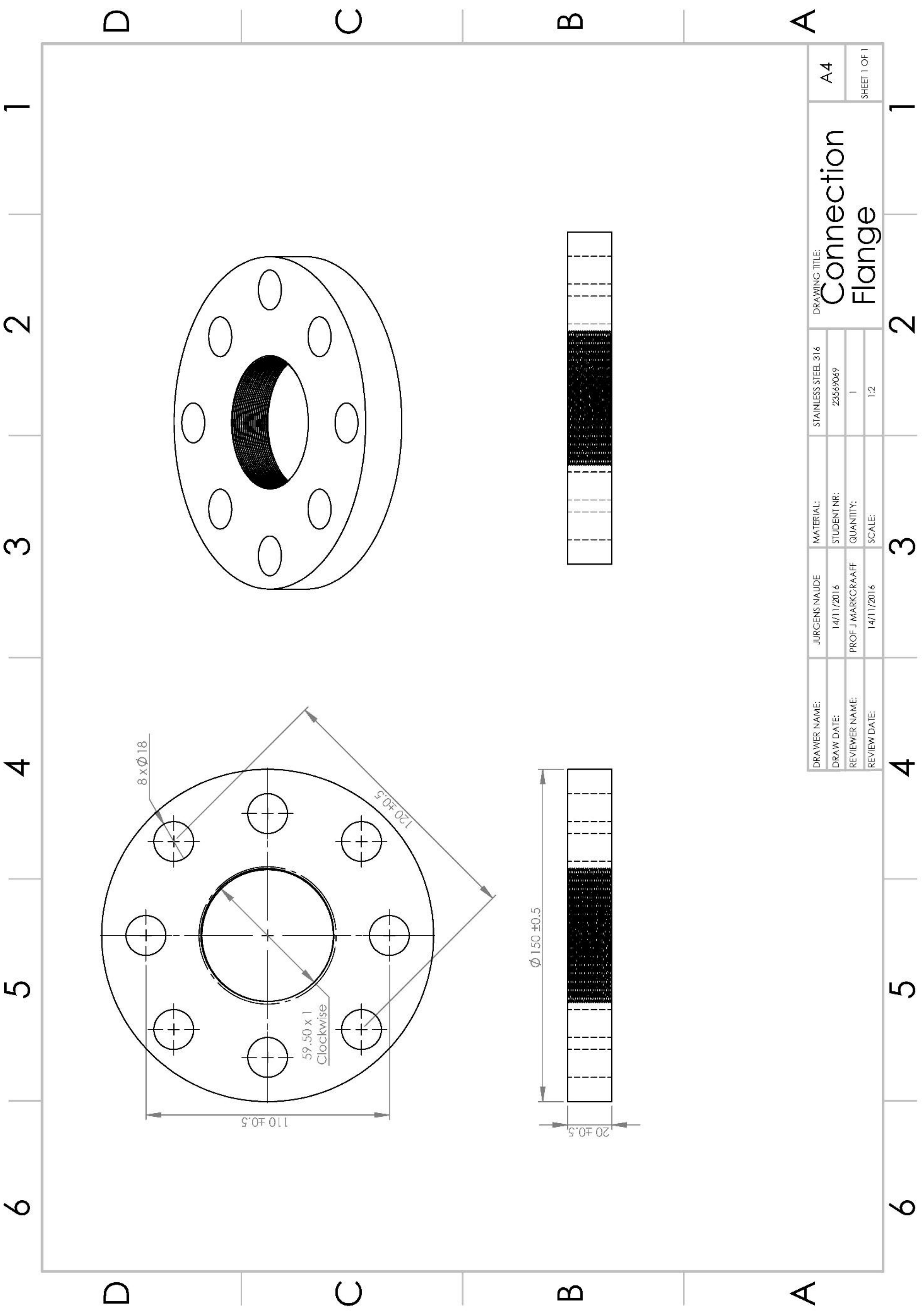
DRAWING TITLE: Bidirectional Flange
A4
SHEET 1 OF 1
Ø165 ±0.5
125 ±0.5
Ø55 ±0.5
120 ±0.5
24.20
4 x Ø35
4 x Ø18



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REVIEW DATE: 14/11/2016
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STUDENT NR: 23569069
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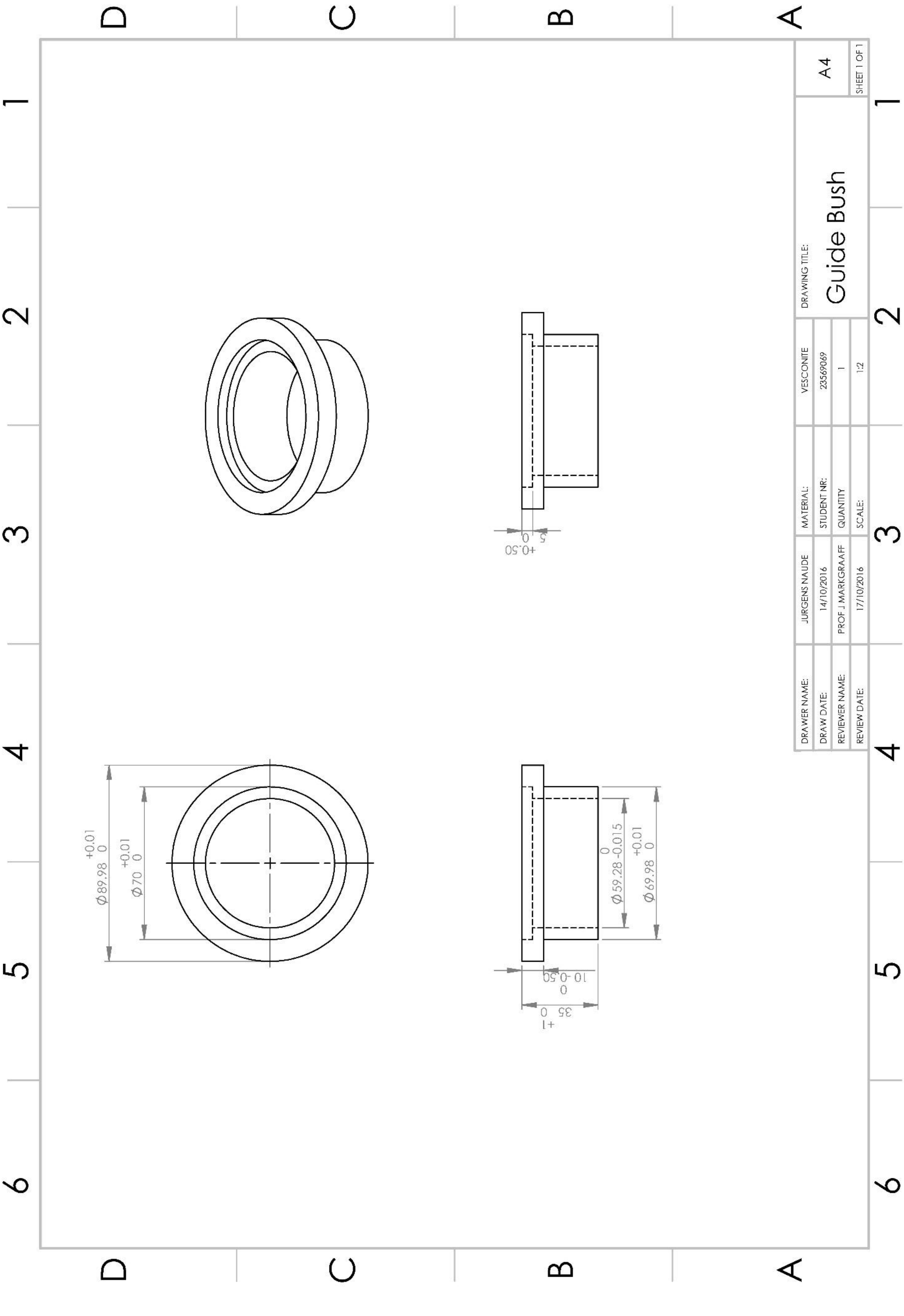
8 x Ø18
120 ±0.5
Ø150 ±0.5
59.50 x 1
Clockwise
110 ±0.5
20 ±0.5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17/10/2016
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23569069
QUANTITY
1
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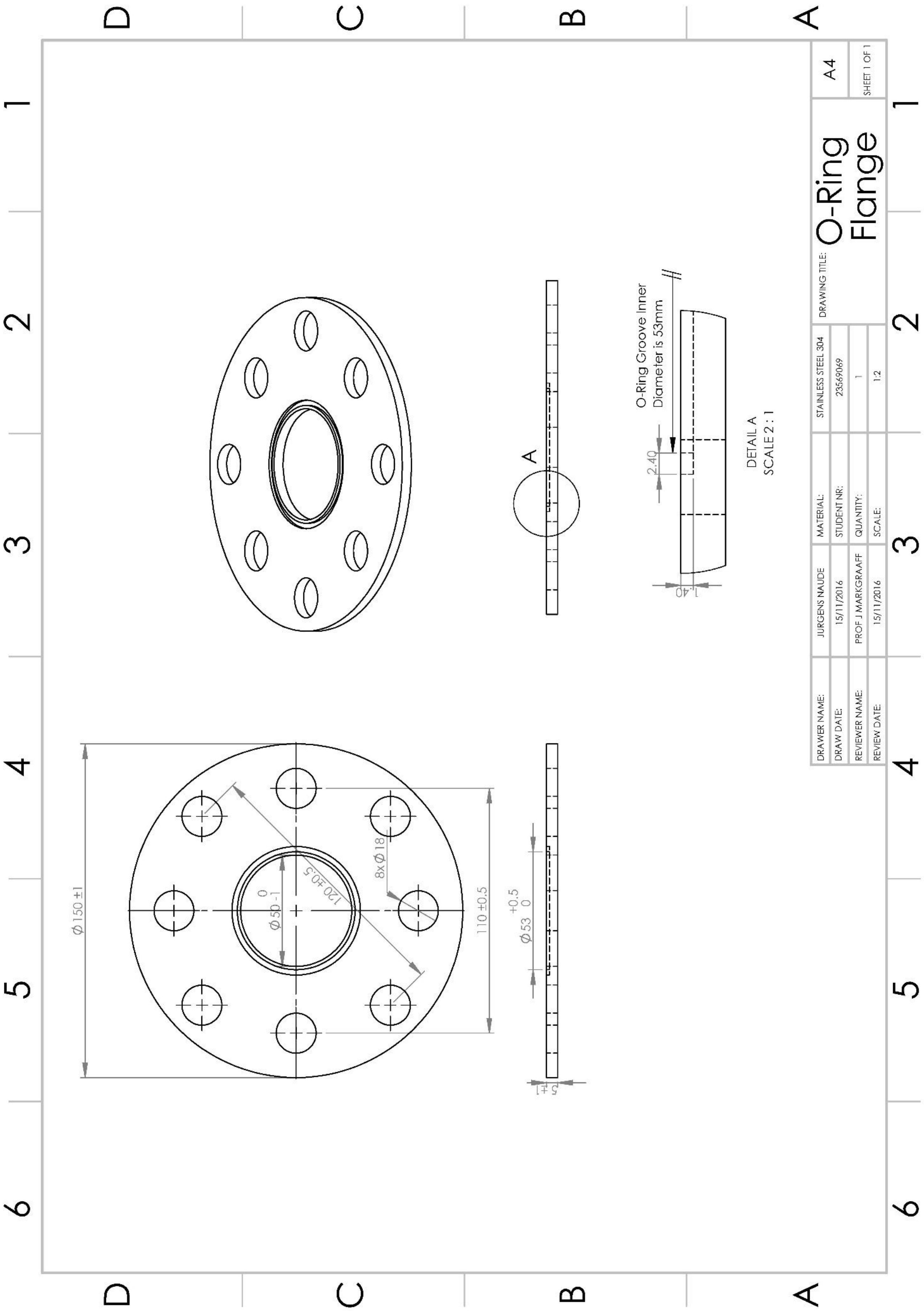
1:2
DRAWING TITLE:
Guide Bush
A4
SHEET 1 OF 1


O-Ring Groove Inner
Diameter is 53mm
DETAIL A
SCALE 2 : 1
2.40
1.40
A
Ø150 ±1
Ø50 0 -1
120 ±0.5
8xØ18
110 ±0.5
Ø53 +0.5 0
5 ±1
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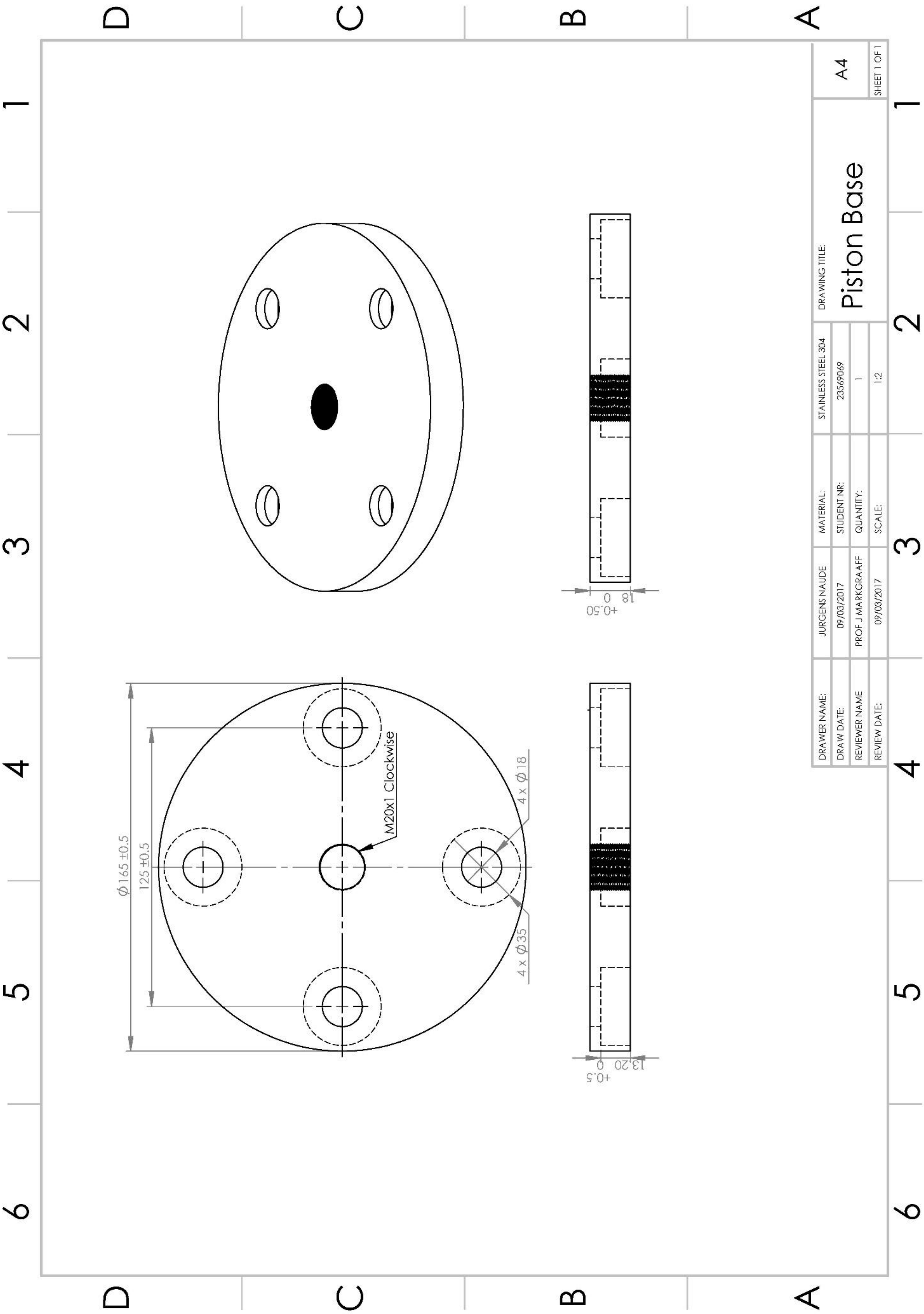
A4
SHEET 1 OF 1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125 ±0.5
M20x1 Clockwise
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4 x Ø35
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REVIEW DATE: 09/03/2017
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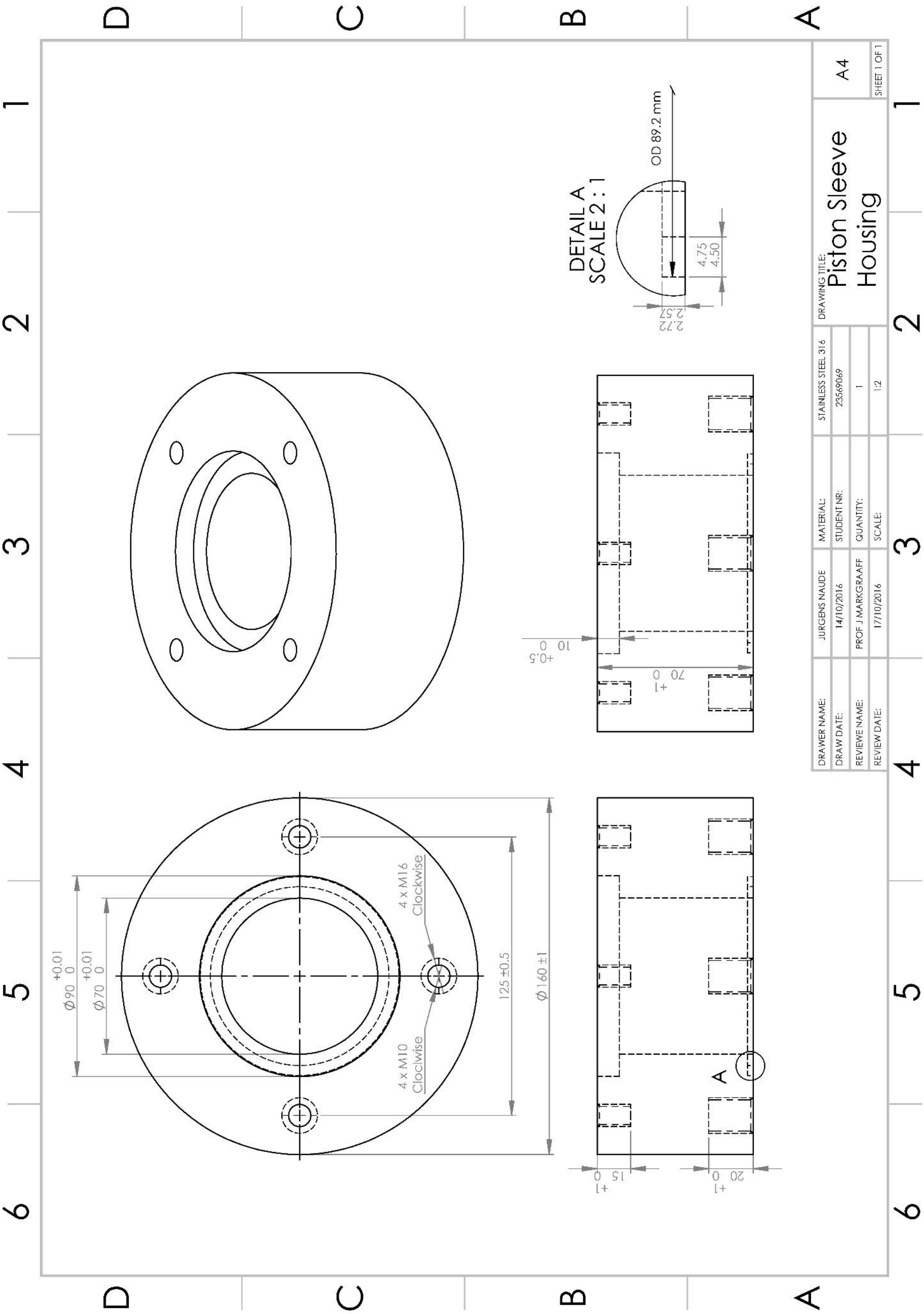
A4
SHEET 1 OF 1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4.50
2.57
2.72
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Clockwise
4 x M10
Clockwise
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Ø70 +0.01 0
125 ±0.5
Ø160 ±1
15 +1 0
20 +1 0
10 +0.5 0
70 +1 0
A
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REVIEW DATE: 17/10/2016
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Piston Sleeve Housing
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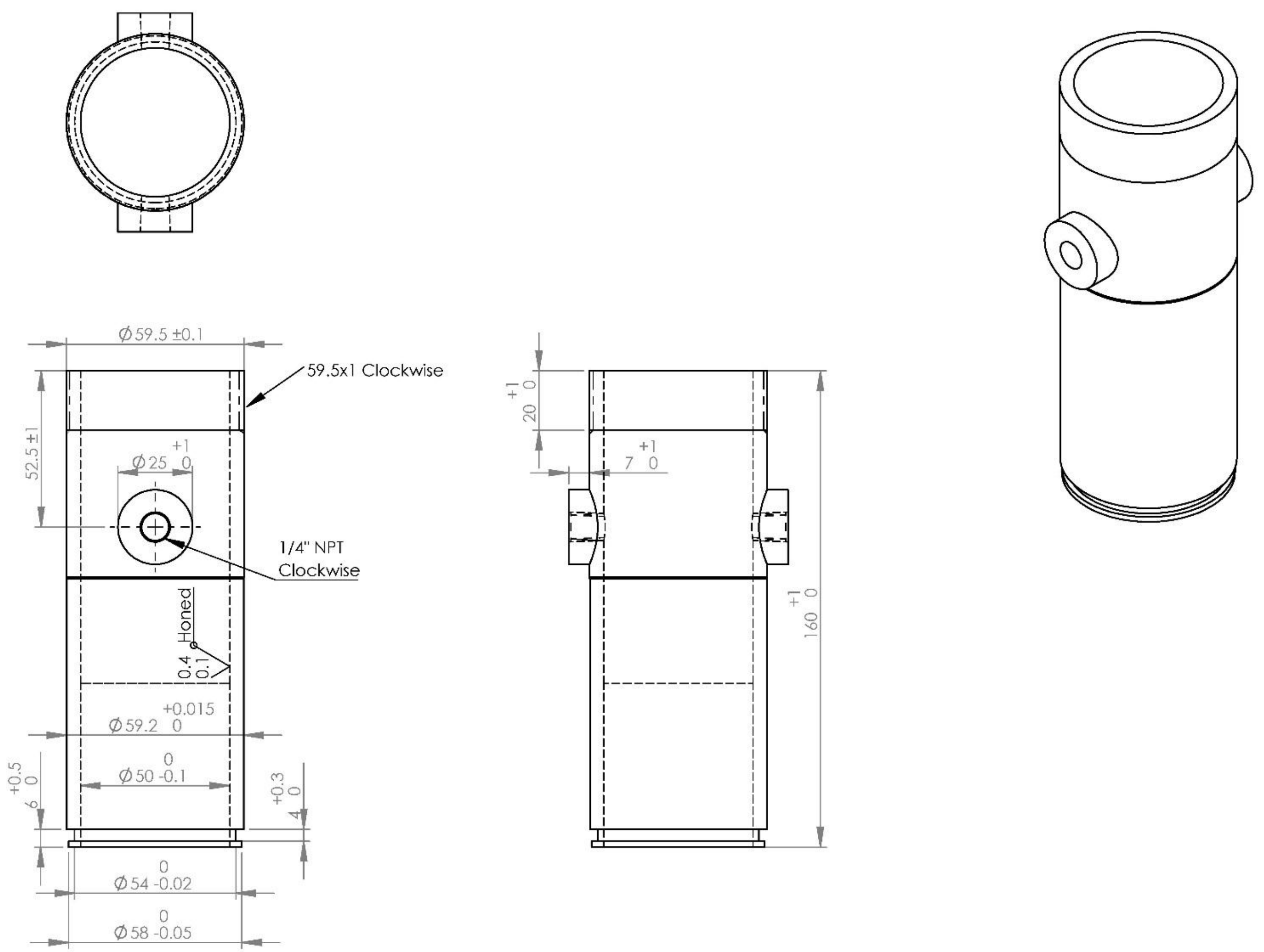
SHEET 1 OF 1



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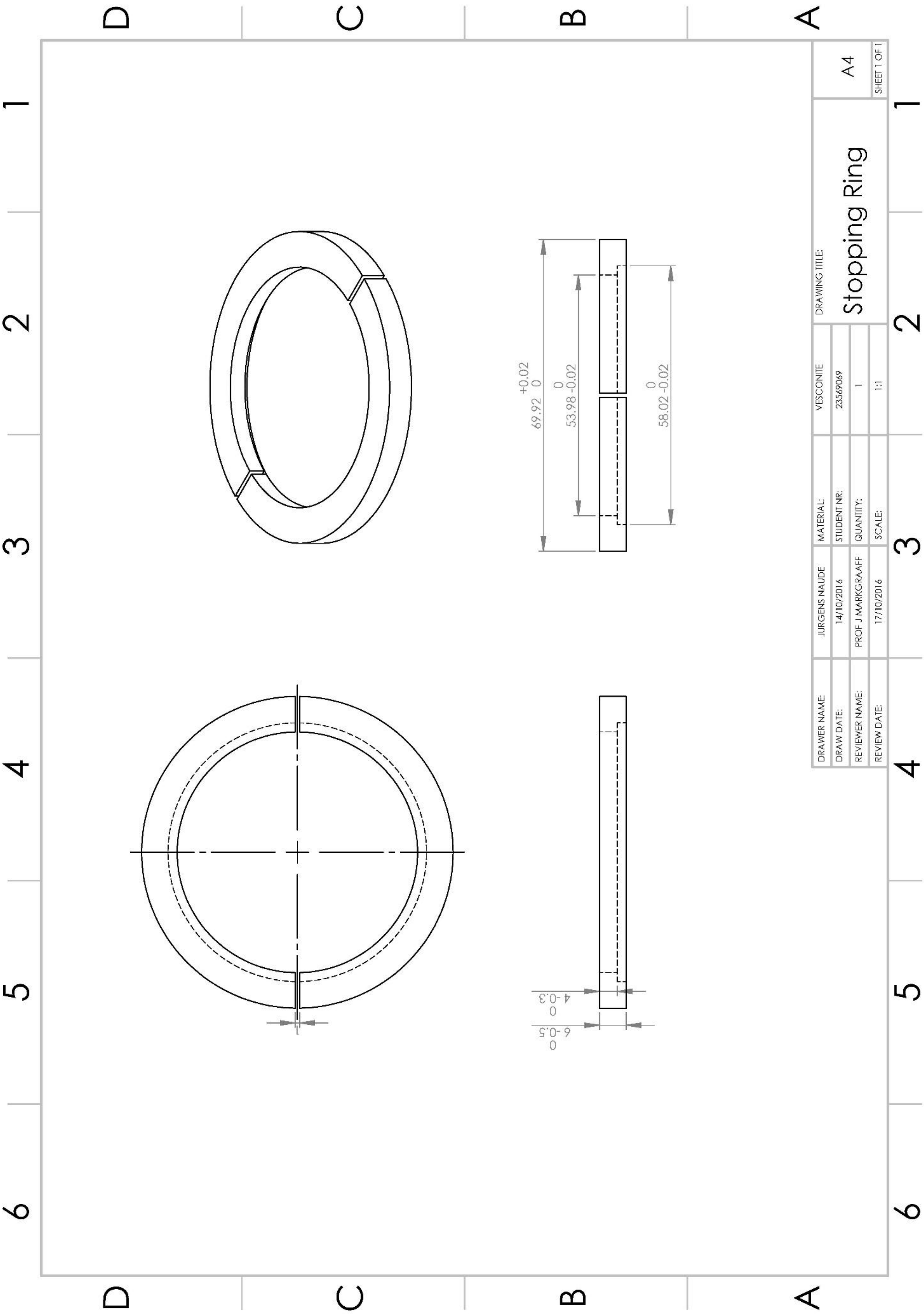
REVIEW DATE:	17/10/2016	SCALE:	1:2	SHEET 1 OF 1


Stopping Ring
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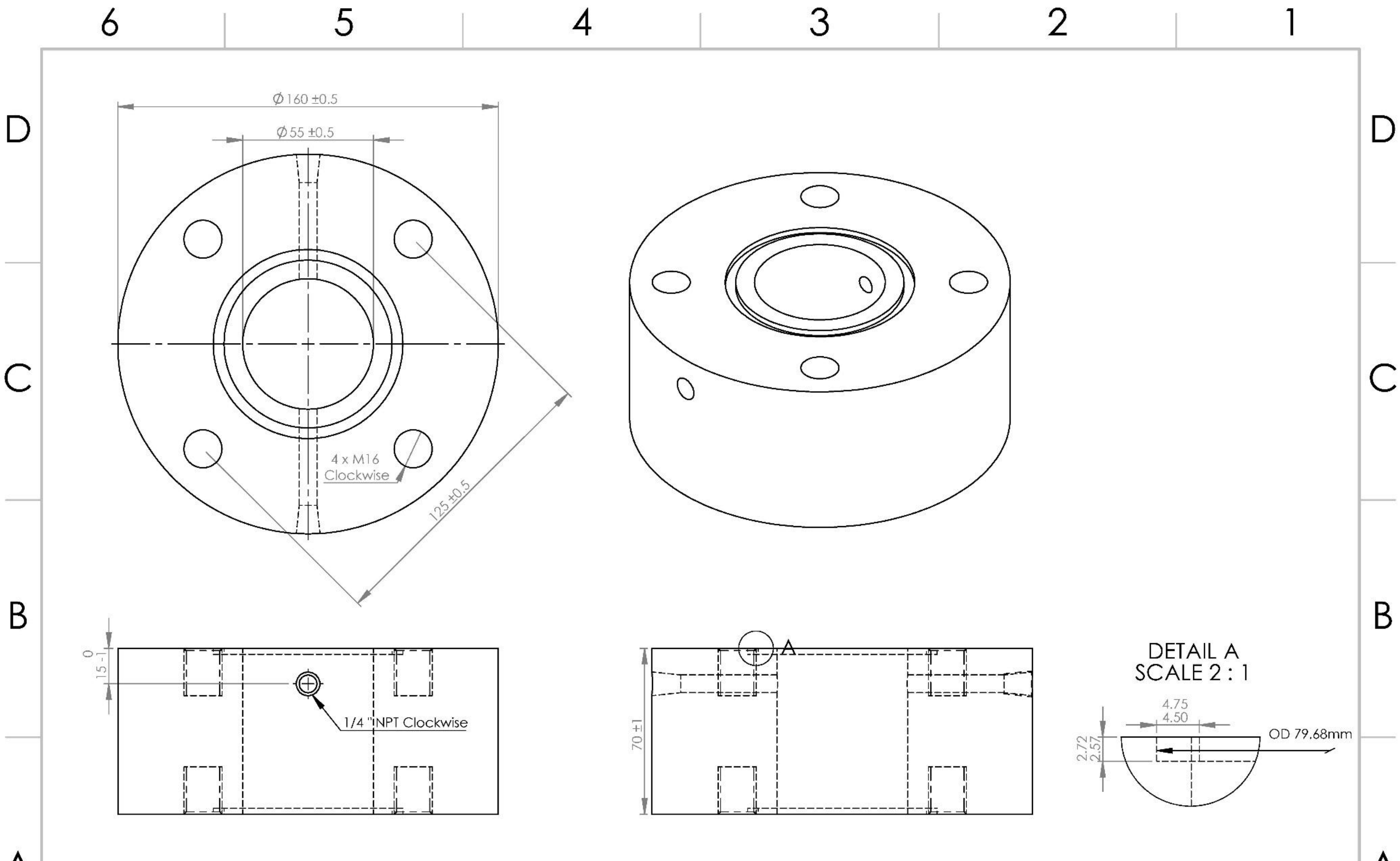
A B C D



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8 APPENDIX B – Calculations

Piston Calculations - Axial and Radial displacement

```
Sigma=10000000                          "Maximum Pressure Rating"
E=2e11                                  "Young's Modulus"
L=0.09650                               "Component Length"
D=0.0493                                "Component
OuterDiameter"
Epsilon_long=-Sigma/E                   "Longitudinal Strain"
delta=Epsilon_long*L                    "Longitudinal Displacement"

v=0.3                                   "Poisson Ratio"
Epsilon_lat=-v*Epsilon_long             "Lateral Strain"
delta_D=Epsilon_lat*D                   "Lateral Displacement (D)"
delta_r=delta_D/2                       "Lateral Displacement (r)"
d`=D+delta_d                            "New Diameter"
```

Sleeve Calculations – Minimum Wall Thickness

```
P=15000000+100000                       "Designed Pressure Rating"
D=d_inner+2*t                           "Component Outer Diameter"
S=115000000                             "Material Stress Value"
E=1                                     "Material Quality Factor"
W=1                                     "Weld Joint Strength Reduction Factor"
d_inner=0,05                            "Component Inner Diameter"
c=0,0005                                "Mechanical Allowance"

Y=0,4                                   "Y - Coefficient"
t=(P*(d_inner+2*(c)))/(2*((S*E*W)-P*(1-Y)))   "Maximum Wall Thickness"
t_m=t+c                                 "Minimum Wall Thickness"
```

Sleeve Calculations – Directional Stress

```
p_i=15000000                            "Designed Pressure Rating"
p_o=100000                              "Atmospheric Pressure"
r_i=0,05/2                              "Component Inner Radius"
r_o=0,0592/2                            "Component Outer Radius"
r=r_o

Sigma_a=(p_i*r_i^2-p_o*r_o^2 )/(r_o^2 - r_i^2)          "Stress in Axial Direction"
Sigma_c=((p_i*r_i^2-p_o*r_o^2)/(r_o^2-r_i^2))-((r_i^2*r_o^2*(p_o-p_i))/(r^2*(r_o^2-r_i^2)))  "Stress in
Circumferential Direction"
Sigma_r= ((p_i*r_i^2-p_o*r_o^2)/(r_o^2-r_i^2))+((r_i^2*r_o^2*(p_o-p_i))/(r^2*(r_o^2-r_i^2)))  "Stress in
Radial Direction"
```

Sleeve Calculations – Radial Displacement

```
p_i=15e6                                                "Designed Pressure Rating"
p_o=100000                                              "Atmospheric Pressure"
d_i=0,050                                               "Componnet Internal Diameter"
d_o=0,0592                                              "Component Outer Diameter"
E=190e9                                                 "Material Young's Modulus"
v=0,29                                                  "Material Poisson Ratio"


u_i=A_1*r_i+A_2/r_i                                     "Component Radial Displacement"
u_o=A_1*r_o+A_2/r_o                                     "Component Outer Displacement"

A_1=((1-v)/E)*((p_i*r_i^2-p_o*r_o^2)/(r_o^2-r_i^2))     "Constant Value A1"
A_2=((1+v)/E)*((p_i-p_o)*(r_i^2*r_o^2))/(r_o^2-r_i^2)   "Constant Value A2"

r_i=d_i/2                                               "Internal Radius"
r_o=d_o/2                                               "Outer Radius"

delta_d_i=2*u_i                                         "Change in Inner Diameter"
delta_d_o=2*u_o                                         "Change in Radial Diameter"
```

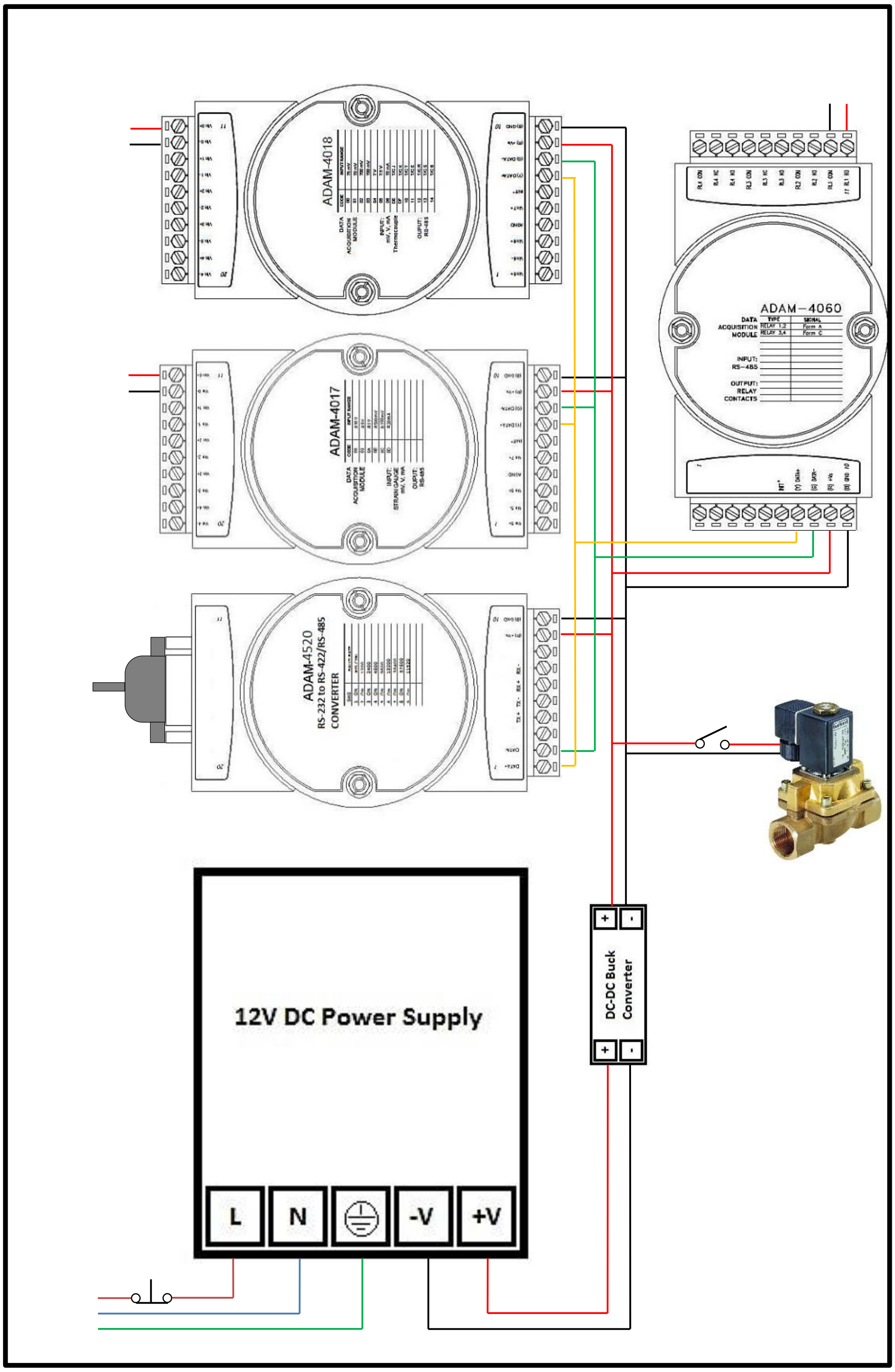
9 APPENDIX C - Source Code

```
Public Class Form1
    Dim Myport As Array             'declare variable Myport for SerialPort
    Dim sec As Integer              'declare variable seconds for timer
    Dim min As Integer              'declare variable minutes for timer
    Dim hr As Integer               'declare variable hours for timer
    Dim TimeSec As String           'declare variable seconds for data graphs
    Dim file As System.IO.StreamWriter

Private Sub Form1_Load(sender As Object, e As EventArgs) Handles MyBase.Load
            Myport = IO.Ports.SerialPort.GetPortNames()     'Find available SerialPorts
            cbx_Connection.Items.AddRange(Myport)           'Show available SerialPorts in Combobox
            SerialPort1.NewLine = vbCr
    End Sub

Private Sub cbx_Connection_SelectedIndexChanged(sender As Object, e As EventArgs) Handles
cbx_Connection.SelectedIndexChanged
            SerialPort1.PortName = cbx_Connection.SelectedItem      'Select SerialPort
            SerialPort1.Open()                                      'Open SerialPort
            ckx_ShellTest.Enabled = True                            'Enable Shell Test Checkbox
            ckx_ClosureTest.Enabled = True                          'Enable Closure Test Checkbox
    End Sub

Private Sub ckx_ShellTest_CheckedChanged(sender As Object, e As EventArgs) Handles ckx_ShellTest.CheckedChanged
            If ckx_ShellTest.Checked = True Then
                        ckx_ClosureTest.Enabled = False             'Disable Closure Test Checkbox
                        ckx_ClosureTest.Checked = False             'Uncheck Closure Test Checkbox
                        ckx_InletValve.Text = " Inlet Valve Closed"     'Inlet Valve Checkbox text changes to Inlet Valve Closed
                        ckx_OutletValve.Text = "Outlet Valve Closed"    'Outlet Valve Checkbox text changes to Outlet Valve
                                                                         Closed
                        ckx_Gate.Text = "Gate Open"                 'Gate Checkbox text changes to Gate Open
                        ckx_InletValve.Enabled = True               'Enable Inlet Valve Checkbox
                        ckx_OutletValve.Enabled = True              'Enable Outlet Valve Checkbox
                        ckx_Gate.Enabled = True                     'Enable Gate Checkbox
                        ckx_InletValve.Checked = False              'Uncheck Inlet Valve Checkbox
                        ckx_OutletValve.Checked = False             'Uncheck Outlet Valve Checkbox
                        ckx_Gate.Checked = False                    'Uncheck Gate Checkbox
            Else
                        ckx_ClosureTest.Enabled = True              'Enable Closure Test Checkbox
                        ckx_InletValve.Enabled = False              'Disable Inlet Valve Checkbox
                        ckx_OutletValve.Enabled = False             'Disable Outlet Valve Checkbox
                        ckx_Gate.Enabled = False                    'Disable Inlet Valve Checkbox
                        ckx_InletValve.Checked = False              'Uncheck Inlet Valve Checkbox
                        ckx_OutletValve.Checked = False             'Uncheck Outlet Valve Checkbox
                        ckx_Gate.Checked = False                    'Uncheck Gate Checkbox
            End If
    End Sub

Private Sub ckx_ClosureTest_CheckedChanged(sender As Object, e As EventArgs) Handles ckx_ClosureTest.CheckedChanged
            If ckx_ClosureTest.Checked = True Then
                        ckx_ShellTest.Enabled = False               'Disable Shell Test Checkbox
                        ckx_ShellTest.Checked = False               'Uncheck Shell Test Checkbox
                        ckx_InletValve.Text = " Inlet Valve Closed"     'Inlet Valve Checkbox text changes to Inlet Valve Closed
                        ckx_OutletValve.Text = "Outlet Valve Open"      'Outlet Valve Checkbox text changes to Outlet Valve
                                                                         Open
                        ckx_Gate.Text = "Gate Closed"               'Gate Checkbox text changes to Gate Closed
                        ckx_InletValve.Enabled = True               'Enable Inlet Valve Checkbox
                        ckx_OutletValve.Enabled = True              'Enable Outlet Valve Checkbox
                        ckx_Gate.Enabled = True                     'Enable Gate Checkbox
                        ckx_InletValve.Checked = False              'Uncheck Inlet Valve Checkbox
                        ckx_OutletValve.Checked = False             'Uncheck Outlet Valve Checkbox
                        ckx_Gate.Checked = False                    'Uncheck Gate Checkbox
            Else
                        ckx_ShellTest.Enabled = True                'Enable Shell Test Checkbox
                        ckx_InletValve.Enabled = False              'Disable Inlet Valve Checkbox
                        ckx_OutletValve.Enabled = False             'Disable Outlet Valve Checkbox
                        ckx_Gate.Enabled = False                    'Disable Inlet Valve Checkbox
                        ckx_InletValve.Checked = False              'Uncheck Inlet Valve Checkbox
                        ckx_OutletValve.Checked = False             'Uncheck Outlet Valve Checkbox
```

```
                ckx_Gate.Checked = False                    'Uncheck Gate Checkbox
            End If
    End Sub

Private Sub btn_Continue_Click(sender As Object, e As EventArgs) Handles btn_Continue.Click
        Timer2.Start()                                      'Start Timer
    End Sub

Private Sub Timer2_Tick(sender As Object, e As EventArgs) Handles Timer2.Tick
        Timer2.Interval = 1000              'Timer Interval
        TimeSec = TimeSec + 1               'data seconds step
        If ckx_OutletValve.Checked = True And ckx_InletValve.Checked = True And ckx_Gate.Checked = True Then
                SerialPort1.WriteLine("#010")                               'Retrieve data in channel 1 of ADAM
                                                                             4018
                lbl_LiqTempData.Text = SerialPort1.ReadLine.TrimStart(">", "+")   'Trim and display Temprature
                SerialPort1.WriteLine("#020")                               'Retrieve data in channel 1 of ADAM
                                                                             4017
                lbl_PressureData.Text = (((5 / 2) * SerialPort1.ReadLine.TrimStart(">", "+")) - 10) / 10     'Convert, trim
                                                                            and display pressure rating
                Chart1.Series("Pressure").Points.AddXY(TimeSec, lbl_PressureData.Text)          'Pressure Time Graph
                Chart2.Series("Temperature").Points.AddXY(TimeSec, lbl_LiqTempData.Text)       'Temperature Time Graph
                file = My.Computer.FileSystem.OpenTextFileWriter("C:\Users\DSTchair\Documents\Pressure.txt", True)
                                                                            'Save Converted Pressure Ratings
                file.WriteLine(lbl_PressureData.Text)
                file.Close()
                file = My.Computer.FileSystem.OpenTextFileWriter("C:\Users\DSTchair\Documents\Temperature.txt",
                True)                                                       'Save Converted Temperture
                                                                             Ratings
                file.WriteLine(lbl_LiqTempData.Text)
                file.Close()
        End If
        If ckx_OutletValve.Checked = True And ckx_InletValve.Checked = True And ckx_Gate.Checked = True And
        ckx_UseSetPressureFunction.Checked = True Then
                SerialPort1.WriteLine("#010")                               'Retrieve data in channel 1 of ADAM
                                                                             4018
                lbl_LiqTempData.Text = SerialPort1.ReadLine.TrimStart(">", "+")   'Trim and display temprature
                SerialPort1.WriteLine("#020")                               'Retrieve data in channel 1 of ADAM
                                                                             4017
                lbl_PressureData.Text = (((5 / 2) * SerialPort1.ReadLine.TrimStart(">", "+")) - 10) / 10     'Convert, trim
                                                                            and display pressure rating

                If Convert.ToDouble(lbl_PressureData.Text) <= Convert.ToDouble(tbx_SetPressure.Text) Then    'Set
                                                                            regulate upper limit
                        SerialPort1.WriteLine("#031101")                    'Power up press
                        SerialPort1.ReadLine()
                Else
                        SerialPort1.WriteLine("#031100")                    'Power down press
                        SerialPort1.ReadLine()
                End If
            Chart1.Series("Pressure").Points.AddXY(TimeSec, lbl_PressureData.Text)          'Pressure Time Graph
            Chart2.Series("Temperature").Points.AddXY(TimeSec, lbl_LiqTempData.Text)       'Temperature Time Graph
            file = My.Computer.FileSystem.OpenTextFileWriter("C:\Users\DSTchair\Documents\Pressure.txt", True)     'Save
                                                                            Converted Pressure Ratings
            file.WriteLine(lbl_PressureData.Text)
            file.Close()
            file = My.Computer.FileSystem.OpenTextFileWriter("C:\Users\DSTchair\Documents\Temperature.txt", True)   'Save
                                                                            Converted Temperature Ratings
            file.WriteLine(lbl_LiqTempData.Text)
            file.Close()
        End If
    End Sub

Private Sub Timer1_Tick(sender As Object, e As EventArgs) Handles Timer1.Tick
        Timer1.Interval = 1000              'Set Timer Interval
        sec = sec + 1                       'seconds step
        lbl_Sec.Text = sec                  'Allocate seconds to Seconds lable
        lbl_min.Text = min                  'Allocate minutes to Minutes lable
        lbl_hour.Text = hr                  'Allocate hours to Hours lable

        If sec = 59 Then
            min = min + 1                   'minutes step
```

```
            sec = 0                         'Reset seconds
        End If

        If min = 59 Then
            hr = hr + 1                     'hours step
            min = 0                         'Reset minutes
        End If
    End Sub

Private Sub btn_Start_Click(sender As Object, e As EventArgs) Handles btn_Start.Click
        Timer1.Start()                      'Start Timer
        sec = 0                             'Reset Seconds
        min = 0                             'Reset Minutes
        hr = 0                              'Reset Hours
    End Sub

Private Sub btn_Stop_Click(sender As Object, e As EventArgs) Handles btn_Stop.Click
        Timer1.Stop()                       'Stop Timer
    End Sub

Private Sub btn_Clear_Click(sender As Object, e As EventArgs) Handles btn_Clear.Click
        sec = 0                             'Reset Seconds
        min = 0                             'Reset Minutes
        hr = 0                              'Reset Hours
        lbl_Sec.Text = "0"                  'Set Seconds lable text to "0"
        lbl_min.Text = "0"                  'Set Minutes lable text to "0"
        lbl_hour.Text = "0"                 'Set Hours lable text to "0"
    End Sub

Private Sub btn_StopPress_Click(sender As Object, e As EventArgs) Handles btn_StopPress.Click
        Timer2.Stop()                       'Stop Timer
        SerialPort1.WriteLine("#031100")    'Power down Press
        SerialPort1.ReadLine()
    End Sub

Private Sub ckx_UseSetPressureFunction_CheckedChanged(sender As Object, e As EventArgs) Handles
ckx_UseSetPressureFunction.CheckedChanged
        If ckx_UseSetPressureFunction.Checked = True Then
            tbx_SetPressure.Enabled = True          'Enable Set Pressure Textbox
        Else
            tbx_SetPressure.Enabled = False         'Disable Set Pressure Textbox
        End If
    End Sub
End Class
```

10 APPENDIX D – DB Board Layout



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