



VALE

Major Project

Lee Wellington – LW094662

How to apply a continuous force upon a gland follower
to improve the reliability of packing gland seals?

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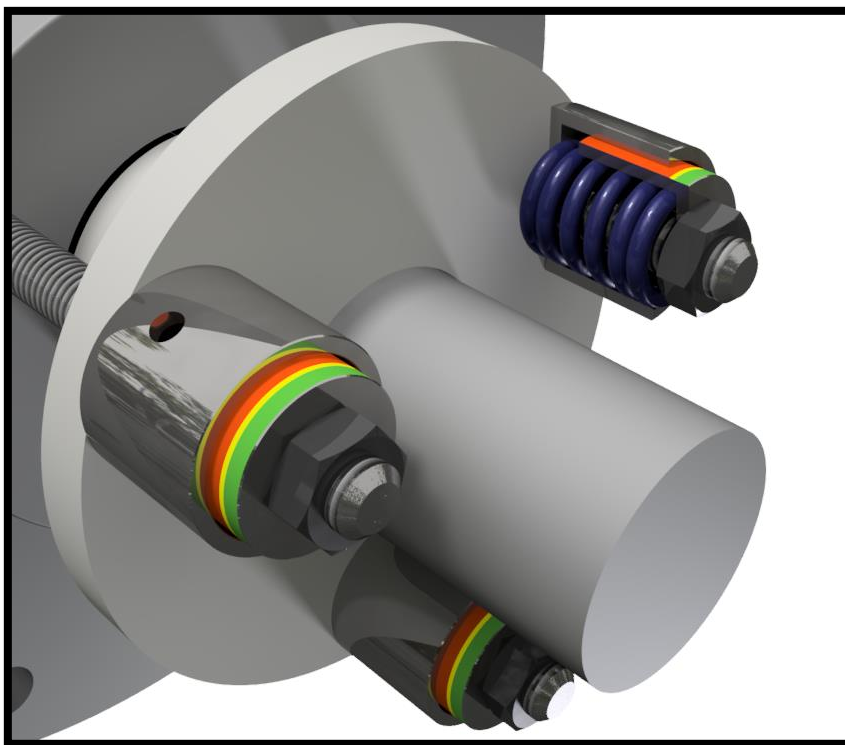
Module Code: SGDM6019



Summary

Underperforming 'packing gland' seals within the Kiln Plant at Vale Europe have caused a significant rise in plant evacuations over the past 3 years. A literature review on the issue revealed that 'packing', when compressed over a prolonged period, is susceptible to the effects of stress relaxation. As the packing gradually fails and deforms, its inability to resist the force subjected by the 'gland follower' causes a loss of compression which invariably results in seal failure. It was therefore hypothesised that if a continuous force could be applied to a gland follower, the seal assembly would self-adjust and negate the effects of stress relaxation providing a more reliable seal. From executing the main elements of the design cycle (research, specification, concept design, detailed design and manufacture) a socket spring gauge device was produced, functionally tested and proposed as a potential solution to the problem.

Figure 1 –Socket Spring Gauge



Acknowledgements

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Nomenclature

H₂ - Hydrogen

CO – Carbon Monoxide

CO₂ – Carbon Dioxide

Ni - Nickel

Ni(CO)₄ – Nickel Carbonyl Gas

COMAH – Control of Major Accident Hazards

SAP ERP – Vale’s Enterprise Resource Planning software used to manage business processes.

SAP IM – Vale’s Incident Management Software for recording, investigating and reporting all dangerous or potentially dangerous events that occur onsite.

N/mm – Newtons per Millimetre

mm H₂O – Millimetres Water Gauge (Pressure)

In H₂O – Inches Water Gauge (Pressure)

PSI – Pounds per Square Inch

Amps – Units used to measure electrical current.

HP – Horsepower (British imperial units used for measuring power).

Autodesk Inventor – Computer Aided Design software that ‘*provides 3D mechanical design, simulation, tooling creation, and design communication*’ (CADSPEC, 2020)

FEA – Finite Element Analysis, *a computerised method for predicting how a product reacts to real-world forces, vibration, heat, fluid flow and other physical effects*’ (Autodesk, 2020).

Chapter 1 – Introduction

Introduction.

Vale – Clydach Nickel Refinery

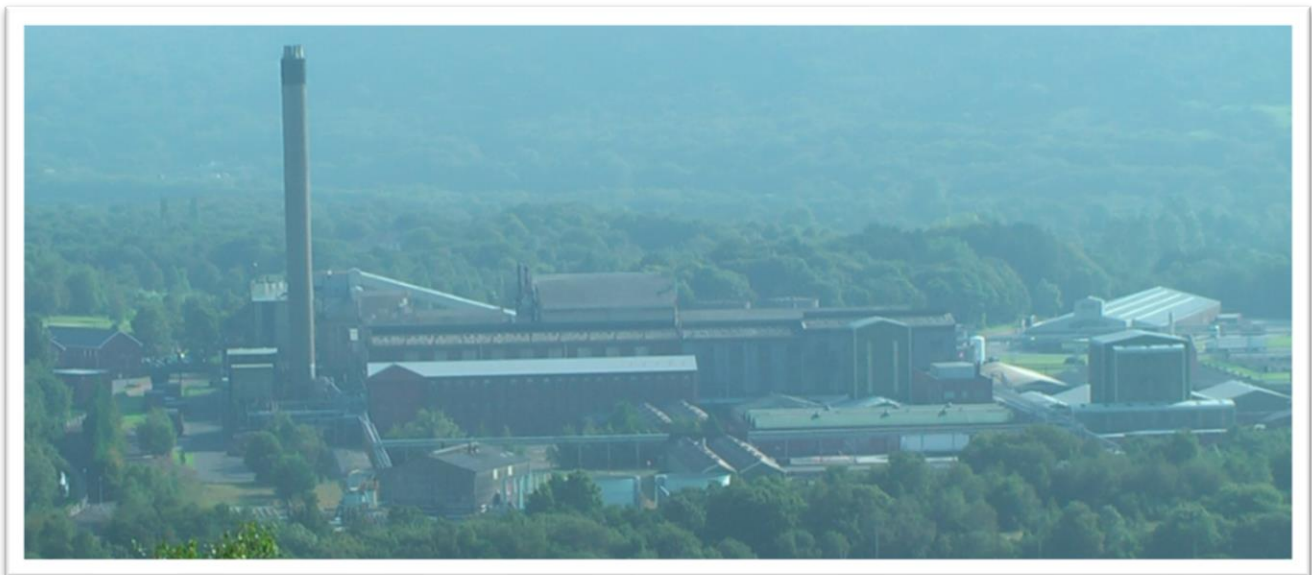
“Located in the Swansea Valley, Clydach’s nickel refinery has been in operation for over 100 years. From using the Mond nickel carbonyl process the site produces high quality refined nickel in the form of powders and pellets.

There are 3 main process areas within the refinery;

- The Gas Plant produces gases required in the refining process (H_2 , CO, CO_2).*
- The Kiln Plant produces nickel carbonyl gas ($Ni(CO)_4$)*
- The Pellet and Powder Plant produces nickel products by thermal decomposition of $Ni(CO)_4$ gas.*

Due to the hazardous nature of the processes and products used in each plant, the refinery is classified under COMAH (Control of Major Accident Hazards) regulations as an upper tier COMAH site.” (Wellington, 2019)

Figure 2 - Image of Vale (Clydach Refinery)



Kiln Plant

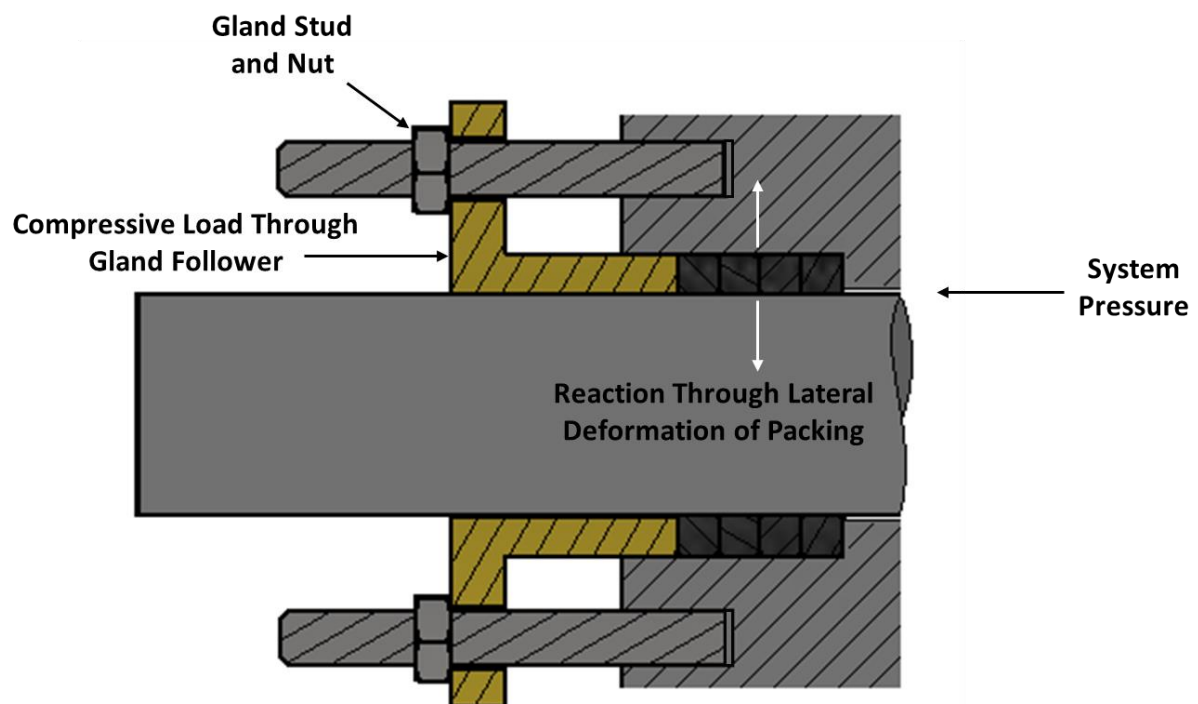
The Kiln Plant is responsible for extracting the product gas (*nickel carbonyl gas - $Ni(CO)_4$*) from impure nickel oxide (75% Ni) that is imported from Canada or Japan as fine granules. The plant consists of two identical lines which each have four rotary kilns that are connected in series via elevators, conveyors and rotary valves. In the first ‘Reduction’ kiln, the oxide is reduced by a counter-current flow of hydrogen between 300°C and 500°C producing a finely divided matte of impure nickel. The matte passes by gravity to the second ‘Activation’ kiln, where sulphur bearing gas and hydrogen are passed over it. Activated matte is then transferred to the water-cooled third ‘Volatilisation’ kiln where it is reacted with carbon monoxide (CO). The nickel is converted to gaseous nickel carbonyl ($Ni(CO)_4$) in an exothermic reaction with an excess of carbon monoxide, which is passed through a series of cyclones and bag filters to

a ring main in the Pellet and Powder Plant. Unreacted metallic residues are transferred to the fourth ‘stabilisation’ kiln where partial oxidation makes the residues non-pyrophoric ready for return to Canada for further processing.

Aim

A literature review on the significant rise of Kiln plant evacuations revealed that over half (57/107) the total (evacuation) incidents recorded since 2015 were attributed to ‘packing glands’ leaking on conveyors and rotary valves. Such ‘loss of containment’ incidents pose a considerable threat to the surrounding environment and health and safety of plant personnel due to the hazardous nature of the products used within the extraction process. It is estimated on average, for every plant evacuation triggered there is a loss of 2 hours productivity per Kiln plant worker. Depending on the level of maintenance activity, there can be between 10 to 30 workers within the plant at any given time. Although there is no automatic plant shutdown feature built into the plant evacuation procedure, there is a substantial risk to production every time an evacuation occurs. If a leak cannot be found and plant issues arise, the plant Superintendent would be forced into activating emergency shutdown valves that would then halt production. If the sealing reliability of packing glands could be improved, the volume of (gland related) evacuation incidents would be greatly reduced therefore, eliminating the risks and losses highlighted. As shown in figure 3, a typical gland packing seal is ‘made up from four or five turns of packing fitted in what is often referred to as the gland housing or stuffing box. The free packing will be a light interference fit in the gland but will be relatively easy to push into place. It is then axially compressed by the gland follower which creates a radial expansion of the packing to create a sealing force’ (Flitney, 2007)

Figure 3 - Cross-section of Simple Gland Arrangement



Research and failure mode analysis suggested the most common cause for seal failure was ‘insufficient load’ upon the gland follower. To explain why there was a loss of (compressive) load over time, a journal produced by Tashiro & Yoshida (1990) revealed that the ‘Tightening

pressure of gland packings decreases with time owing to stress relaxation of the packings'. It was therefore hypothesised that, as a result of stress relaxation that occurs in packing, the current gland 'loading' method (gland stud and nut) is unable to continuously apply enough force upon the packing to maintain a seal over a prolonged period of time. Consequently, the aim of this project is to design an improved method of 'loading' the 'gland follower', so a continuous force is applied to negate the effects of stress relaxation. For simplification, the design process will be based on a conveyor packing gland and after proof of concept a cost proposal for applying the solution to all Kiln plant packing glands will be made. The roles identified within the project (organisation) structure will consist of a;

- Project Engineer (L.Wellington) - responsible for planning, coordinating and executing the work entailed in the project.
- Project Engineering Manager (M.Baynham) – responsible for monitoring the project progression and determining the budget and timeframe available.
- Kiln plant Superintendent (D.Bousher) – responsible for ensuring the design solution meets the customer requirements.

A successfully project will be deemed 'on-time' if completed by the 6th March 2020.

Chapter 2 – Research into Packing Glands

Background

A 'packing gland' is one of the oldest and simplest form of dynamic seal that can be used on rotating and reciprocating applications. In its earliest form, the design was primarily used on pumps whereby, *'gland packing was made from old ropes and natural fibre products that were packed around the shaft physically stuffing the gap with the material to reduce leakage'* (AESSEAL, 2019). Through advancements in packing material and braiding technology, gland packing seals are still commonly used today on various valve and pump designs.

Figure 4 - Braided Graphite Packing



(Sunwell Seals, 2019)

In comparison to other seals such as lip or mechanical seals, a packing gland is deemed a much more economic and robust sealing solution. Some of the strengths of using the simplistic design consist of;

- Low installation, operation and maintenance costs.
- Ability to seal aggressive and abrasive media.
- Can tolerate large amounts of axial and radial shaft movement.
- Can tolerate large amounts of vibration and cavitation.
- Seal failure is gradual and often not catastrophic. If leakage does occur it can be stopped by adjusting the gland.
- Quick seal turnaround (does not require decoupling the shaft drive).

As a result of the forementioned benefits, packing glands are still considered to be the most effective seal on *'many applications such as fire water pumps, effluent water pumps, valve packing's, etc,'* (Mechanical Engineering Site, 2018).

There are however, some disadvantages with using packing gland seals such as;

- Shaft wear can occur on due to friction.
- Shaft speed limitations.
- Operating pressure limitations.

Main Components

The main components of a typical ‘packing gland’ assembly consist of a;

Stuffing box / Gland Housing

A ‘stuffing box’ is a cylindrical cavity that forms a chamber between the end cover (or casing) and the drive shaft. Its purpose is to house packing material to create a seal where the drive shaft enters the application. When the packing is compressed inside the assembly it expands radially, applying pressure to the stuffing box bore and (rotating) drive shaft. To assist sealing, a relatively low surface roughness is required on the bore of the gland housing. At Vale a surface roughness of 32 Ra (μin) microinches is required on all (conveyor and rotary valve) end covers (appendix 1).

Figure 5 - Stuffing Box

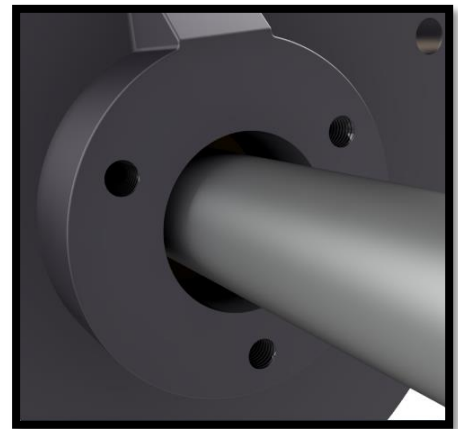


Figure 6 - Compression Packing



Compression Packing

Typically, compression packing ‘is braided yarn made from a variety of materials such as graphite, aramid, and PTFE’ (Robco, 2019) that is extensively used on sealing applications. As shown in figure 6, packing has a square cross sectional area which can be purchased in various different sizes. To determine the most suitable packing for a given application, a number of factors are to be considered, such as;

- ‘Size – What are the stuffing box dimensions?’
- Temperature – What is the temperature of the medium?’
- Application – What type of equipment is it?’
- Media – What medium is being sealed?’
- Pressure – What is the internal pressure being sealed?’
- Shaft speed – What is the surface speed of the shaft?’

(Specialist Sealing Products, 2019)

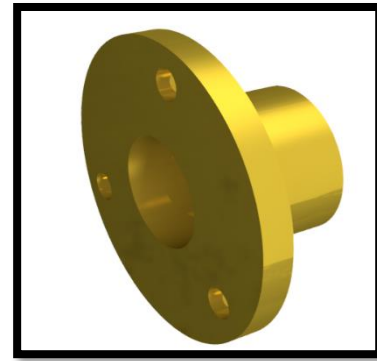
(Pumps & Systems, 2015)

Packing sets usually consist of 4 or 5 runs that are cut to length using either ‘butt’ or ‘skive’ joints. When placed within the stuffing box each joint is offset at least 90° from one another to eliminate any potential leak paths. Most gland packing has high resilience and compressibility with low frictional properties. ‘The ideal seal must be soft and conformable to correctly fill the void but resilient enough to withstand the operating pressures, resist extrusion and accommodate shaft movement’ (Flexitallic, 2018). To further reduce the effects of friction, packing is impregnated with oil which is released under the effects of compression to provide built-in lubrication. The current packing used at Vale on conveyors and rotary valves is ‘Flexitallic 310’. This packing ‘is constructed of X-braiding high quality Carbon filament reinforced Graphite tape with a secondary reinforcement using additional strands of Carbon yarn’ (EPDM, 2018). According to its technical datasheet (appendix 2), 310 packing can withstand most chemical media in addition to handling high temperatures and pressures.

Gland Follower

A 'gland follower' consists of a flange and spigot that is used to subject pressure upon the packing to compress the set inside the stuffing box. Typically, to apply force upon the gland follower, gland studs and nuts are used to drive the component forward. At Vale, tightening of the gland is usually performed using the technician's / operator's 'feel and experience'.

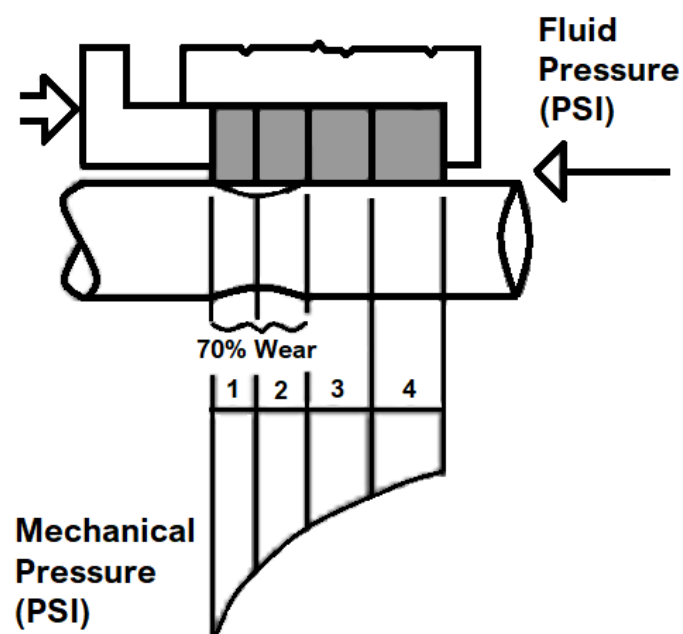
Figure 7 - Gland Follower



Sealing Principle

A 'packing gland' seal operates on the principle of differential pressure. To successfully create and maintain a seal the mechanical pressure inside the gland housing must exceed the internal system pressure. *'The mechanical pressures consist of the gland follower pressure applied axially to the packing rings, which, in reaction, expands the cross-section radially and generates pressure against the bore and the shaft'* (Fluid Sealing Association, 2018). If the radial pressure exerted by the packing were to fall below the pressure of the media being sealed, a leak would occur. The radial pressure profile of a typical packing gland seal is shown below in figure 8. As a result of axial friction, the radial pressure across the packing set does not evenly distribute causing a gradual (pressure) loss from one packing to the next. Due to high levels of compression at the front of the seal, 70% of shaft wear occurs on the first two packing rings. This suggests that the sealing principle has conflicting requirements whereby high mechanical pressures are required to create a seal but low levels of friction are required not to excessively wear the shaft.

Figure 8 - Radial Pressure Profile of a Packing Gland



(Phelps, 2020)

Standards

According to Fluid Sealing (2019) *‘valves account for more than 51 percent of all fugitive emissions, including greenhouse gas. Eighty percent of these emissions come from the valve stem’* whereby the predominant seal of choice is a packing gland. *‘Fugitive emissions are accidental emissions of vapours or gases from pressurised apparatus, either due to faulty equipment, leakage or other unforeseen mishaps’* (Environmental Technology, 2015). In order to reduce fugitive emissions, various governmental and industrial bodies have established the following standards to assess and regulate the performance of packing;

- API 622
- API 624
- ISO 15848 (Parts 1 & 2)
- TA-Luft
- VDI 2440

At Vale there are currently no standards in place for the selection of packing used on plant equipment such as conveyors and rotary valves. The current packing (Flexitallic 310) utilised is non-approved with no accreditation to the standards above.

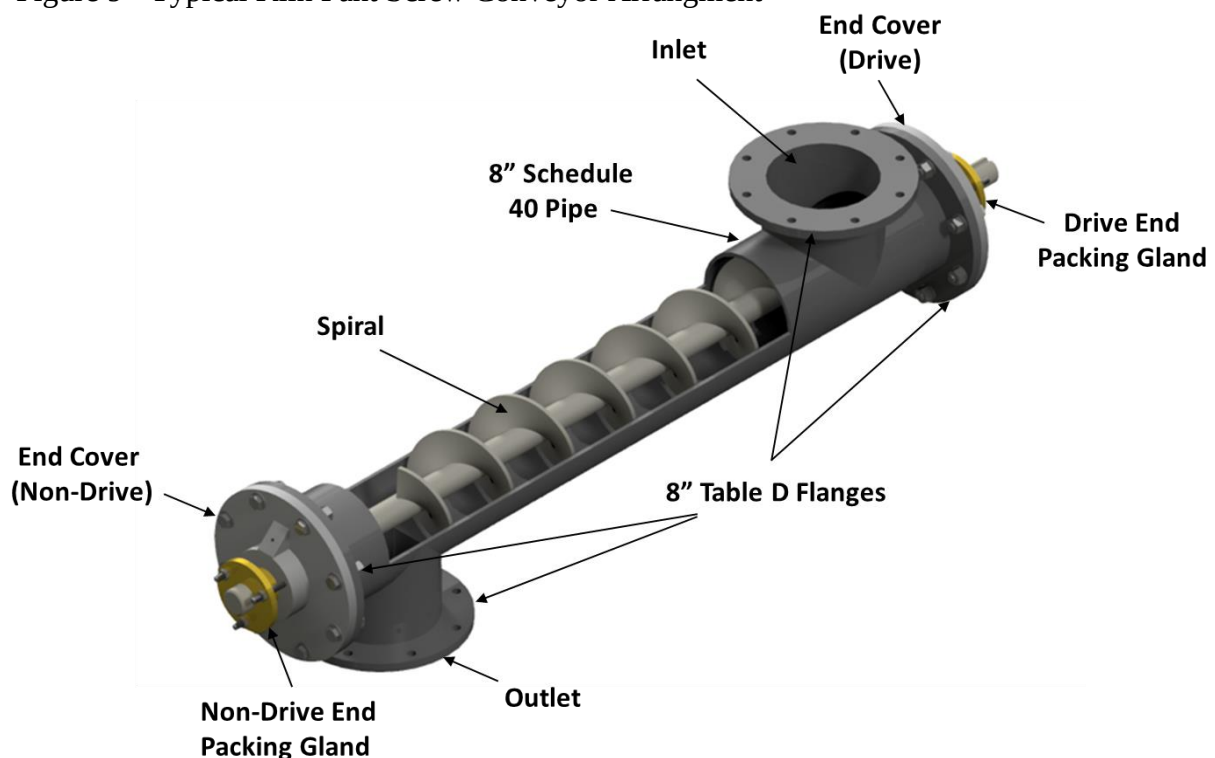
Chapter 3 - Kiln Plant Packing Glands

Within the Kiln plant, ‘packing glands’ are primarily used for sealing the (rotary) shaft movements of ‘screw conveyors’ and ‘rotary valves’.

Screw Conveyor

A screw conveyor or auger conveyor ‘is a mechanism that uses a rotating helical screw, called “flighting”, that is installed within a tube or trough, to move liquid or granular materials’ (Velodyne, 2016). As shown in figure 9 the shell of a typical Kiln plant screw conveyor is fabricated using 8” (shedule 40) pipe and 8” Table ‘D’ flanges. End covers are mounted on the drive and non-drive end to hold the spiral concentric within the pipe. To seal the protrusion of the shaft at each end a packing gland is utilised.

Figure 9 - Typical Kiln Pant Screw Conveyor Arrangment



All conveyor end covers are standard and their stuffing boxes have a 2 ¾” diameter that accepts a 2” shaft and 3/8” packing.

$$\text{Packing Size} = \frac{\text{Stuffing Box Diameter} - \text{Shaft Diameter}}{2}$$

$$\text{Packing Size} = \frac{2\frac{3}{4} - 2}{2}$$

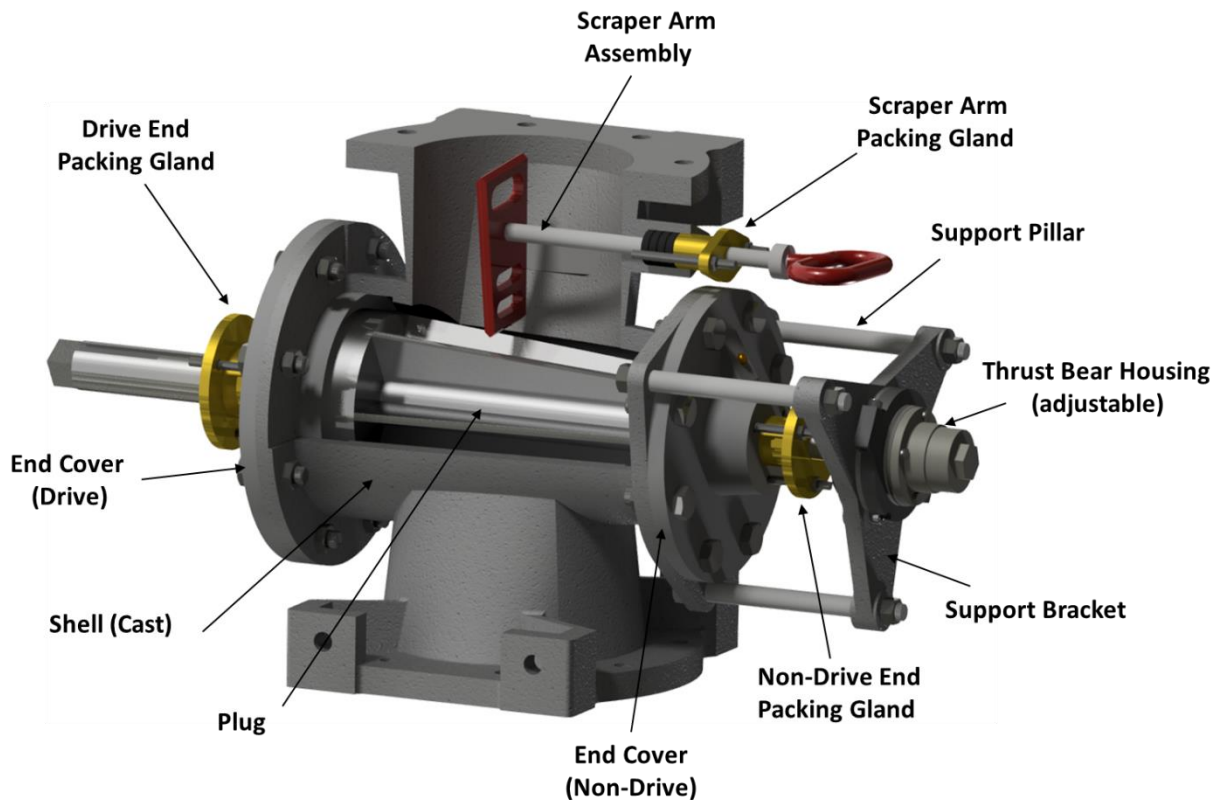
$$\text{Packing Size} = \frac{3}{8}$$

In addition to transferring nickel oxide feed through the extraction process, the large majority of conveyors (and rotary valves) also contain process and product gases such as Hydrogen, Carbon Monoxide and Nickel Carbonyl.

Rotary Valve

‘The prime function of a rotary valve is to regulate the flow from one chamber to another while maintaining a good airlock condition’ (Rotolok, 2020). As shown below in figure 10, the rotary valves used within the Kiln plant, each have three packing gland seals. Similarly to conveyors there are glands on both the drive and non-drive end covers plus an additional seal on the reciprocating scraper arm component.

Figure 10 - Typical Kiln Plant Rotary Valve Arrangement



The drive end stuffing box on a rotary valve has a 2 ½” diameter that accepts a 1 ¾” shaft and 3/8” packing.

$$\begin{aligned}\text{Drive End Packing Size} &= \frac{\text{Stuffing Box Diameter} - \text{Shaft Diameter}}{2} \\ \text{Drive End Packing Size} &= \frac{2\frac{1}{2} - 1\frac{3}{4}}{2} \\ \text{Drive End Packing Size} &= \frac{3}{8}\end{aligned}$$

The non-drive end stuffing box has a 1 ¾” diameter that accepts a 1” shaft and 3/8” packing.

$$\text{Non-Drive End Packing Size} = \frac{\text{Stuffing Box Diameter} - \text{Shaft Diameter}}{2}$$

$$\text{Non-Drive End Packing Size} = \frac{1\frac{3}{4} - 1}{2}$$

$$\text{Non-Drive End Packing Size} = \frac{3}{8}$$

The scraper arm stuffing box has a 1 ¼” diameter that accepts a 5/8” shaft and 5/16” packing.

$$\text{Scraper Arm Packing Size} = \frac{\text{Stuffing Box Diameter} - \text{Shaft Diameter}}{2}$$

$$\text{Scraper Arm Packing Size} = \frac{1\frac{1}{4} - \frac{5}{8}}{2}$$

$$\text{Scraper Arm Packing Size} = \frac{5}{16}$$

From extracting asset information from the Refinery’s computerised maintenance management system (SAP ERP) (appendix 3), it was found there are 63 rotary valves and 57 conveyors within the Kiln plant. Therefore, on the basis that conveyors have two packing gland seals and rotary valves have three, there are in total 303 seals in use.

Operating Pressure

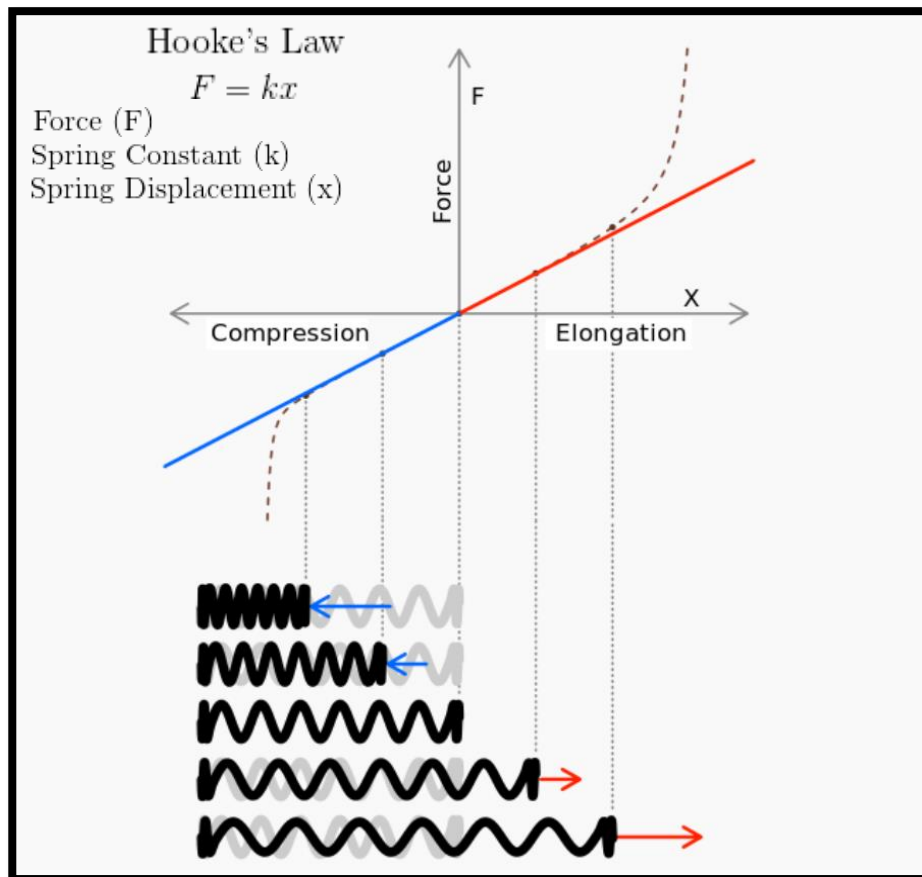
The operating pressure of both components is determined by the plant operating circuit pressure. The Kiln Plant overview manual states the extraction process has a design pressure of 200mm H₂O and operates at a pressure of 160mm H₂O (appendix 4). According to the American Society of Mechanical Engineers (2001) code of practice for pressure piping (ASME B31.1), when performing a pressure test ‘*the pneumatic test pressure shall not be less than 1.2 nor more than 1.5 times the design pressure of the piping system*’. Therefore, to ensure the integrity of all circuit components after a shutdown, the Kiln circuit is pneumatically pressure tested at 300mm H₂O as per standard operating procedure PSO203 (appendix 5).

Chapter 4 - Research into Continuous Force Mechanisms

Mechanical Springs

A mechanical spring is an elastic body usually constructed from spring steel *‘that changes its shape in response to an external force, returning to its original shape when the force is removed. The energy expended in deforming the spring is stored in it and can be recovered when the spring returns to its original shape’* (Kokuyo Spring, 2018). Most springs are of helical form and obey ‘Hooke’s law’ when operating within their elastic limit. Meaning, force (F) is directly proportional to displacement (x) as shown in figure 11.

Figure 11 - Hooke’s Law of Helical Spring



(Word Press, 2017)

There are four main classification of helical springs which all vary in design and application.

1. Compression
2. Extension
3. Torsion
4. Spiral

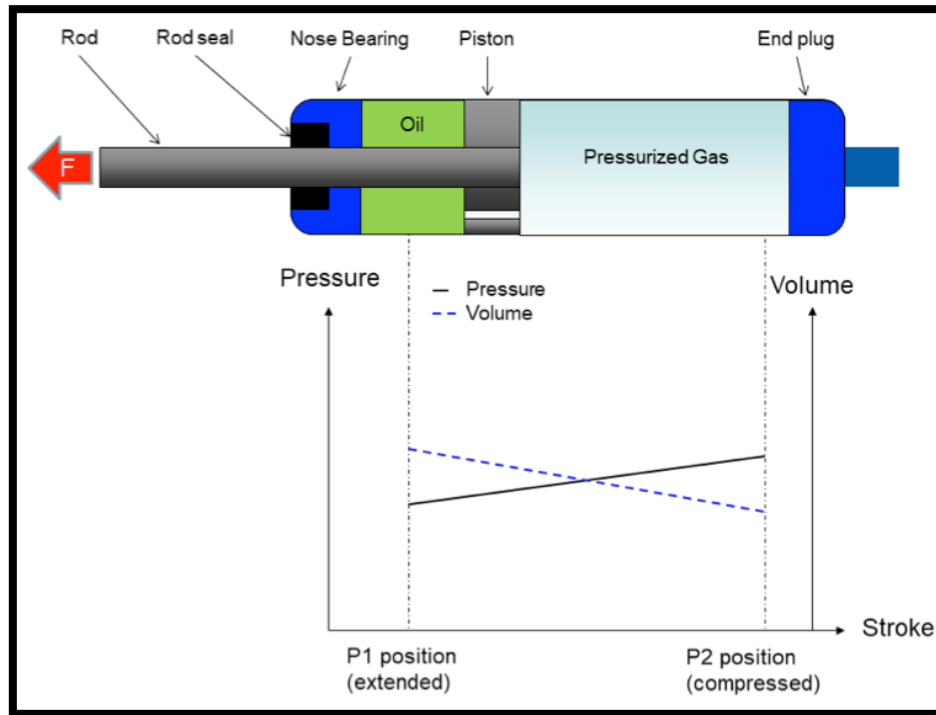
Each spring type has a variety of functions but most are primarily used to;

- Absorb energy
- Apply and maintain a force or torque
- Store energy
- Indicate or control load or torque

Gas Springs

Similarly to mechanical springs, gas springs are designed to store energy. As shown in figure 12 ‘gas springs use a compressed gas that is contained in a cylinder. There is a piston at the top of the spring that exerts a force via the compressed gas’ (European Springs, 2017).

Figure 12 - Function of a Gas Spring



(Industrial Gas Springs, 2020)

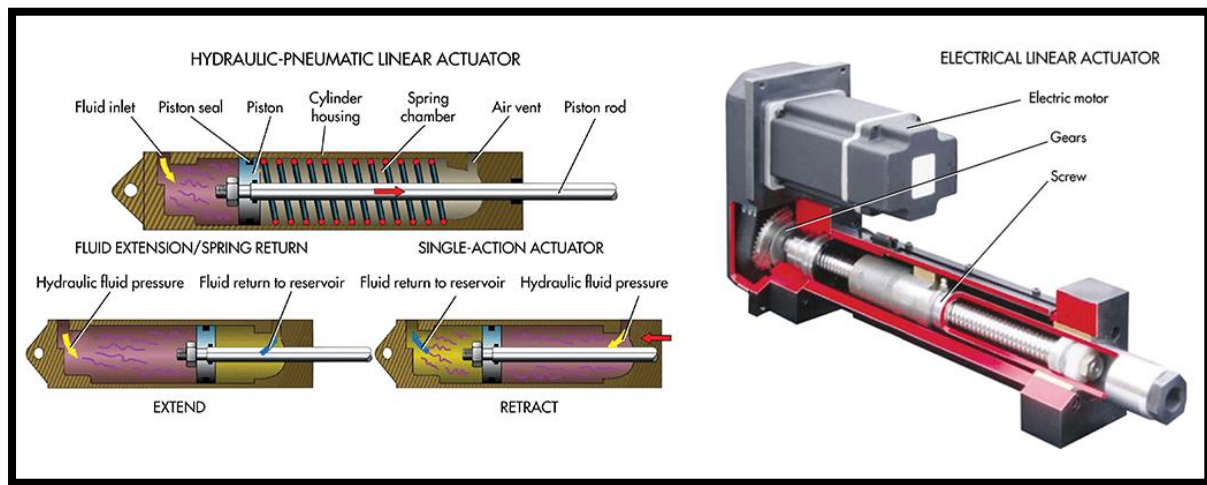
Gas springs are typically filled with Nitrogen gas and operate on the principle of Boyle's Law, whereby pressure is inversely proportional to volume. The (initial) pressure inside the cylinder can be set to apply a specific pre-loaded force upon the piston. Thus enabling the spring to instantly subject a large force from the beginning of its stroke. Some gas spring designs allow the metering and passing of Nitrogen gas into the rod side of the cylinder (as it is compressed) to reduce the effects of force progression. Therefore the spring can exert an output force that is almost constant across its stroke.

Linear Actuators

As its name suggests, a linear actuator is a device that converts energy into straight line movements. Typically, in industry, there are three main energy sources or power transmissions used (hydraulic, pneumatic or electro-mechanical) to operate an actuator. As shown in figure 13, the design concept of hydraulic and pneumatic actuators ‘is very simple; fluid is pumped into the cylinder, and it pushes against the piston, causing it to extend. The velocity of extension is a function of the flow, and the force developed is a function of the pressure’ (Cundiff, 2001). Single acting actuators can only perform (pressure driven) movement in one direction and typically use a spring to return the piston rod to its starting position. Double acting actuators

however, can perform push and pull movements using inlet and outlet valves either side of the piston.

Figure 13 - Operating Principle of Hydraulic, Pneumatic & Electro-Mechanical Actuators



(Machine Design, 2015)

Whereas hydraulic and pneumatic designs actuators inherently produce linear motion, electro-mechanical designs do not. They require mechanical linkage (gears) to transform rotary motion from a motor into linear displacement via a lead screw. The advantages and disadvantages of using hydraulic, pneumatic and electro-mechanical actuators are shown in table 1.

Table 1 - Advantages and Disadvantages of Different Actuators

Hydraulic Actuator		Pneumatic Actuator		Electro-Mechanical Actuator	
Advantages	Disadvantages	Advantages	Disadvantages	Advantages	Disadvantages
Robust and suitable for high-force applications	Can leak	Cheapest form of transmission	Cannot produce large forces	Very good positional control	Cannot be used in hazardous environments (unless deemed intrinsically safe)
Due to incompressibility of fluid, actuator can hold force constant	Requires large infrastructure	Very quick response time	Poor positional control	Environmentally friendly	Requires mechanical linkage (drive gears)
No mechanical linkage required	Expensive	Intrinsically safe	Moisture penetration can cause damage to device.	Can be integrated with most electronic control systems	Drive motor can overheat if continually operating
Good positional control		No mechanical linkage required	Can leak		

Chapter 5 – What is the Required Gland Load to Create an Effective Seal ?

Gland Load

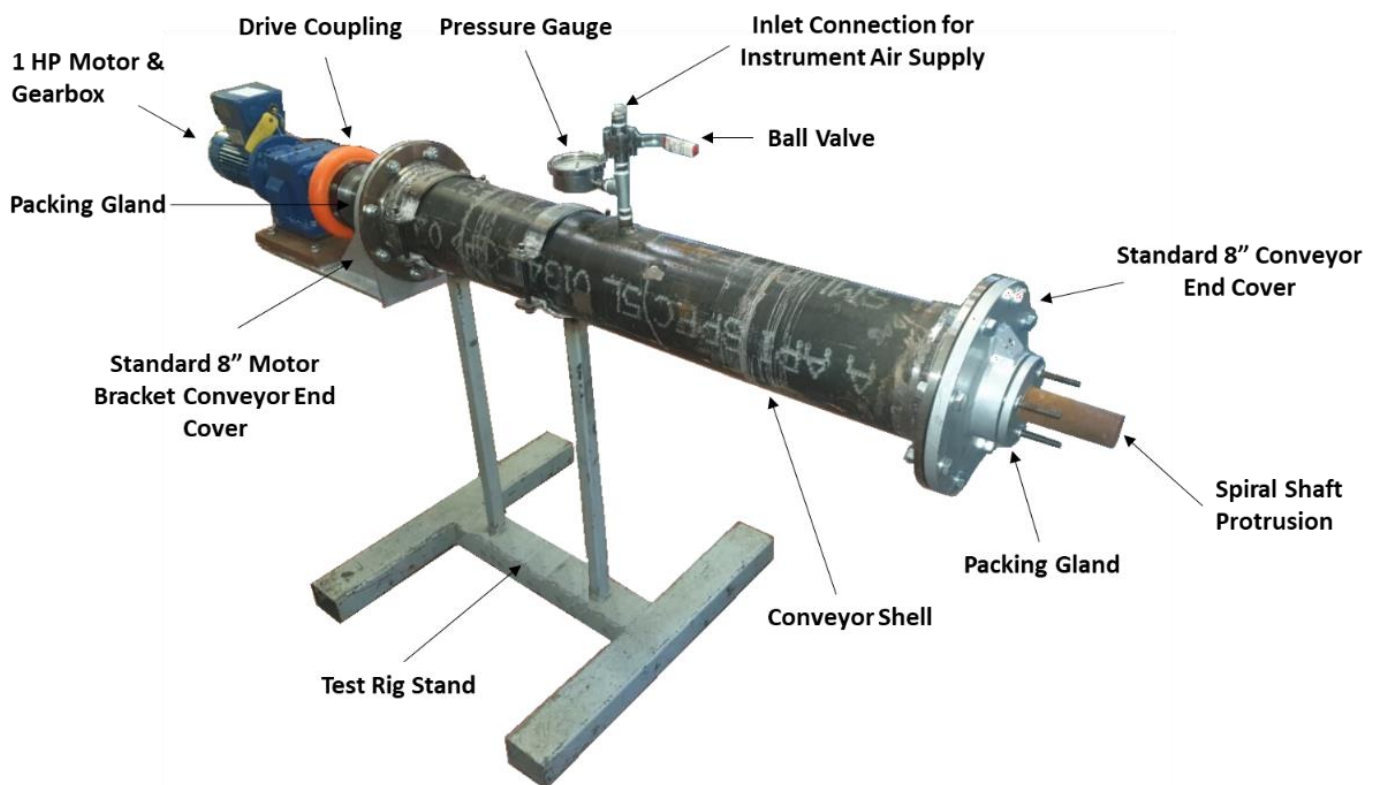
To determine the gland load, Vale's document management systems were interrogated in an attempt to find technical information pertaining to the gland force requirements of conveyors and rotary valves. Disappointingly however, due to the (design) age of the components no such information existed. The only documents found were manually drafted drawings from the late 1960's that specified the dimensional data of both items. Based on these findings, it was concluded that experimental testing was required to determine the required gland load.

Experimental Testing

In order to perform experimental testing a 'conveyor test rig' was assembled as shown in figure 14. The rig mainly consisted of;

- 1 x 8" conveyor shell with a stabbing point midway along its length
- 1 x standard 8" end cover
- 1 x motor bracket end cover
- 1 x 1 HP motor and gearbox
- 1 x test rig stand
- 1 x ball valve
- 1 x pressure gauge

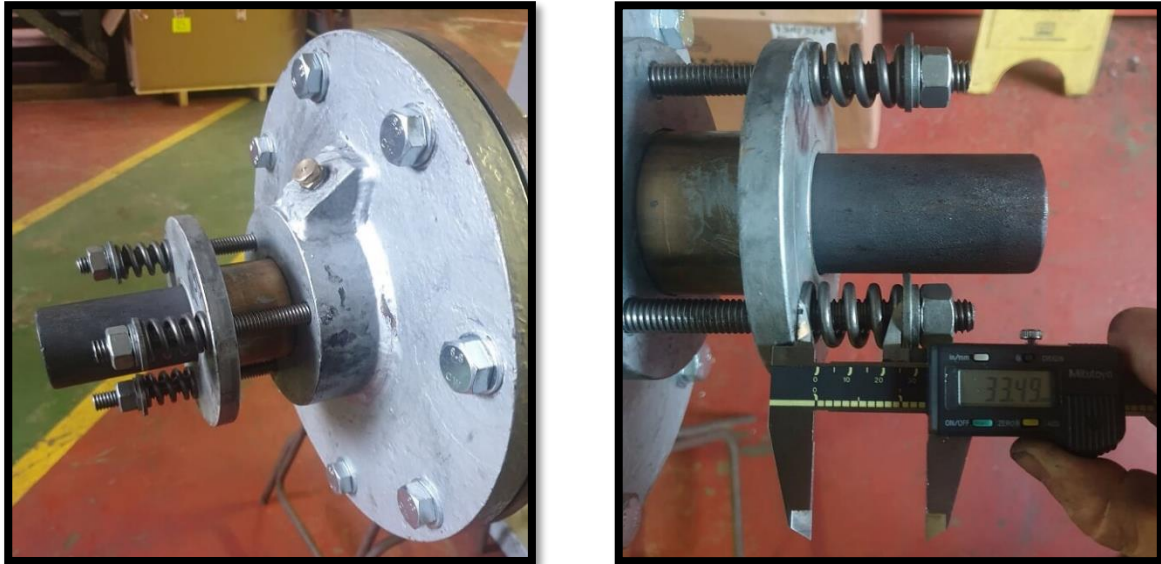
Figure 14 - Conveyor Test Rig Assembly



The assembly enabled the function and pressure conditions of a Kiln plant conveyor to be simulated without the presence of nickel oxide granules. By pressurising the conveyor shell to

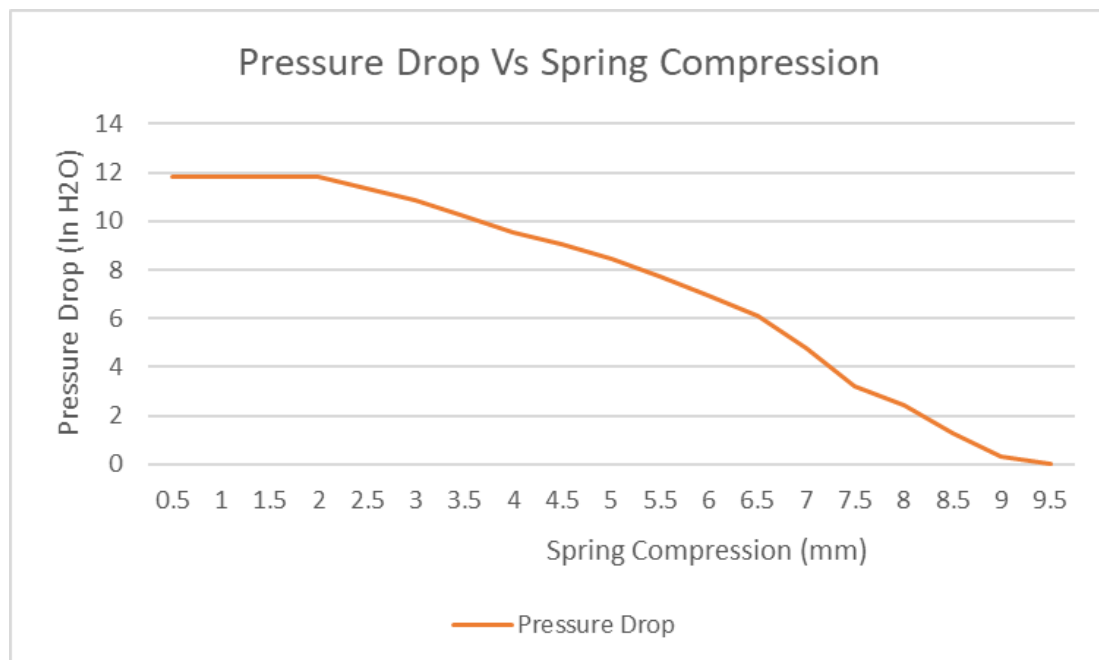
300ml H₂O and performing multiple ‘fall tests’ the packed end covers can have their gland followers incrementally tightened until a seal is achieved. By applying compression springs (with a spring rate of 93.069 n/mm – appendix 6) behind each gland, the applied force can be measured as shown in figure 15.

Figure 15 - Image of Gland with Springs Applied



From following the procedure detailed in appendix 7 the results of the experiment are shown in graph 1.

Graph 1 - Results of Pressure Drop Experiment



The results indicated that a spring compression of 9.5mm was enough to hold a pressure of 11.8 inches water gauge (300 mm water gauge).

Based on a spring compression of 9.5mm, the total gland force was calculated as;

Total Gland Force = Spring Rate * Spring Compression * Number of Springs

Total Gland Force = $93.069 * 9.5 * 3$

Total Gland Force = 2652.47 N (2dp)

This value will be taken as the minimum force required to create a seal.

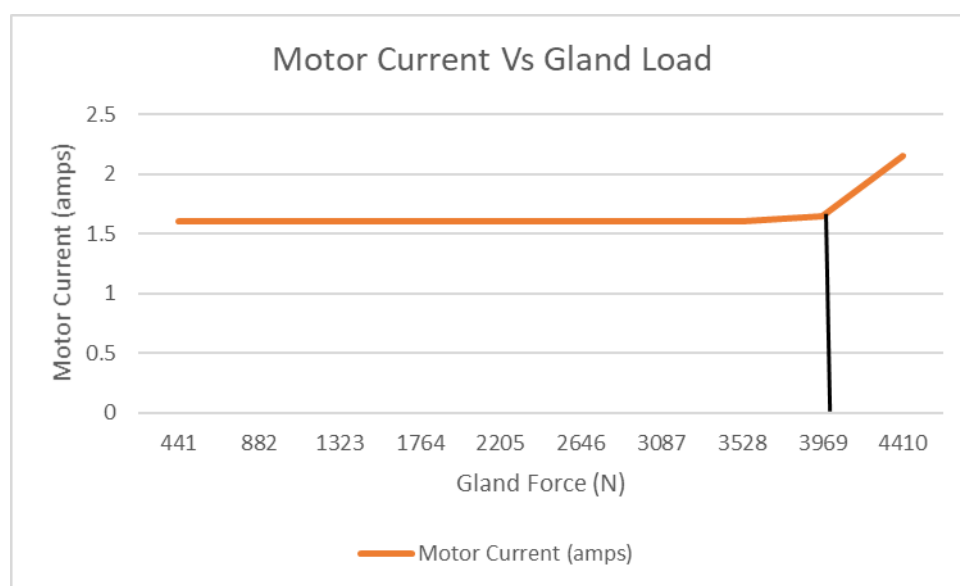
To determine the impact of further tightening the gland, additional testing was required. Prior research has indicated, increasing the gland load can cause increased power consumption on the motor as the packing hugs the shaft more tightly. In an attempt to gauge the frictional effects of the packing (on the shaft) when tightening a gland follower, another experiment was required using the conveyor test rig. By introducing an ammeter to the circuitry of the motor, its current could be measured when increasing the gland load.

Figure 16 - Ammeter Added to the Motor Circuitry of Conveyor Test Rig



From following the procedure detailed in appendix 8 the results displayed in graph 2 were retrieved.

Graph 2 - Effects of Further Increasing Gland Load on Motor



The data confirms that, as the gland load is increased it consequently causes the current to increase, indicating the motor must work harder to overcome the increased friction applied to the shaft. The increased gland load however does not significantly impact the motor amps until approximately 3969 Newtons. Based on these findings, it is apparent a conveyor packing gland will maintain a seal with no adverse effects as long as the gland force applied is between 2652.47 Newton and 3969 Newtons.

Chapter 6 - Project Specification

The next step in the design process is to write a Product Design Specification. This details and provides a list of requirements that the success of the design can be compared to. According to Stuart Pugh a PDS “acts as the mantle or cloak that envelopes all the subsequent stages in the design core” (Pugh, 1990)

Design Brief

To improve the reliability of packing gland seals within the Kiln plant, an improved method of applying a continuous force upon a gland follower is required. A suitable design must have the ability to be retrofitted on current conveyor assemblies.

Product Design Specification

Performance

- The device must be able to subject a continuous force over a prolonged stroke.
- The device must be able to withstand and subject a force between 884 Newtons to 1323 Newtons.
- The device must give indication when force is below minimum requirement.
- The device must be able to be retrofitted to current conveyor and rotary valves glands.
- The device must be interchangeable whilst the process is running.
- The device is to be easily installed.
- The device is to be easily adjusted.
- The device is to be easily maintained.

Size

- Due to the proximity of other components, upon installation, the device body at back of gland follower must not be wider than 40mm nor longer than 30mm.

Environment

- The temperature of the environment will likely range between -5°C to 50°C.
- The device must operate in a dirty industrial environment.
- The device must withstand mild vibrations.

Product Life Expectancy

- Product should have a life span of 3 years.

Life in Service

- The device should be inspected every 6 months.
- No special tools should be required for maintenance.
- The device must be robust enough to take impact or shock loadings.

Safety

- Pinch points should be kept to a minimum.
- The device must have no sharp corners.

Ergonomics

- Operators must receive training before operating the device.
- One person should be able to install / adjust the device.

Quality and Reliability

- Quality should be such that the device should not generally fail within a period of three years and only 1 in 50 should fail within the first year.

Cost

- The cost of purchasing, manufacturing or assembling the device must not exceed £300.

Disposal

- The device must not contain any environmentally hazardous materials.
- The device must be easily disassembled for recycling.

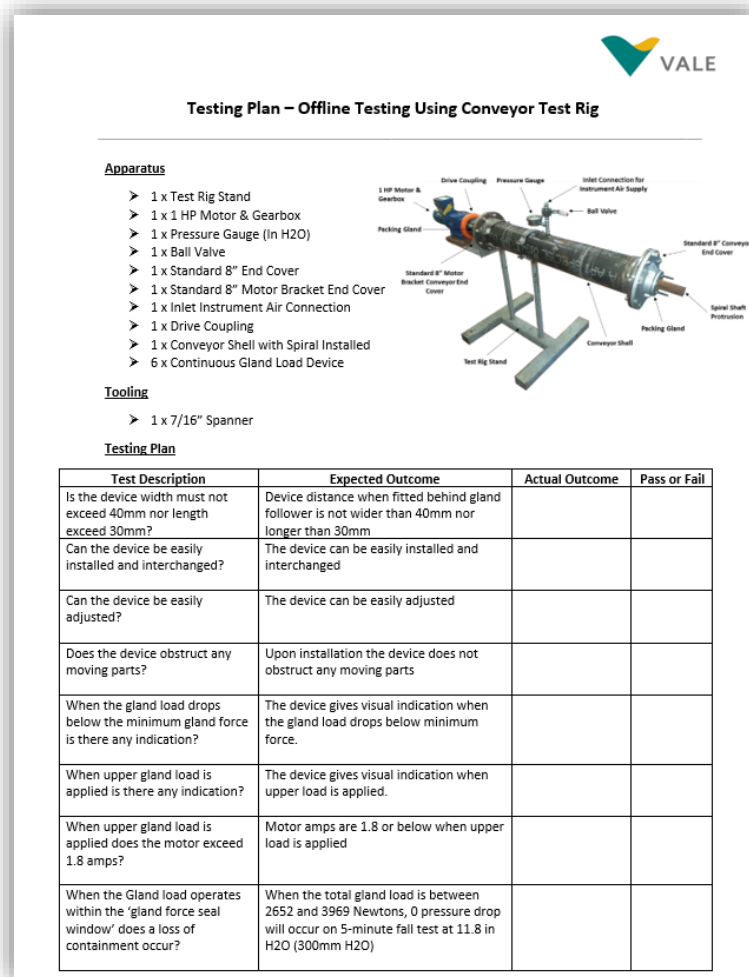
Testing

- The device should be functionally tested to ensure it can fulfil its purpose.

Testing Plan

To ensure the new design functions correctly and meets the project deliverables, a testing plan is required. The tests will be performed offline using a 'conveyor test rig' with a witness present. The plan shown in figure 17 must be passed in order to deem the product suitable for online trials.

Figure 17 - Testing Plan



Budget

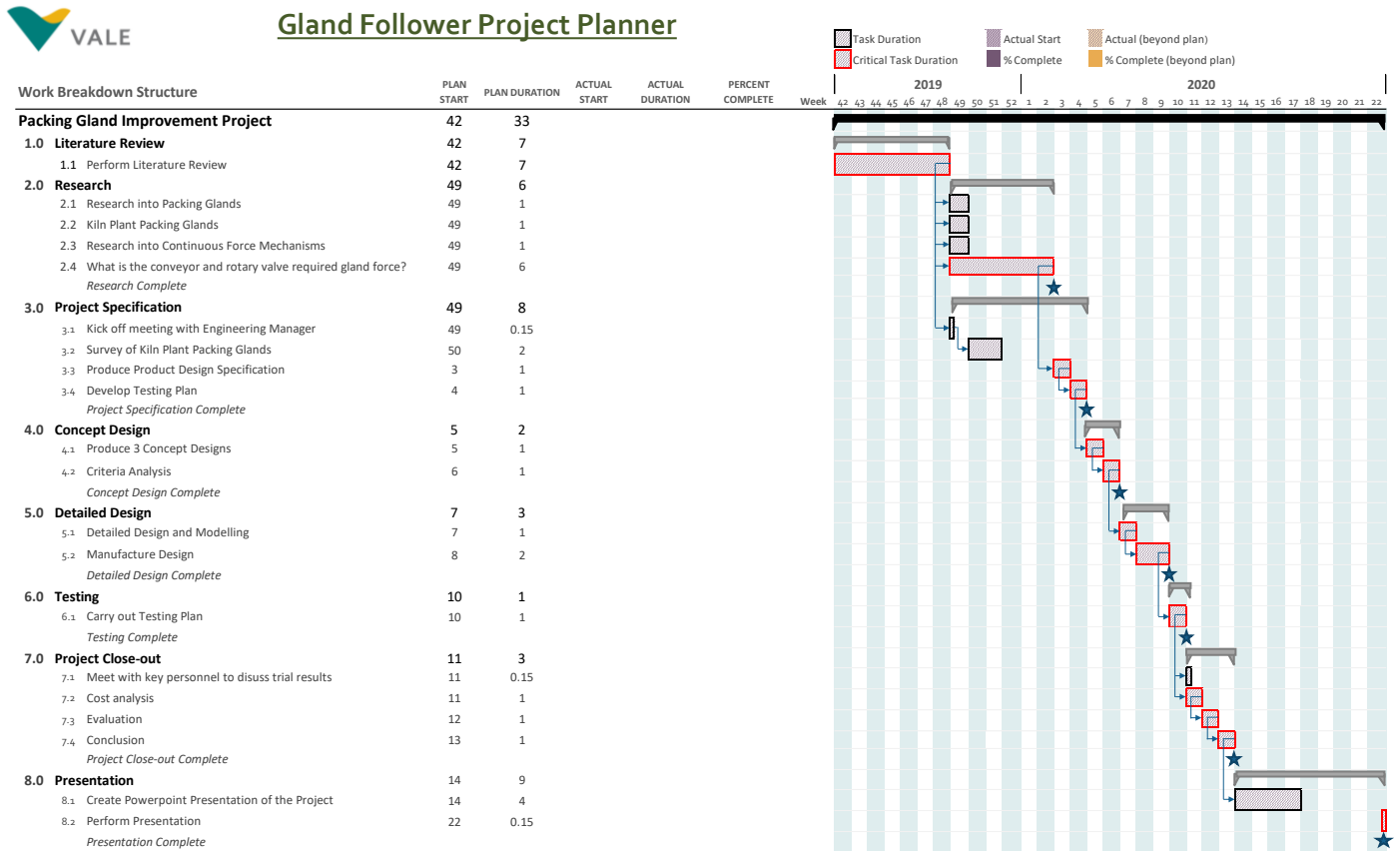
To determine the budget of the project, a ‘kick-off’ meeting was called with the Project Engineering Manager (appendix 9). From discussing the findings of the literature review it was agreed a budget of £2000 would be allowed to develop a new design with proof of concept. The timeframe for the completion of the project was confirmed as 3 months whereby a prototype design must be manufactured and tested to demonstrate the improved sealing performance of the gland.

Chapter 7 – Project Planning

Project Planning

To ensure the project is completed within the set timeframe and to highlight any potential problems with the workload and timescale, a timing plan was developed as shown in figure 18.

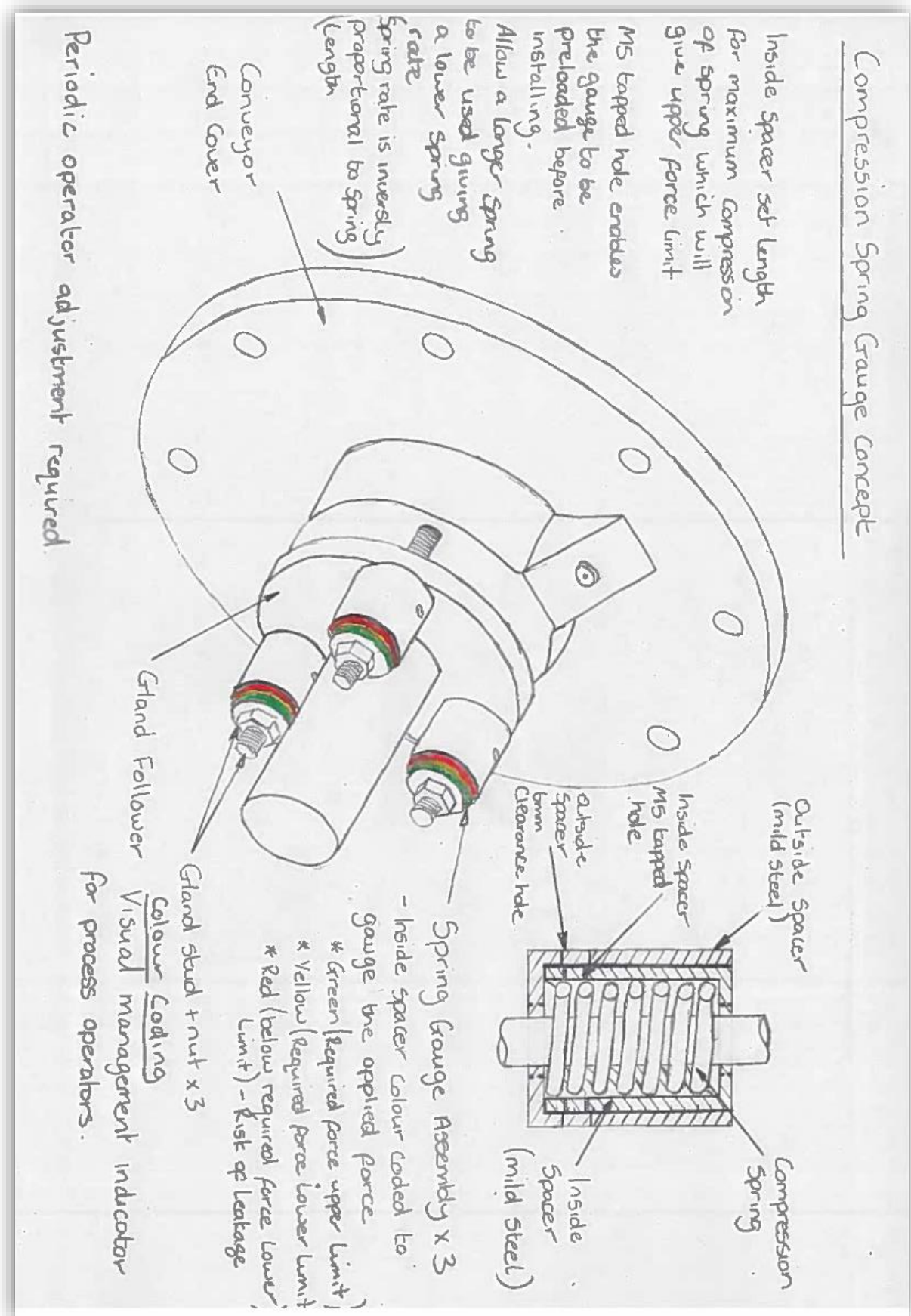
Figure 18 - Timing Plan



The work breakdown structure (WBS) of the plan includes the main elements of the design cycle and the tasks involved to execute each stage. Critical tasks are highlighted to establish the critical path in which the project must try to follow. Each design element has a milestone whereby the proposed schedule is to be reviewed to monitor the progress of the project. If any critical tasks are delayed, the review provides an opportunity to re-evaluate the schedule and determine if extra resources are required to bring the schedule back into alignment. Due to the non-interdependent nature of non-critical tasks an element of 'float' can be tolerated, meaning, any minor delays (in execution) will not have a significant impact on the next milestone or expected project deadline.

Chapter 8 – Concept Designs & Criteria Analysis

Figure 19 - Concept Design 1



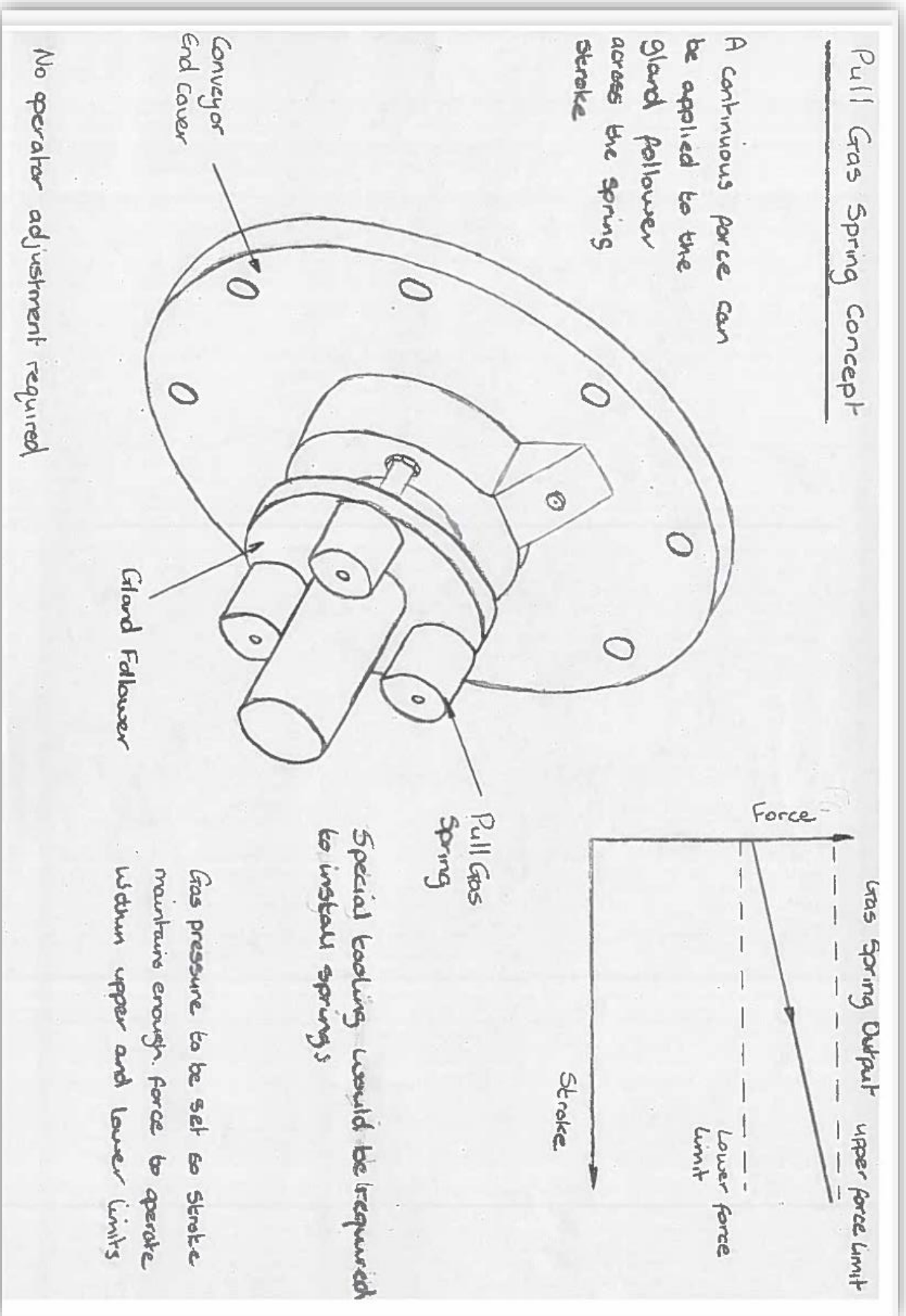
Proposal 1 - Evaluation

Concept design 1 is a spring gauge proposal that consists of a compression spring and two socket shaped spacers. When assembled the compression spring sits inside the cavity created by the two spacers. When the spacers are compressed together the spring deflects and exerts a force, obeying Hooke's law ($F=kx$). When the socket spacers are fully compressed they will 'bottom out' and the spring cannot be deflected any further. At this point the compression spring will exert a force close to the upper seal force limit. Using tapped holes (M5) on the inside spacer, bolts can be applied to lock and preload the device. The advantage of preloading means a longer spring with lower spring rate can be used therefore maximising the springs deflection so the applied load stays within the seal force window over a longer stroke. Also the device can be fitted whilst locked at its minimum length to ensure it does not interfere with any other components in close proximity. When the devices are fitted behind the gland follower, a standard gland nut will be screwed onto the gland studs until they touch each device. The locking (preloading) bolts can then be removed, transferring the load upon the gland follower. When the effects of stress relaxation and wear take effect over time, the inside socket will gradually move out of the outside socket indicating the spring load loss. From colour coding the (outside) surface of the inside socket, the load loss can be put into zones using a traffic light system. This will provide visual management of the condition of the applied load and determine when adjustment is required.

Table 2 - Advantages and Disadvantages of Proposal 1

<u>Advantages</u>	<u>Disadvantages</u>
Simple and robust design	Periodic adjustment required
Minimal maintenance required	Applied load is not constant – over time load will reduce
Can be easily retrofitted on current packing gland assemblies - minimal modification required	Longer gland studs are required
Low cost	
Indication is given when device load is at upper limit	
Indication given when device load is below lower limit	
Compression spring (main loading component) is concealed and protected from dirty industrial environment.	

Figure 20 - Concept Design 2



Proposal 2 - Evaluation

Concept design 2 involves the use of (pull) gas springs that are positioned behind the gland follower with their rods directly connected to the end cover boss replacing the gland stud and nut arrangement. The pressure inside each gas cylinder is to be set so the spring force exerted operates within the upper and lower gland load limits. The stroke of the spring would be limited due to the close proximity of other components and consequently the gland follower spigot would require modification (to be shorter in length). Thus reducing the available gland adjustment and seal life span. In order to extend the spring fully whilst installing the device, special tooling would be required. Due to the use of pressurised gas inside the cylinder body, the performance of the spring could be affected by the transfer of heat.

Table 3 - Advantages and Disadvantages of Proposal 2

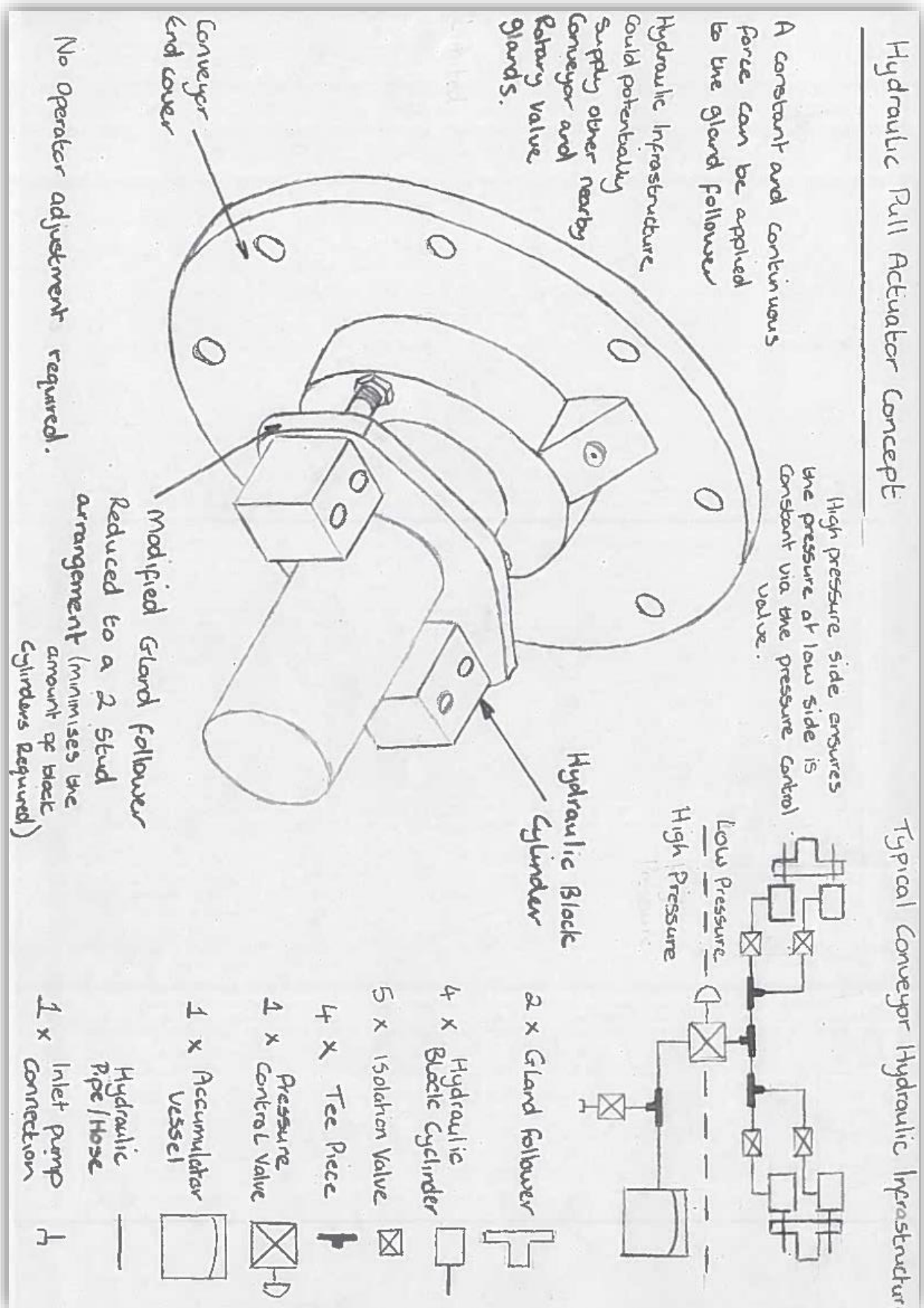
<u>Advantages</u>	<u>Disadvantages</u>
Simple design	More expensive than spring gauge concept
Minimal maintenance required	Special tooling required to install
No operator adjustment required	Heat can affect spring performance
Minimal modification required to install	Reduced life span of seal - available gland adjustment reduced

Figure 21 - DADCO Gas Springs



(GW Technologies, 2020)

Figure 22 - Concept design 3



Proposal 3 - Evaluation

Concept design 3 involves the use of hydraulic pull actuators that are positioned behind the gland follower with their rods directly connected to the end cover boss, again replacing the gland stud and nut arrangement. The design requires a relatively large hydraulic infrastructure that would contain high and low pressure. The low hydraulic pressure would supply the gland actuators. The high pressure would supply the low pressure via a control valve to ensure pressure loss does not occur when there is gland movement. An accumulator would ensure there was enough volume of high pressure to maintain a pressure differential between both sides. Potentially the hydraulic circuit could feed a number of packing glands that are in close proximity. Such an infrastructure would require frequent maintenance to ensure the integrity of the system is maintained. To reduce the number of actuators and cost of the design proposal it is deemed necessary to modify the gland follower from a 3 hole arrangement to a 2 hole arrangement. Similarly to the gas spring concept, the stroke of the actuator would be limited due to the close proximity of other components and consequently the gland follower spigot would also require modification (to be shorter in length).

Table 4 - Advantages and Disadvantages of Proposal 3

<u>Advantages</u>	<u>Disadvantages</u>
Applies a constant and continuous force	Extensive modifications required to install
No operator adjustment required	High Expense
	Large infrastructure required
	High level of maintenance required

Figure 23 - Hydraulic block actuator



(Vektex, 2020)

Criteria Analysis

To assist in the decision making process of selecting the most suitable design, a weighted table was constructed as shown in Table 5. The design parameters chosen were mainly elements from the PDS whereby their importance (weighting) were given a value between 1 – 4.

Table 5 - Criteria Analysis Results

Criteria Analysis - Weighted Table				
Design Parameter	Weighting	Design 1	Design 2	Design 3
Cost of Manufacture	4	9	6	3
Maintenance Required	2	8	8	4
Durability	2	8	6	7
Reliability	3	7	7	7
Performance	3	6	8	8
Operator Intervention	3	3	10	10
Ease of Installation	2	8	2	6
Infrastructure Required	2	9	9	3
Aesthetics	1	6	6	4
Safety	2	8	6	4
		172	167	139

From scoring each design from 1 to 10. Where 1 is low (poor) and 10 is high (good). Concept design 1 was deemed the most appropriate with a total score of 172. Although designs 2 and 3 were potentially better performers and required no operator intervention, design 1 was a much simpler and cost effective solution. The 'Cost of Manufacture' had the highest weighting of any other parameter due to the limited funds available and consequently affected the design selection process the most.

Chapter 9 –Detailed Design and Implementation

Compression Spring Selection

To maximise the effectiveness of concept design 1, the compression spring selection is very important. If the spring is too stiff, the force loss when stress relaxation occurs will be extreme and the available gauge length will be limited. Alternatively, if the spring rate is too low the spring will be unable to exert enough force within the space available. In an attempt to balance these conflicting requirements, Hooke's Law was analysed;

$$F(\text{spring force}) = k(\text{spring constant}) * x(\text{spring displacement})$$

Transposed for 'k'

$$k = \frac{F}{x}$$

From transposing the formula for 'k' (spring constant), it can be seen the spring rate is inversely proportional to the amount of spring displacement available, meaning a longer spring will exhibit a lower spring rate. This is evident when comparing the two compression springs in figure 24. Both springs are potentially suitable for the design, but the longer spring (B – CM26X38) gives a much lower spring rate whilst exerting a similar (Max. Deflection Load) output.

Figure 24 - Compression Spring Comparison



A - CM26X25

ISO 10243 Colour Coded	Blue
Strength	Medium Duty
Hole Dia (mm)	25
Rod Dia (mm)	12.5
Free length (mm)	25
Rate N/mm	147
Max. Deflection Load	1381.8
Max. Deflection (mm)	9.4



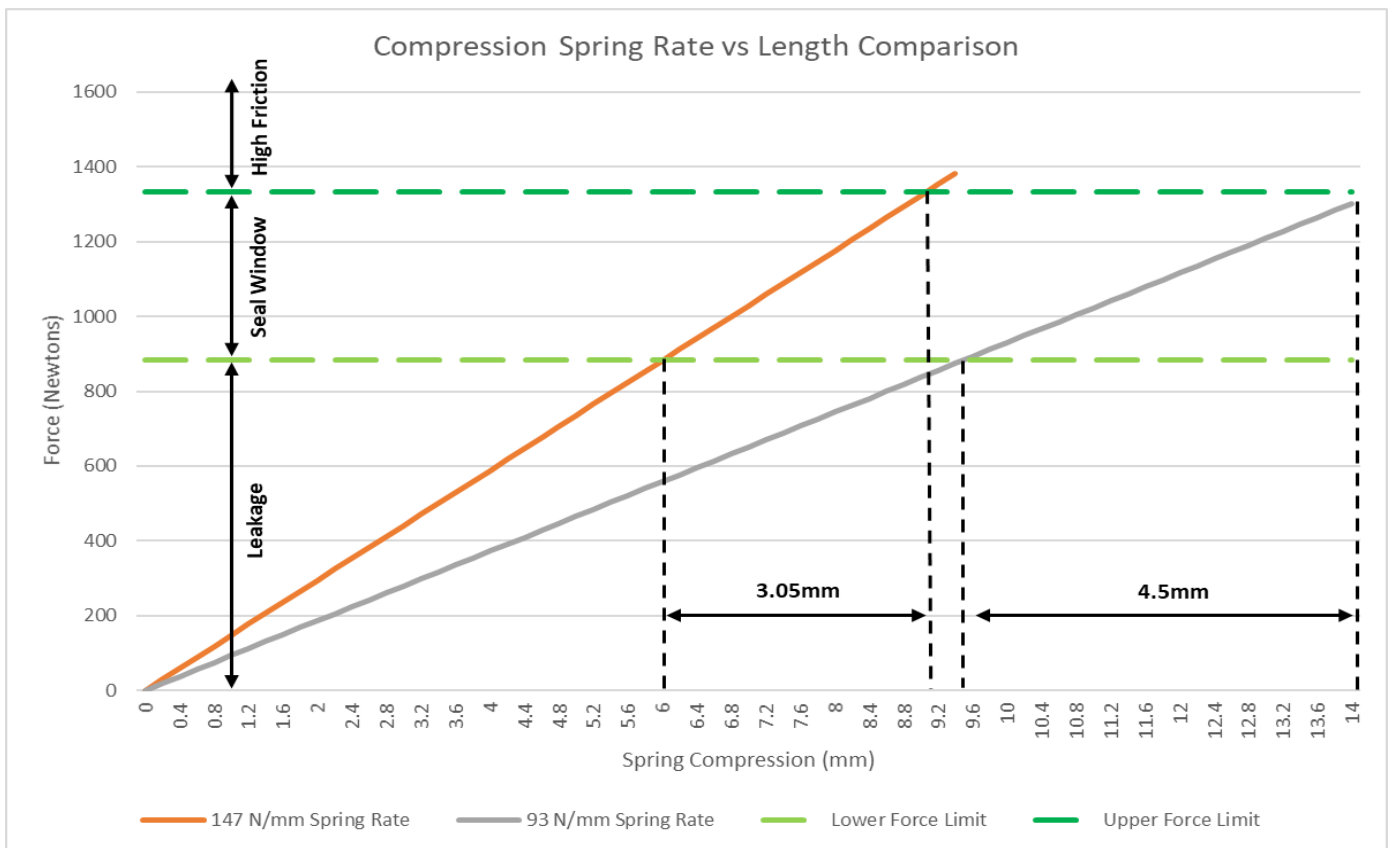
B - CM26X38

ISO 10243 Colour Coded	Blue
Strength	Medium Duty
Hole Dia (mm)	25
Rod Dia (mm)	12.5
Free length (mm)	38
Rate N/mm	93
Max. Deflection Load	1302
Max. Deflection (mm)	14

(Spring Masters, 2020)

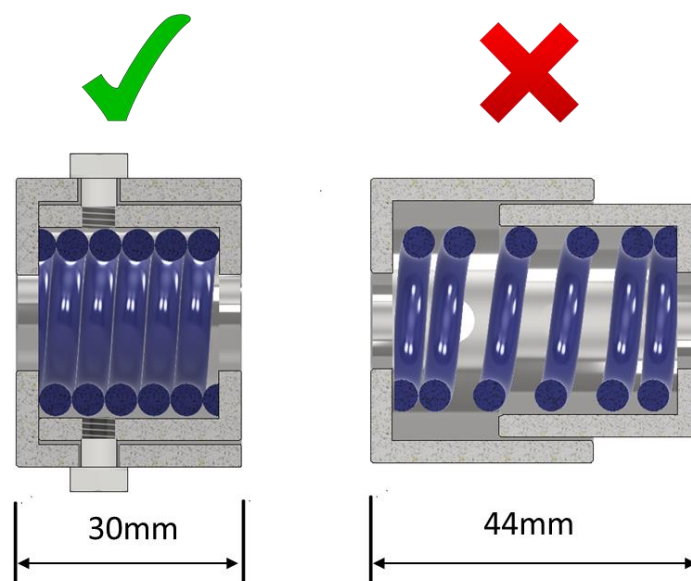
The benefit of using a longer spring is displayed in graph 3 whereby the 93 N/mm spring rate provides a sealing window of 4.5mm in comparison to 3.05mm for the shorter spring (147N/mm spring rate). Maximising the amount of spring displacement across the seal window will reduce the likelihood of leakage caused by stress relaxation and the frequency of 'operator intervention' required.

Graph 3 - Comparison of Spring Rate



The consequence of using option 'B' however will mean the device will exceed a length of 30mm when the spring is at its free length. To ensure the design remains within the constraints of the PDS, the device will need to be locked prior to installation in its closed position as shown in figure 25.

Figure 25 - Benefit of Pre-Loading Device



Model Design

Using Autodesk Inventor, the device components were modelled as shown in figures 26, 27 and 29.

Figure 26 - CM26X38 Compression Spring



Figure 27 - Inner Socket Spacer

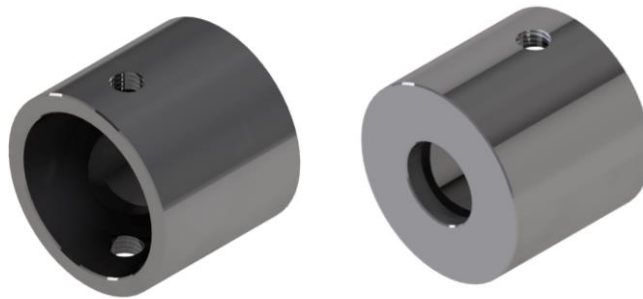
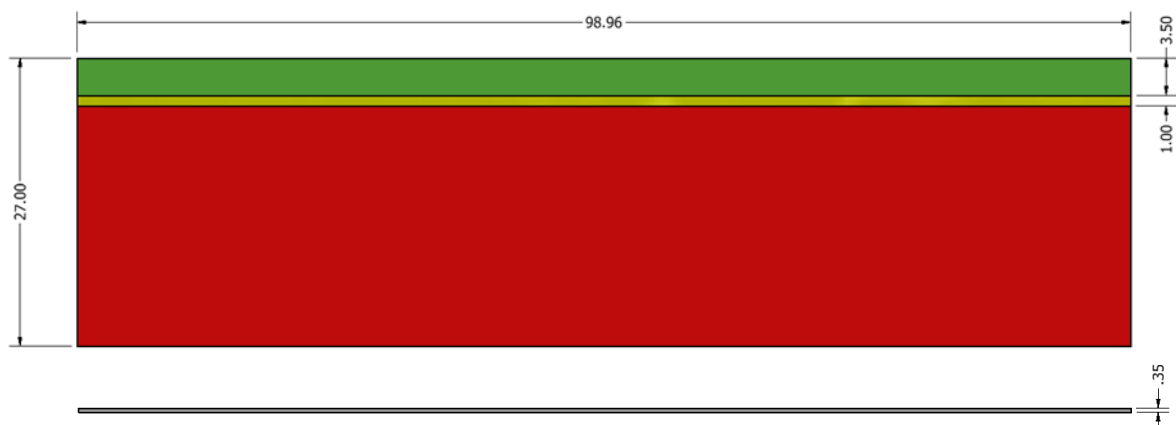


Figure 28 - Outer Socket Spacer



Figure 29 - Gauge Label



FEA Design

To ensure the device can withstand the maximum force exerted by the spring, Finite Element Analysis was performed using the Stress Analysis environment within Autodesk Inventor.

Figure 30 - FEA Results of Mild Steel Inner Socket (3mm wall thickness)

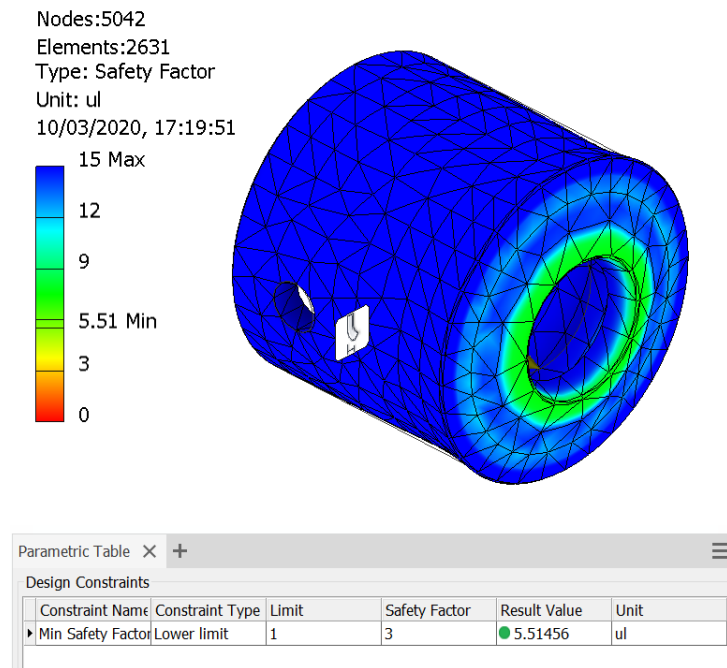
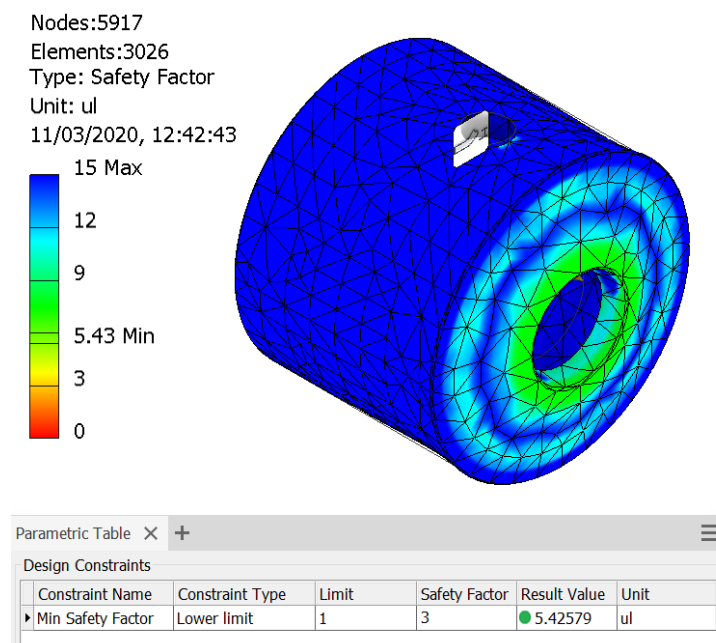


Figure 31 - FEA Results of Mild Steel Outer Socket (3mm wall thickness)

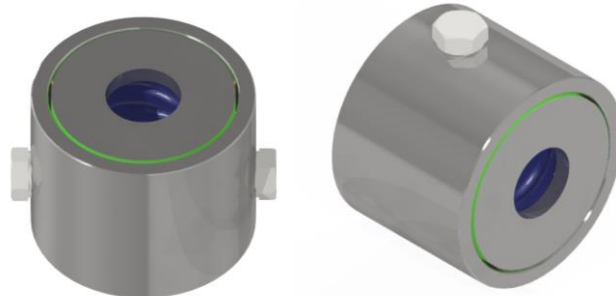


The results produced in figures 30 and 31 show both components (inner and outer sockets) are capable of withstanding a force of 1323 Newtons with a safety factor of 5.51 and 5.43 respectively.

Assembly

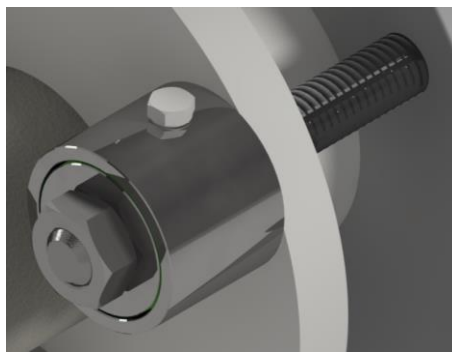
To demonstrate the configuration of the device prior to installation, the components were assembled together as shown in figure 32. From compressing the assembly together, M5 bolts would be used to lock and preload the device in its closed position.

Figure 32 - Device Assembly Prior to Installation



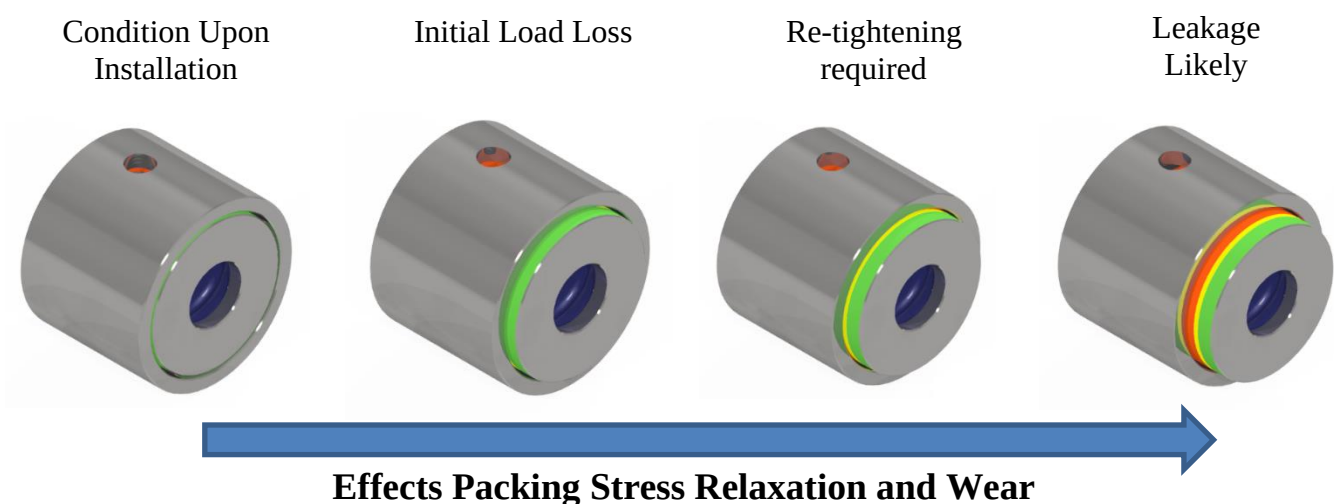
Upon installing the device, the gland nuts are to be tightened until they touch the face of the inner spacer as shown in figure 33.

Figure 33 - Method of Installing Device Assembly



Then, by removing the M5 bolts the preloaded force will be transferred to the gland follower. The expected function of the device whilst in service is shown in figure 34.

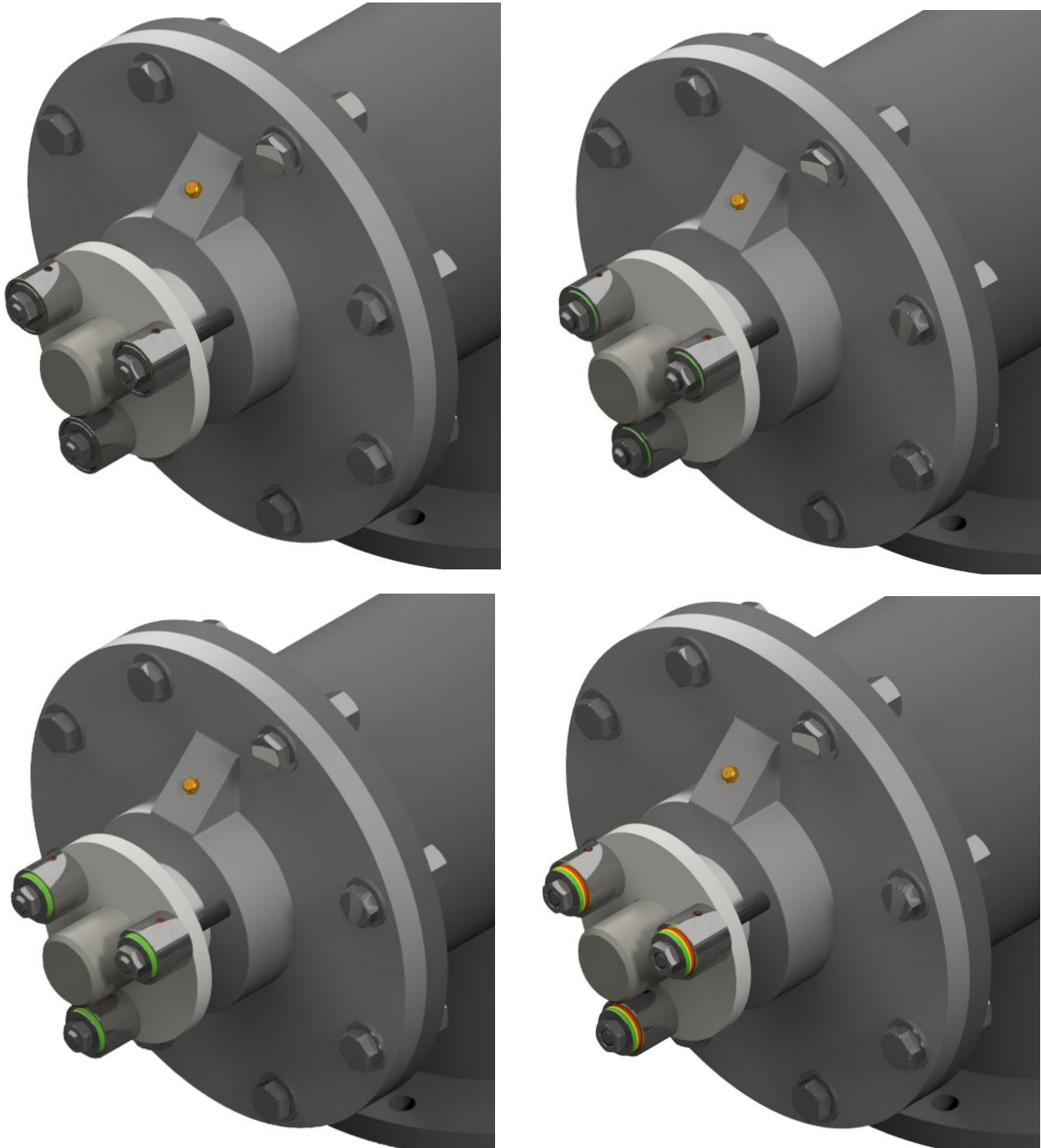
Figure 34 - Device Functionality in Service



Conveyor Assembly

To ensure there are no dimensional issues or obstructions upon installation, the devices were modelled on a conveyor end cover as shown in figure 35.

Figure 35 - Device Modelled on Conveyor End Cover



Detailed Drawings

To manufacture the socket spacer components, the following detailed drawings were created and supplied to Vale's Central Engineering Workshop.

Figure 36 - Detailed Drawing of Outer Socket Spacer (001)

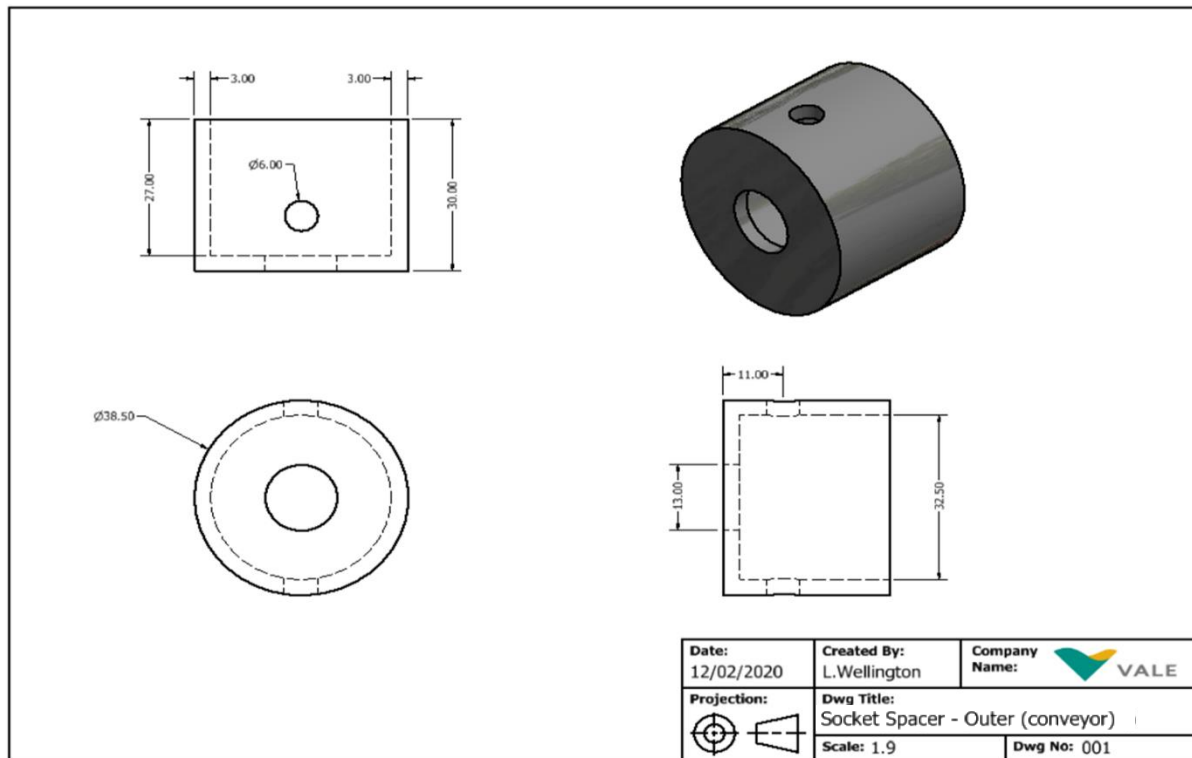
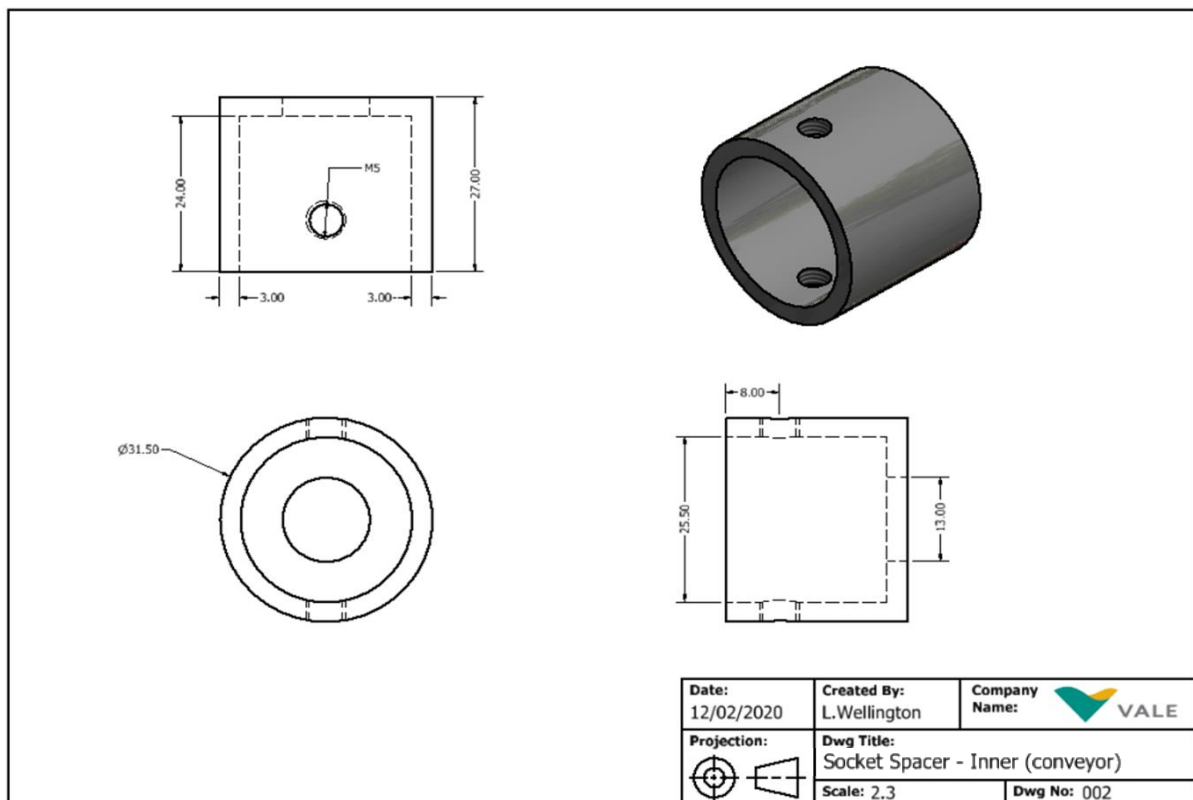


Figure 37 - Detailed Drawing of Inner Socket Spacer (002)



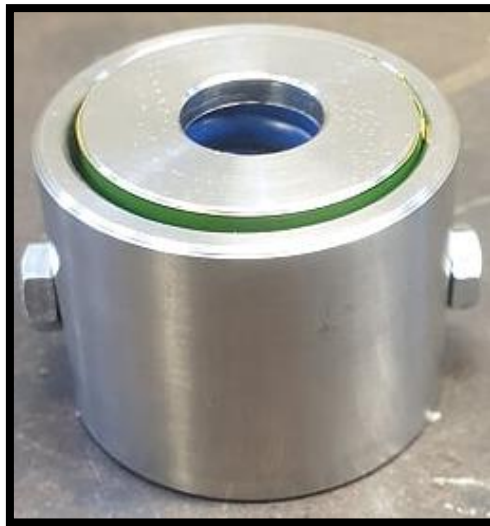
From the detailed drawings produced, 6 prototype socket spring gauges were manufactured as shown in figure 38.

Figure 38 - 6 Manufactured Socket Spring Gauges



Prior to installation, each socket spring gauge was individually locked in its closed position to ensure the overall length did not exceed 30mm.

Figure 39 - Locking Device in Closed Position



To transfer the preloaded force upon the gland follower, the gland nuts are screwed onto the gland studs until they touched the face of the inner spacer.

Figure 40 - Initial Installation of Devices



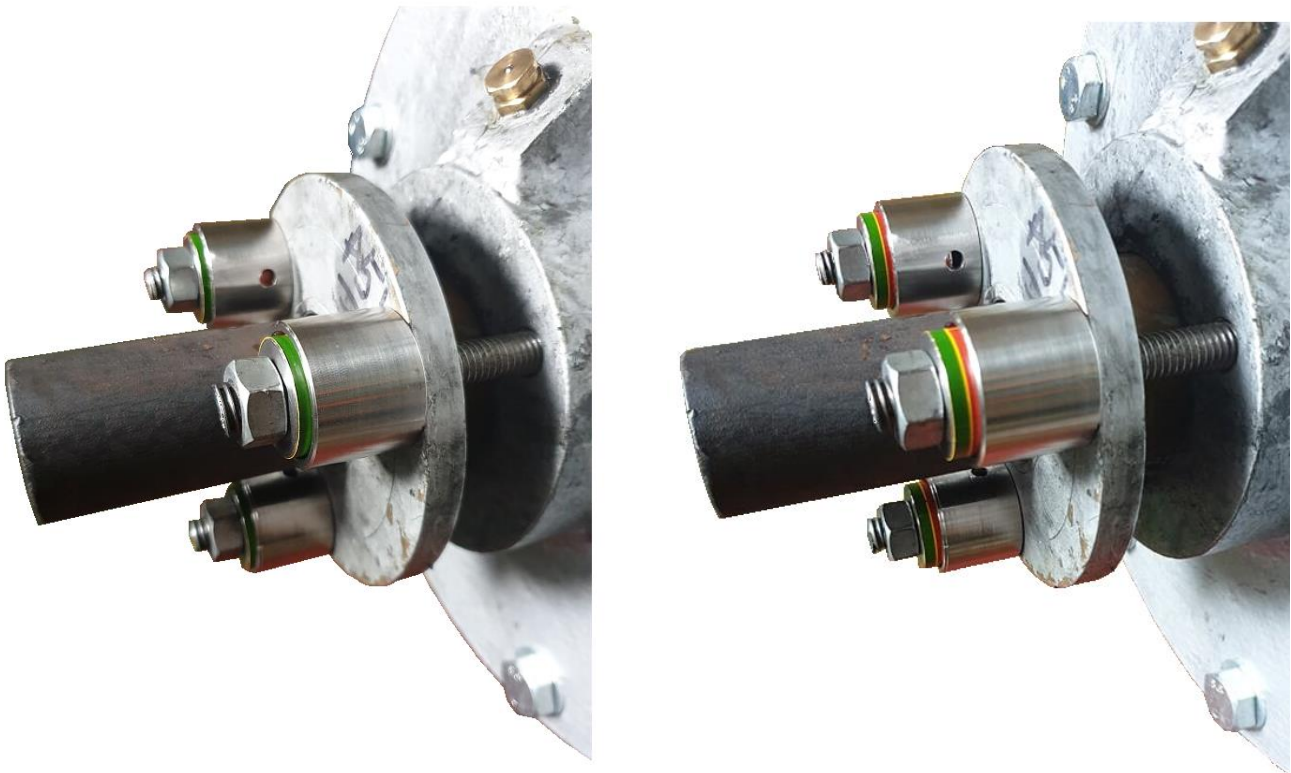
Then, the M5 bolts are removed using an 8mm spanner.

Figure 41 - Removal of M5 Bolts (transferral of load)



Over time, the effects of stress relaxation and packing wear will cause the gland load to lower. To help ensure the device does not drop below the lower gland load limit, indication of load loss is given through (gauged) colour coding of the inner socket (figure 42). Process operators would be expected to inspect the condition of the gland load as part of their shift routines and re-tension if required. The process of tensioning of the gland is simplified and no longer relies on the 'feel or experience' of the operator / technician which invariably causes variation. By returning the device to its closed position the correct gland load can be easily reset by low skilled personnel with minimal training.

Figure 42 - Effects of Gland Load Loss



Chapter 10 - Cost Analysis

Project Cost

To collect the costs associated with the project, a work order was raised within Vale's computerised maintenance management system (SAP ERP). All the required tasks were listed within the work order and assigned to appropriate work groups for planning, scheduling and execution. Additionally, the requisitioning of components and materials was performed via the components tab on the work order. From analysing the cost report of work order 201904541899 (appendix 10), the following expenditure was accrued.

Assembly of test rig

Fabrication of conveyor shell = $\text{£}30 * 7.5 \text{ hours} * 2 \text{ men} = \text{£}450$

Electrical work for connecting up the motor and adding an ammeter = $\text{£}35 * 20 \text{ hours} = \text{£}700$

For the additional components used on the conveyor test rig (motor, spiral, end covers, etc) an agreement was made with Vale's Stores Manager to borrow the parts on a temporary basis until all testing was complete. The components would be then returned to the stores and all costs reimbursed.

Manufacture of 6 Prototype Socket Spring Gauges

Cost of purchasing 6 CM26X38 springs (appendix 11) = **£25.38**

Cost of Manufacturing 12 sockets (6 inner and 6 outer) = $\text{£}35 * 21 \text{ hours} = \text{£}735$

Cost of 6 Labels (labour to produce on label maker) = $\text{£}35 * 1 \text{ hour} = \text{£}35$

The overall project cost was calculated as **£1,945.38**, £54.62 below the agreed budget of £2,000.

Budget Cost Proposal for Implementation

To calculate a budget cost for implementing the new proposal on conveyors and rotary valves within the Kiln plant, it is assumed the manufacturing costs for producing conveyor socket spring gauges will be same for rotary valves. Although pressure testing is required to confirm the gland force requirements on rotary valves, their fundamental design is not expected to drastically vary in terms of cost.

Therefore, based on;

- 63 rotary valves that each require 8 socket spring gauges.
- 57 conveyors that each require 6 socket spring gauges.

A total of 846 devices are required.

Cost of Springs

Unit cost $\text{£}1.82 * 846 = \text{£}1,539.72$

Cost of Socket Spacers (inner with gauge label & outer)

Due to the volume required, a quotation request was sent to one of Vale's preferred contractors Ben Hughes (appendix 12)

Unit cost of £55.54 * 2 * 846 = **£93,973.68**

Cost of Replacing Gland Studs

Unit cost of stud £1.10 * 846 = **£930.60**

Cost of Installation

Hourly Rate of £30 * (10 / 60) * 846 = **£4,230.00** (estimated 10 minutes to install)

Total Budget Cost = **£100,674.00**

Payback Duration

Based on data retrieved from Vale's SAP IM (Incident Management) reporting system (appendix 13) in the past 2 years, there has been on average of 39 plant evacuations caused by leaking packing glands per year. A typical plant evacuation takes a minimum of 2 hours to resolve, whereby plant operators must don (bottled) breathing apparatus, re-enter the plant to find and resolve the cause of the plant issue. Additionally, during normal operating conditions, on average there can be between 10 to 30 workers within the Kiln plant at any one time. To establish an average hourly labour cost of the main disciplines working within the Kiln Plant, Vale's computerised maintenance management system (SAP ERP) was interrogated. From the data retrieved (appendix 14) an average rate of £33.70 was calculated.

Therefore; based on the following;

- 39 evacuations per year.
- 20 plant workers.
- 2 hours waiting for problem to be resolved.
- Average hourly rate of £33.70.

Total Productivity Cost Per Year = 39*20*2*33.70

Total Productivity Cost Per Year = **£52,572.00**

It is estimated that £52,572 per year is being wasted through loss of worker productivity. This cost analysis does not consider potential environmental penalties that are associated with high containment losses, nor does it consider the substantial risk to plant production. If a leak cannot be found and plant issues arise, the plant Superintendent would be forced into activating emergency shutdown valves whereby the loss of production is estimated at £25,000 per hour per kiln line.

However, based on labour productivity losses alone the project proposal has an estimated payback period of 1.91 years (£100,674 / £52,572).

Chapter 11 – Evaluation of New Design

Evaluation

To help evaluate the performance of the design improvement, the testing plan established within the project specification was executed. All testing elements were carried out (offline) on a conveyor test rig and witnessed by G.Havard (Systems Engineer). The results of the test (appendix 15) deem the device suitable for online trials as all elements were successfully 'passed'. A number of benefits were demonstrated whilst performing the testing plan such as;

1. Standardisation of Gland Tightening

Currently, the method of tightening a gland is dependent on an operator's feel and experience which is prone to inconsistency and difficult to standardise. However, from introducing the new design and encapsulating a spring inside two alternated sockets the same gland load can be repeatedly set with minimal training needed. As the device can only be compressed a limited amount before 'dead ending', the 'inner' socket fully inserts into the 'outer' socket visually indicating the maximum force has been set.

2. Visual Management of Gland (load) Condition

Another benefit of the inverted socket arrangement is that when load loss occurs, the 'inner' and 'outer' sockets diverge from one another. Therefore, by colour coding and gauging the 'upper and lower seal force limits' onto the outer surface of the 'inner' socket, a visual indication of the load loss can be given. The current method of checking the gland load condition at Vale requires mechanical intervention whereby an operator would apply a spanner to each gland nut to 'feel' the load. Although this may not appear to be a labour-intensive task, the volume and frequency at which it is required is significant to an operator's workload. If the spring devices were installed, then only a visual inspection would be required with adjustment needed when the load loss has reached the lower limit.

3. Continuous Applied Load

Since a spring's primary function is to deflect under load (storing energy), it will inevitably apply a force over a longer distance when compared to a rigid mild steel stud under tension. Although technically a stud when within its elastic zone, deflects and stretches similarly to a spring (under load) it's not its primary function to do so. In an attempted to evaluate the significant improvement of using a spring, the following calculations were performed.

Effective Spring Rate of Gland Stud Under Tension Using Current Method

Effective spring rate (stud) = Force (N) / Change in Length of Stud (mm)

Effective spring rate (stud) = $1302 / 5.87 \times 10^{-3}$

Effective spring rate (stud) = 221,805.8 N/mm

(Calculations for theoretically determining the Change in Length can be found in appendix 16)

Spring Rate of Design Improvement

CM26X38 spring rate = Force / Displacement

CM26X38 spring rate = $1302 / 14$

CM26X38 spring rate = 93 N/mm

As already proven within the spring selection part of the detailed design, the spring rate is key in controlling the load loss when gland movement occurs. Hence, the effects of stress relaxation are more significant when the spring rate is increased.

4. Simple to Retrofit and Interchange

To meet key (performance) requirements of the PDS the design required the ability to be retrofitted and interchangeable on current conveyors and rotary valves. Although longer studs were required upon initial installation, the devices were relatively simple to retrofit and interchange whilst on the run. They were easily adjusted with no obstructions apparent and their diameter did not exceed 40mm nor did their length exceed 30mm.

To ensure the validity of the results retrieved throughout the testing process, a witness was present at all times and the measuring equipment used was in good order and repeatable results were achieved in all the tests. All the benefits listed are believed to be key in negating the effects of stress relaxation and preventing the likelihood of a packing gland leaking. To quantify and better understand the improvements gained, online comparative testing would be beneficial.

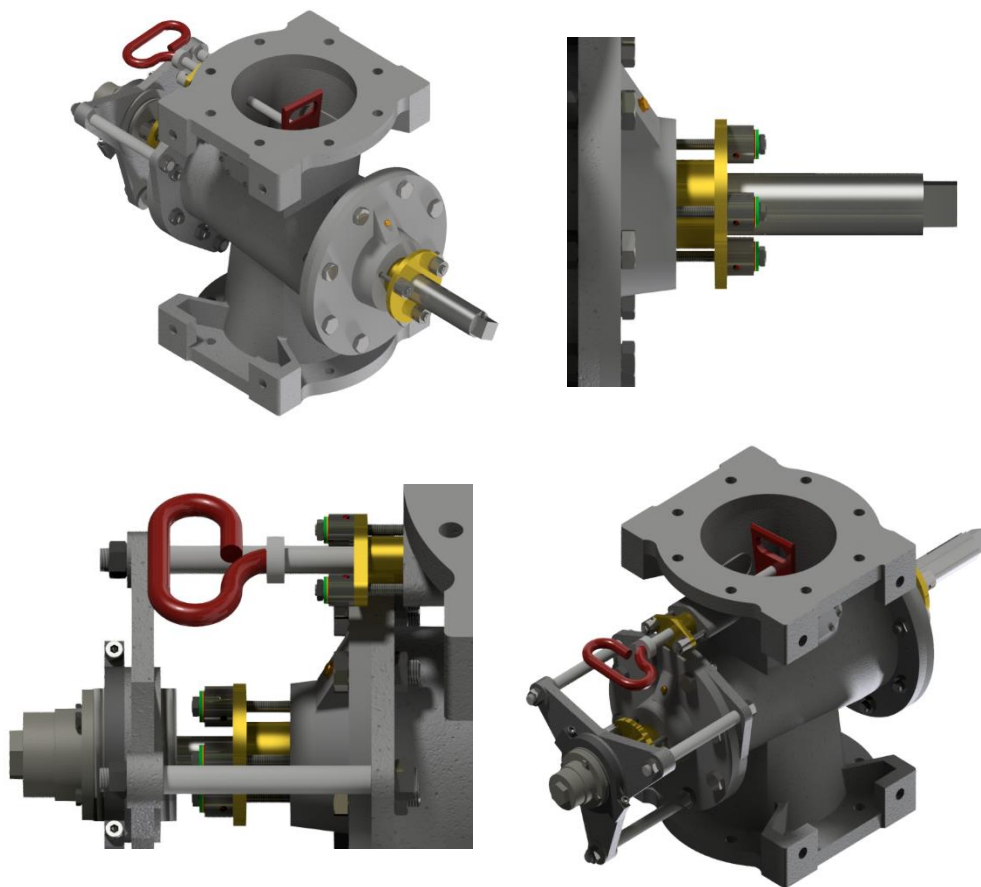
Chapter 12 – Conclusion

Conclusion

From analysing the project outcomes, it is fair to say the proposal is a valid solution to improving the reliability of packing gland seals given the budget provided. Although the solution is relatively basic in design, it has many beneficial features. In addition to applying a continuous force, the design also considers error proofing, standardisation and visual management, fully conforming to the requirements of the PDS. Although some aspects of the design cycle over ran, the use of a timing plan (appendix 17) was essential in planning, scheduling and executing the necessary tasks to ensure the project was completed on time. Unfortunately, offline testing was unable to quantify the amount of improvement gained due to the amount of time it takes for stress relaxation to occur. Although the design improvement is theoretically proven to apply a more continuous load, it is difficult to measure how much the sealing reliability will improve. Comparative online testing would provide a better indication of the improvement gained whereby the performance of two identical conveyors from either Kiln line could be extensively monitored over a prolonged period. Such a trial would allow plant operators to test and evaluate the benefits of managing the new improved solution versus the current method. From conducting a project close out meeting (appendix 18) the proposal of carrying out online testing was approved with the option of requesting capital for 2021 if the results are deemed successful.

For the design proposal to be applied to Rotary Valves, further testing is required to confirm the associated gland load requirements. However, a virtual representation of how the component would look with the design improvement installed can be seen below (figure 43).

Figure 43 - Design Solution Modelled on Rotary Valve



It is strongly believed if the improvement was implemented on conveyor and rotary valves across the plant, it has the potential to significantly reduce gland leakage and subsequently prevent plant evacuations from occurring. Operators would also have the ability to manage their areas more effectively through visual management and less frequent interventions.

Although there is still a lot work and investment required before the design can be rolled out, the feasibility aspect of this project can overall be deemed a success with the deliverables completed on time (before 6th March 2020) and within budget (<£2,000).

Chapter 13 – Recommendations

Recommendations

From performing this project, a number of recommendations are believed to be worth pursuing to further improve the reliability of packing gland seals.

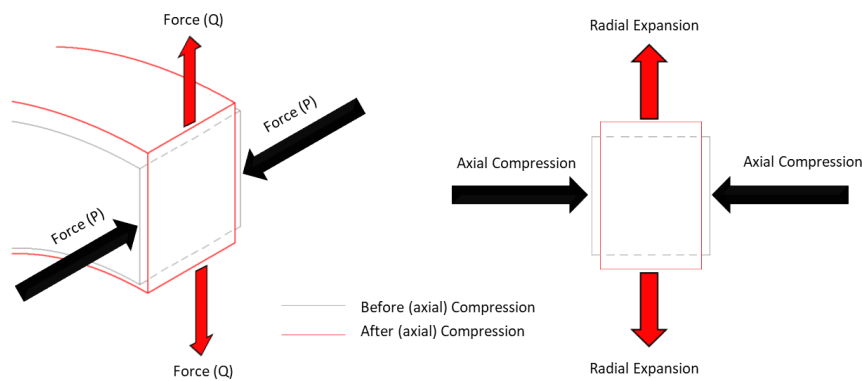
1. Use of Alternate Packing that Conforms to International Standards

Research has revealed the current selection of packing (Flexitallic 310) used at Vale does not conform to any recognised standards. From switching to an accredited product, the plant can have the assurance that the packing performance is of a standard and proven through rigorous testing.

2. Investigate the Possibility of Using Combination Packing

Look into possibility of using packing sets with differing K values in an attempt to level out the (uneven) pressure profile exerted. As shown in figure 44 the amount of radial expansion that occurs when packing is compressed is dependent upon the 'lateral deformation factor' (K)

Figure 44 - Packing Lateral Deformation Factor (K)



Typically, a rigid packing would attain a low 'K' value and a soft packing would have a high 'K' value.

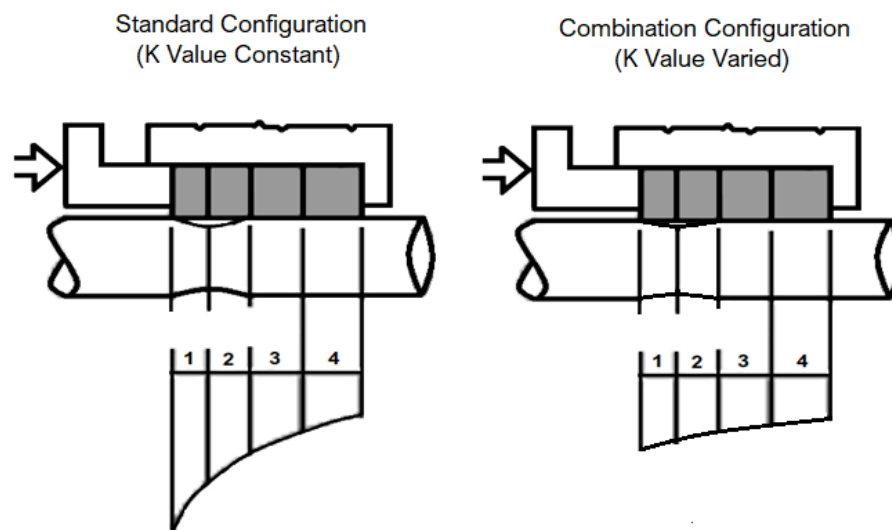
Figure 45 - Rigid Packing Vs Soft Packing



(Gallagher Fluid Seals, 2019)

If a combination is feasible it would create a more uniform sealing profile across the packing and reduce wear over the 1st and 2nd packings.

Figure 46 - Comparison of Sealing Profile



3. Development of a standard work procedure.

If the design solution is mass implemented, then a standard work procedure (SWP) for installing and adjusting a packing gland would be very beneficial. The SWP can then be used for training purposes to ensure the same approach is used by all to ensure consistent results.

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