

Development of an unloading machine for nuclear fuel pebbles

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ABSTRACT

The project goal is to develop a machine that can unload nuclear sphere fuel pebbles in a controlled sequence. The unloading machine will be operational in an environment filled with graphite dust and the gas medium is helium. Furthermore, the environment is radioactive and therefore maintenance activities must be reduced to a minimum.

The unloading machine must fit in the bottom of a fuel storage tank. Access to the tank is only from the top, so as to control radioactive releases. The unloading machine must additionally be capable of unloading usable spheres and separate pieces of spheres to the bottom of the tank.

A scale model was built to confirm the functionality of two unloading principles, gravity unloading and suction unloading. The gravity unloading concept was selected and further developed. Different improvements were made to the original concept used for the scale model. At first sequencing was not achieved, and therefore the concept was improved to separate spheres mechanically. This caused a possibility to damage spheres. The concept was further improved to remove the disadvantage of sphere damage. Spheres were now again unloaded through the centre of rotation of the unloading head.

This mechanism achieved sphere separation, but not sequencing control. Another unloading level was added to the unloading head which proved theoretically to improve sequencing.

At this stage of the design it proved theoretically possible to achieve sequentially controlled unloading of spheres, with separation of pieces of spheres to the bottom of the tank. The tests proved that the concept would work, but another improvement was required to prevent strung unloading of spheres.

The tank unloading machine was not further tested with the required improvement. Therefore the tests should be repeated when the improvement has been incorporated. It is proposed to add bumps in the housing at the level of the second level of the unloading head. These bumps will prevent spheres from rolling towards the exit in the head. The spheres will then maintain their position in the second unloading level until the exit hole will pass the sphere to take it in.

The conclusion is that the concept will probably be acceptable when the latest test was performed. It is however believed that this last improvement will be successful. The tank unloading machine is developed to be low on maintenance. It is believed that the bearings will not need replacement, thus no maintenance is foreseen. The tank unloading device can separate pieces of spheres before the usable spheres are delivered into the sphere transportation pipe.

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ABBREVIATIONS, DEFINITIONS AND ACRONYMS

All Abbreviations and Acronyms applicable to this document are detailed below:

Abbreviation or Acronym	Explanation / Definition
ACU	Air Conveying Unit
AVR	Arbeitsgemeinschaft Versuchsreaktor (Joint venture experimental reactor)
DiD	Defence in Depth
FSF	Fundamental Safety Functions
PBMR	Pebble Bed Modular Reactor
SSS	Sphere Storage System
TUD	Tank Unloading Device

DEFINITIONS

Below the definitions applicable to this document:

Description	Explanation
High-energy storage tank	A storage tank provided with reliable cooling to store high-energy fuel
Low-energy storage tank	A storage tank with low cost cooling to store low energy fuel
Singularization	This is a term used for the action where one sphere is removed into a single state from the bed of spheres.
Singularizer	An item or machine that removes a single sphere from a sphere bed.

LIST OF SYMBOLS

Symbol	Explanation	Unit
α_D	Acceleration of head	rad/s^2
C	Spring index	-
D	Spring nominal diameter	m
d	Spring wire diameter	m
D_{osp}	Head outer diameter	m
D_{sp}	Sphere exit diameter (on the head where the spheres passes through)	m
D_{sh}	Head shaft diameter	m
F_μ	Force due to friction	Nm
J_{sh}	Momentum of shaft inertia	$\text{m}^2 \text{ kg}$
J_{sp}	Momentum of head inertia	$\text{m}^2 \text{ kg}$
J_t	Total moment of inertia of head and shaft	$\text{m}^2 \text{ kg}$
L_{pl}	Plate spring total length	m
L_t	Plate spring effective length (where spring action takes place)	m
m_{sp}	Head mass	kg
m_{sh}	Head shaft mass	kg
p_μ	Pressure on head from the spheres as result of sphere mass on the head in friction calculations	Pa
r	Radius of head used in integration calculation	m
θ_D	Angle of head acceleration	$^\circ$ (deg)
S_{se}	Maximum endurance limit in shear for the spring	MPa
S_{sy}	Torsional yield strength on coil spring	MPa
T_D	Torque required to accelerate head	Nm
T_μ	Torque required to overcome friction	Nm
T_a	Stress amplitude of spring	MPa
T_{max}	Mean stress on spring	MPa
ω_0	Beginning velocity of head	rad/s
ω_{sp}	Running velocity of head	rad/s
W	Weight of object	N
y	Distance from plate spring top, this distance equals the coil spring radius	m

1. INTRODUCTION

1.1 BACKGROUND

The Tank Unloading Device (TUD) is a machine in the Sphere Storage System (SSS) of the Pebble Bed Modular Reactor (PBMR) with the function of unloading spheres from a storage vessel. The SSS interfaces with the fuel handling system, which is responsible for re-circulation of fuel through the reactor, [1]. The fuel is in the form of graphite spheres with a diameter of approximately 60 mm, [1] and [2].

Fuel circulation is required for burn-up measurement of the fuel. When the fuel is burned up, the fuel handling system removes the fuel from the reactor to the sphere storage system and replaces it with fresh fuel from the fresh fuel storage tank, [1]. The fuel is transported through a pipeline by means of a combination of gravity and pneumatic forces where helium is used as the transportation medium.

Burned-up fuel that is removed from the reactor is moved to the high energy storage section of the sphere storage system. It is cooled there till the heat energy is low enough to move the fuel over to the low energy storage part of the sphere storage system.

When the reactor is unloaded for maintenance, the partially burned-up or used fuel is unloaded to the high energy section of the sphere storage system for storage. Graphite spheres are then unloaded from the graphite tank into the reactor. After the maintenance is completed the fuel spheres are returned from the SSS to the fuel handling system for refuelling of the reactor. The refuelling process is to be done within a sphere circulation rate of approximately 242 spheres per hour (normal operation), [1].

At the end of the plant life of 40 years, the spent fuel is stored for another 40 years before final unloading will take place, moving the spent fuel to a waste handling facility, [1]. This unloading will also be done by the fuel unloading machine.

Redistribution of fuel within the SSS is done in a nitrogen environment. Spent fuel is transported from a high energy storage tank to the low energy storage tank after the fuel has cooled down.

The fuel spheres that have to be removed from the storage tanks must be individually loaded into the sphere pipe for transportation in a gas stream of helium or nitrogen to the required destination. Before the spheres are loaded into the sphere pipelines, the broken spheres must be separated from the usable spheres because broken spheres can block sphere transportation pipelines.

The fuel handling system is a controlled system and operated sophisticatedly to ensure accurate measurements of fuel and optimum fuel handling, [1]. Pipe routings and blowers are

sized for reactor performance and the sphere loading to the reactor must be correctly timed to maintain optimum reactor performance with regard to fuel balance, [2].

The fuel handling system is one of the key systems of the pebble bed reactor. The components of the fuel handling system are subjected to the helium atmosphere and high radiation. Therefore components must ensure high reliability and must be maintainable, [8]. Experience gained on the AVR (first German high temperature reactor) indicated that the fuel handling components must become simpler, safer and easier to maintain, [8].

The SSS concept is a new system and at the development phase lags behind the other systems. The previous SSS concept was cancelled due to nuclear safety requirements and it is now required to enhance the new SSS concept to align with the basic design status of the fuel handling system. The success of the current concept design for the whole spent fuel storage system is dependent on the successful development of a tank unloading device which is a main component of the SSS.

1.2 GOAL OF STUDY

The primary goal of this study is to develop a tank unloading device for nuclear spheres that will comply with all the design requirements. To reach this goal within the available time frame, the first step would be to upgrade the current unloading concept which has a risk of damaging fuel spheres. This might be achieved by developing and fitting an impact limiting device to protect the spheres against damage during impact.

If the implementation of the impact limiting device is not successful, another unloading machine concept must be developed, but it must make use of work that has already been done on the present concept.

1.3 SCOPE OF STUDY

The scope of this study is to develop an unloading machine within a helium environment, with available and proven helium technology concepts. This is required to limit expensive development tests in helium for the unloading machine and to save development and testing time. This study will make use of proven helium technology concepts to develop the unloading machine.

Furthermore the study must demonstrate that the developed machine will be reliable and safely maintainable. Equipment failure must be identified to prevent damage to nuclear fuel.

The study will focus mainly on the sphere unloading function of the tank unloading device. The drive of the unloading machine will be from outside the pressure boundary (or containment boundary). This boundary will be penetrated with equipment that has already been developed and tested for the PBMR project. Thus the boundary penetration equipment will not form part of the development in this study. Refer to Figure 1 for a block diagram of the present SSS tank

layout and the main components of the tank unloading device arranged in the allocated spaces. The components and geometry which form part of the scope of the study are encircled with a dotted line. It is important to note that the pressure boundary of the tank is positioned in such a manner that when the pressure boundary is opened, it is opened into the controlled area of the building.

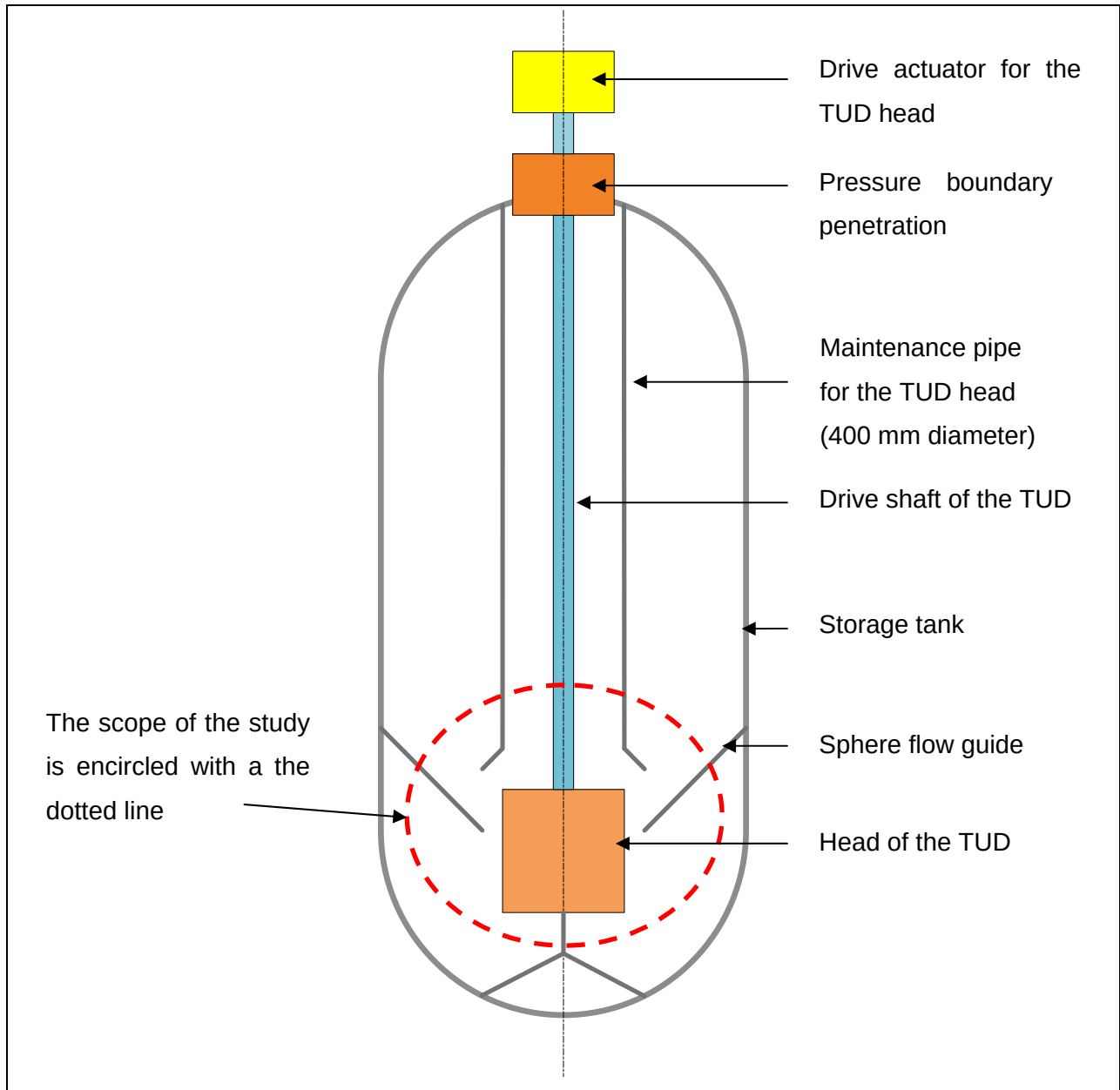


Figure 1: Tank unloading device block diagram

2. LITERATURE SURVEY

2.1 INTRODUCTION

The literature study was performed to understand the environment where the tank unloading device will be operational. The environment is a critical factor when materials are selected. Furthermore, nuclear safety requirements were researched to ascertain what safety requirements will be placed on the design. Other sphere loading and unloading concepts were investigated for possible application in the present study.

2.2 HELIUM TECHNOLOGY

Helium technology is a term used in high-temperature gas reactors, [4]. This type of reactor uses helium gas as a cooling medium, as opposed to water that is used in most other reactors. Due to this new technology physicists are organising regular international meetings that are held on the topic of High-Temperature Reactor Technology. At these meetings papers are presented on helium technology and new findings are discussed. Some of these findings are discussed in the following paragraphs.

Sliding connections, e.g. bearings and slides, are subjected to excessive wear due to the atmospheric and thermal conditions created by the dry helium, even more so when the helium temperature increases, [14]. "Hochtemperatur Reaktorbau GmbH, Mannheim, Federal Republic of Germany", did a study where ceramic coatings were applied to protect components in the helium environment against frictional wear and diffusion welding. However, these new technologies do not have an established track record, even though thoroughly tested.

Graphite becomes abrasive when it is operated in a helium environment, especially at high temperatures [4]. Due to the graphite-coated fuel pebble, which is transported in helium through the fuel handling pipe system, fine graphite dust will form, [5]. Most of this dust will however be captured in the filter system.

Due to the small molecular size of helium it is difficult to prevent helium leakages, [4]. Therefore penetrations through the helium pressure boundary should be limited. Usually drives are placed outside the pressure boundary for maintenance purposes, requiring the remainder of the mechanisms inside the pressure boundary to be reliable and resistant to cold welding in helium.

In France a research programme was conducted to investigate the feasibility of helium technology, [4]. Results indicated that mechanisms in high temperature gas cooled reactors need wear protection. The nuclear regulatory agencies will require tests in helium to determine the durability of coatings on materials.

The friction coefficient between graphite and graphite, as well as between graphite and steel, is higher in helium than in atmospheric air. It has been determined by studies performed, as supported by the Department of Energy, [5], that the friction coefficient increases as the

temperature increases and that it can get as high as 0.7. The results of a study performed in China on the friction coefficient of graphite indicated that the highest friction coefficient of just over 0.4 was obtained in helium at room temperature, [7]. The University of California reported that their test results indicated a lower friction coefficient than indicated by previous tests. According to their research, reference [9], the friction coefficient of graphite should be between 0.5 and 1.0 but it is most often around 0.7. Figure 2 indicates results of previous tests performed by University of California to determine the friction coefficient of graphite, [9]. Due to the fact that the friction coefficient is most often around 0.7, this value will then be used for this study.

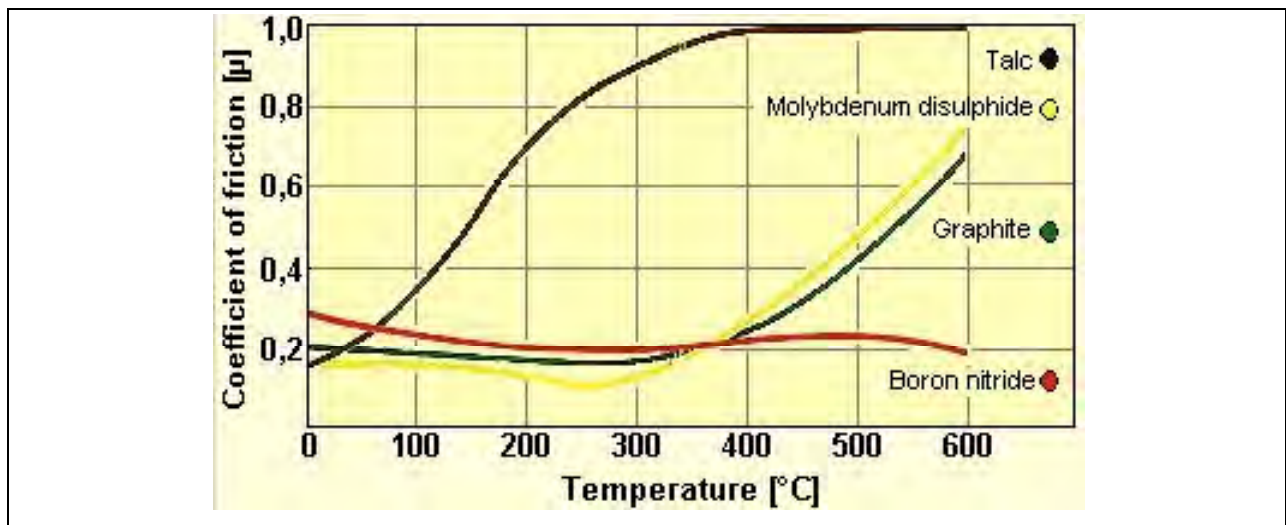


Figure 2: Graphical representation of the friction of graphite, [9]

2.3 FUEL PEBBLE FOR THE PBMR

The typical nuclear fuel sphere has a diameter of 60 mm and has a mass of 0.22 kg. The sphere is made of a 50 mm graphite sphere with an inner fuel matrix. The sphere is then coated with a 5 mm graphite lining, resulting in a final 60 mm diameter sphere, [13].

When a force of 18 kN is applied on the sphere it will be damaged or could even be sheared into two pieces, [15]. Fuel sphere handling machines must be designed to limit impact forces on a sphere to a force lower than 18 kN.



Figure 3: Typical nuclear fuel sphere, [13]

The inner sphere matrix consists of coated fuel particles which are distributed homogenously in the core of the sphere. This core is then seamlessly connected to the graphite outer layer. Figure 4 indicates a cross section of a fuel sphere and an enlarged coated fuel particle from the fuel matrix.

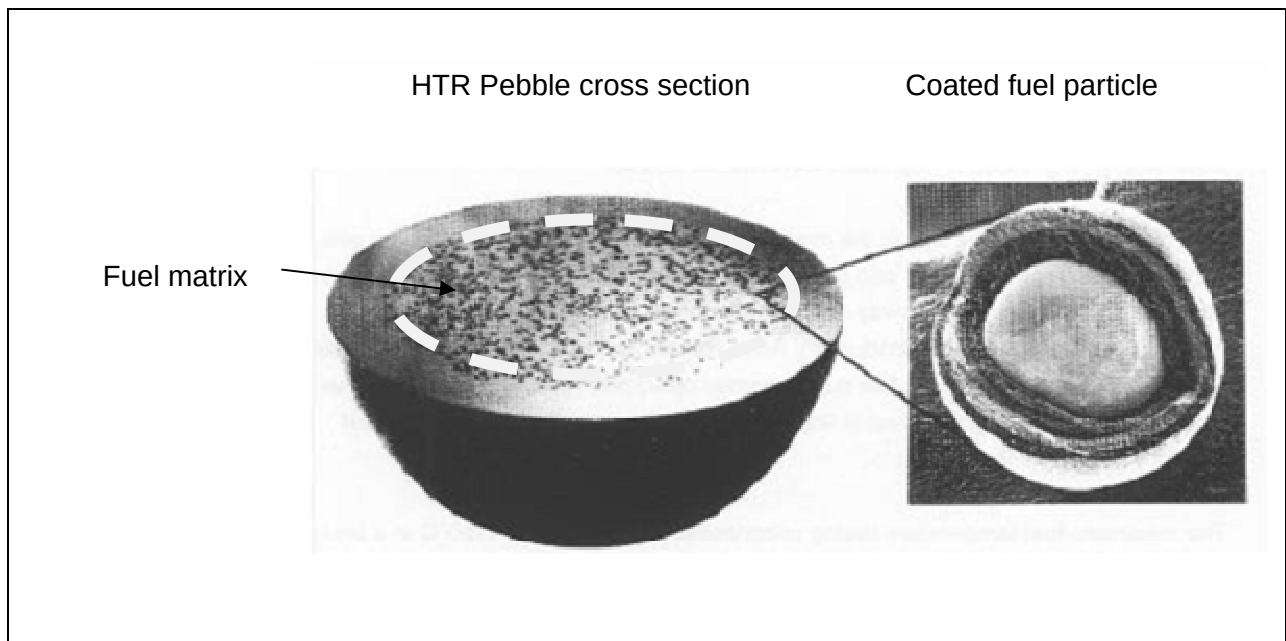


Figure 4: Sphere cross section, [13]

Figure 5 is a graphical representation of the fuel sphere matrix and also an enlarged coated fuel particle as in Figure 4, but the different coating layers around the coated particle are also identified.

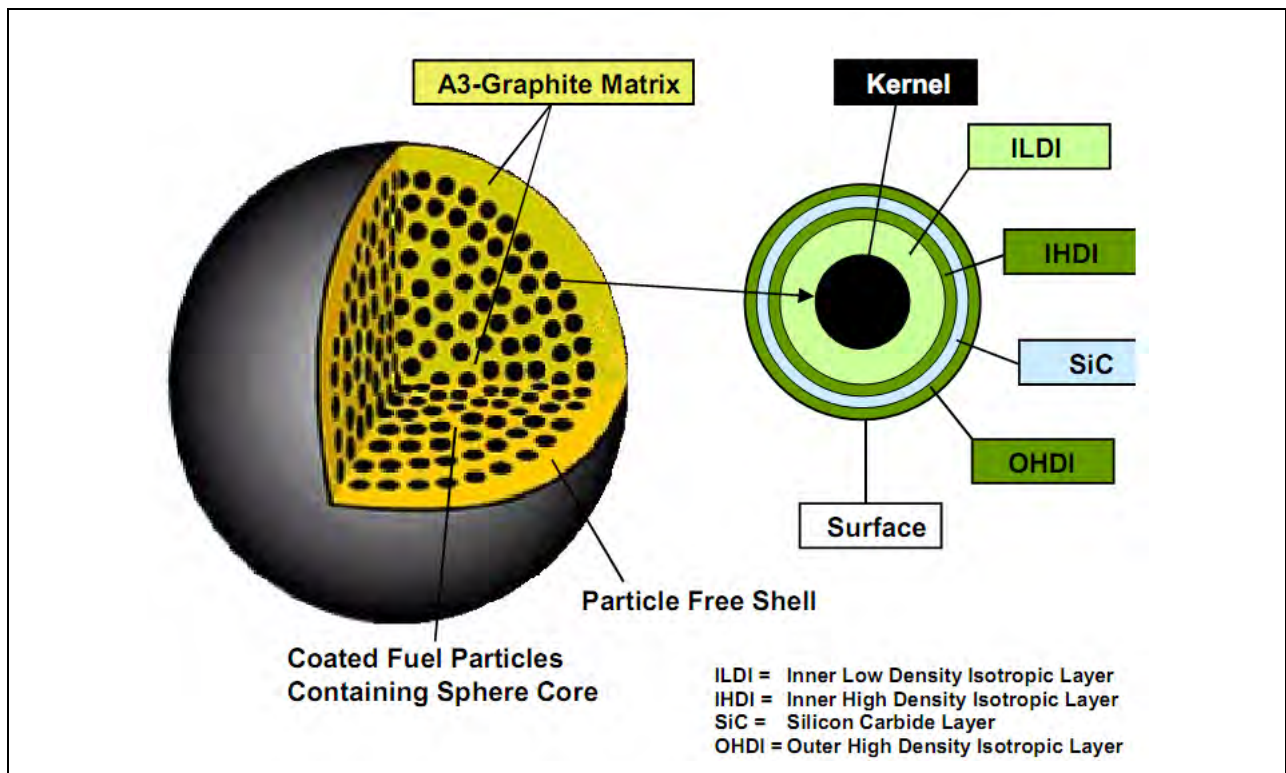


Figure 5: Fuel sphere structure, [11]

2.4 BRIDGE-FORMING

Bridge-forming happens when spheres crystallize in a pattern and no further flow of spheres occurs. Bridge forming is when spheres support each in a locked position to prevent further sphere flow. In an a study done by Dong, [10], it was described that when a pipe diameter is more than five times the sphere diameter, there will be no possibility of bridge forming, thus the spheres will not be able to support themselves to form an arched bridge. These dimensions are indicated in Figure 7. Figure 6 shows a sphere bridge and how the bridge prevents further sphere flow due to the blockage.

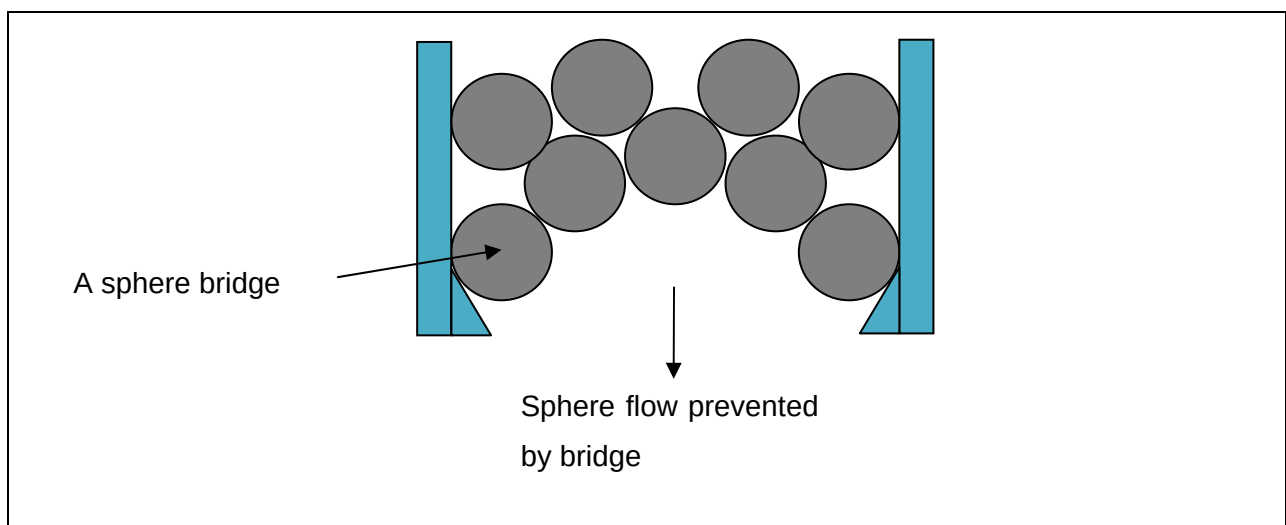


Figure 6: A sphere bridge

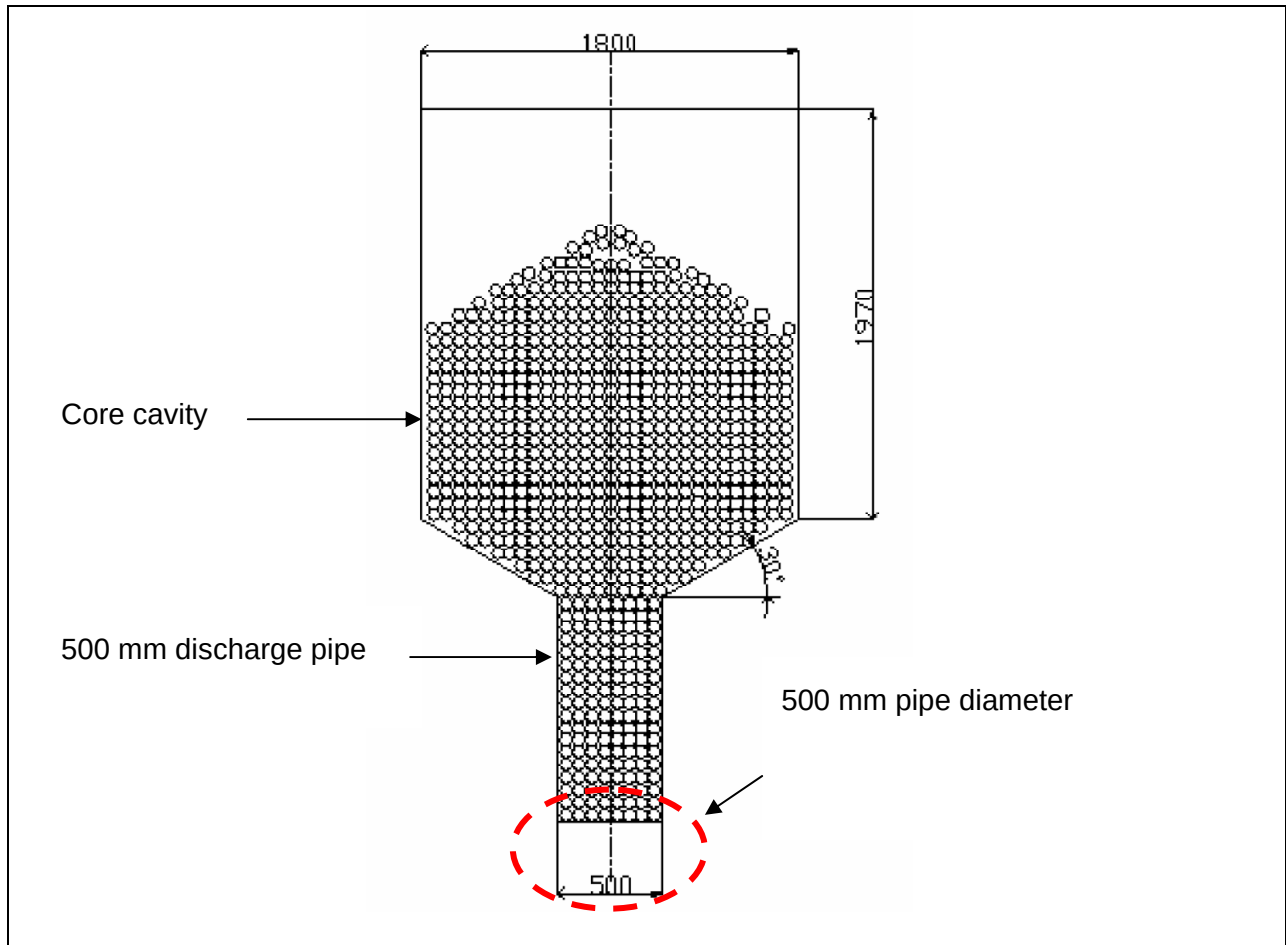


Figure 7: Sketch of the core cavity and discharge pipe of the HTR-10, [8]

In the full scale test setup of the fuel handling system of the HTR-10 the diameter of the funnel which transported the fuel was too small to prevent bridge forming. Therefore gas pulses were used to break the bridge when a bridge was formed, [8].

2.5 SYNTHETIC MATERIALS IN THE NUCLEAR ENVIRONMENT

The application of materials in the PBMR environment is directly linked to helium technology. As previously discussed, the materials will be introduced into a helium environment where they will be subjected to a high-abrasive environment. Additionally materials in high-temperature gas reactors will be subjected to helium temperatures of up to a 1000°C. However, the PBMR fuel handling and storage system will be limited to a maximum temperature of 260 °C, [1]. Therefore the material specification on the tank unloading device allows more materials to be used as in the reactor itself due to the lower temperatures. The abrasiveness of the helium environment will also be lower due to the lower temperature, [4].

Silicone properties were researched to determine whether it could be used in the nuclear industry for seals and springs. It was found that silicone is not resistant to radiation. The effect

of radiation on silicone is the same as the exposure of silicone to high temperatures. According to *Wacker Silicones* brochure, [21], some rubbers can absorb a radiation dosage of approximately 40 – 50 mrad. This dosage reduces elongation at break by up to 50%.

Silicones have however been improved for use in the nuclear industry. The company James Walker claims that they are producing silicones that have been developed for use in the nuclear industry that are very reliable, [24]. Silicone Engineering in the United Kingdom provides datasheets, [25], where the silicone resistance to radioactivity is mentioned.

Silicone hardness is measured in durameter, [26]. This measurement unit describes the force that is required to compress silicone. The higher the durameter reading, the higher the force that is required to increase the dent in the silicone. These properties were used to determine the spring k-value for the silicone which is a function of the force used to compress the silicone, $F = k \times \Delta x$.

Vespel is a material that is used in the nuclear industry for bearing surfaces. The *Dupont* design handbook for Vespel, [27], describes that it is used for applications in the nuclear industry and resistant to radiation. Vespel could therefore be used in bearing applications in this design since there is already a history of Vespel applications in the nuclear industry.

2.6 NUCLEAR SAFETY PRINCIPLES

Nuclear safety is always of primary concern pertaining to designs of systems and equipment in the nuclear industry. According to INSAG 12, [12], there are three fundamental safety functions (FSF) that must be complied with in the nuclear industry:

- Controlling reactor power;
- Cooling the fuel;
- Confine radioactive materials within physical barriers.

The tank unloading device has no fuel cooling or reactor power control function, or influence on these functions. Therefore the design of the tank unloading device will only have to comply with the third requirement, confinement, when the unloading machine has to penetrate through the radioactive barriers.

Defence in Depth (DiD) is a system where different levels of equipment and procedures are applied to maintain physical barriers, [16]. The function of these barriers is to protect plant personnel and the public against radioactive products. Thus, containment barriers are required to prevent the escape of radioactive products from the pressure boundary of the system containing the radioactive products into the environment.

There are five levels of protection. They are listed in Table 1, as extracted from INSAG, [16]:

Table 1: DiD Levels

Levels of DiD	Objective	Essential means
Level 1	Prevention of abnormal operation and failures	Conservative design and high quality in construction and operation
Level 2	Control of abnormal operation and detection of failures	Control, limiting and protection systems and other surveillance features
Level 3	Control of accidents within the design basis	Engineered safety features and accident management
Level 4	Control of severe plant conditions, including prevention of accident progression and mitigation of the consequences of severe accidents	Complementary measures and accident management
Level 5	Mitigation of radiological consequences of significant releases of radioactive materials	Off-site emergency response

The design of the tank unloading device must comply with levels one and two of the five levels of DiD, as applicable, to protect plant personnel and the public during normal operation and anticipated operational occurrences. These requirements will become effective when the head needs maintenance and the pressure boundary must be opened.

2.7 SPHERE UNLOADING DEVICES

2.7.1 SPHERE INDEXING DEVICE

A sphere-unloading tool is indicated in Figure 9. The sphere release finger from *Pipeline Engineering* enables a multi-loaded sphere launcher to sequentially (thus one at a time) launch spheres without opening the launcher. This reduces the time to de-pressurize and re-pressurize the system after each launch. When fitted to receivers, they allow the controlled, safe unloading of multiple spheres, one at a time.

For a multiple sphere launching capacity the launching vessel can be designed to contain up to 10 individual spheres with the launch of each controlled by a pair of release pins or flaps. In the case of the present study, the release pins and flaps will not be maintainable in the fuel storage tank and will get damaged in the helium atmosphere. This machine can be blocked when broken sphere pieces lodge between the finger and the pipe. Figure 8 shows how a piece of broken sphere can wedge between the indexing finger and the sphere pipe. The piece of broken sphere will thus also prevent the next sphere from being indexed because the sphere will not be able to fully enter the sphere pipe.

Therefore this indexing with a finger principle will not be effective in the present study where broken sphere pieces will be present in the sphere lines.

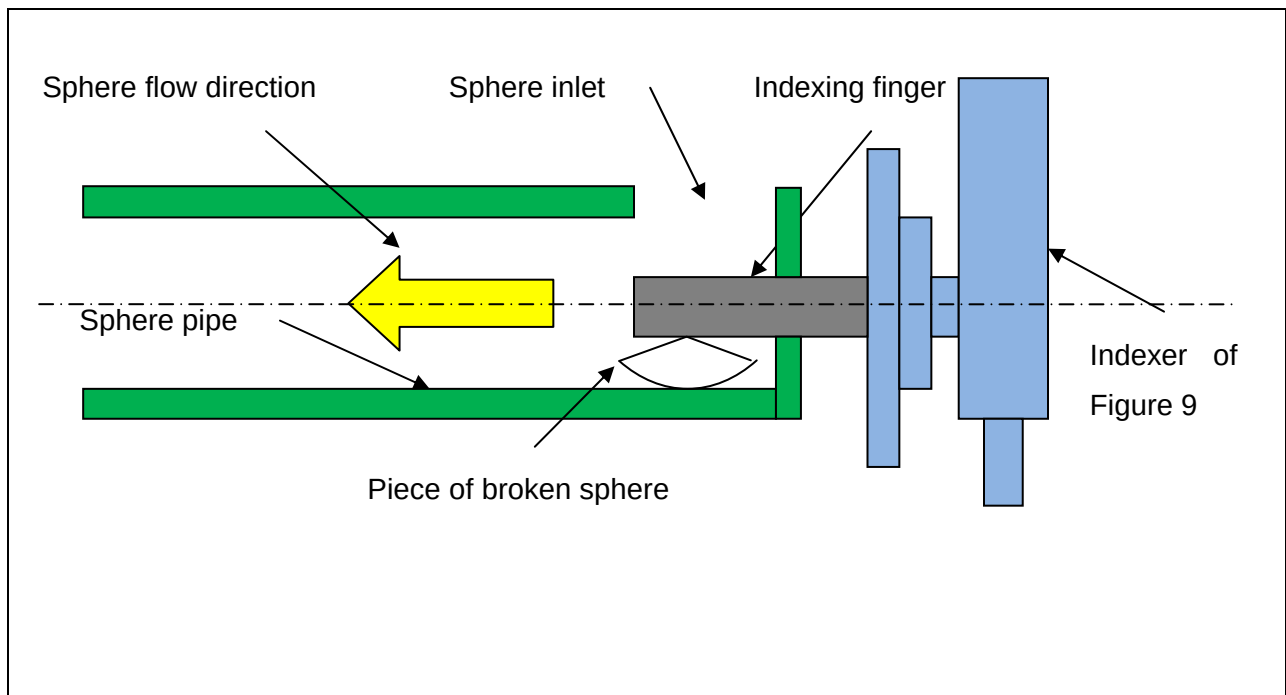


Figure 8: Diagram indicating blocked indexing finger



Figure 9: Unloading concept: *Pipeline Engineering*, [17]

2.7.2 SPHERE PICK-UP MECHANISM

The machine shown in Figure 10 has been developed for the use on tennis courts to collect tennis balls after a game. The machine required for the unloading of nuclear spheres will be fixed to the tank, therefore no wheels are required. However, the tennis ball machine's rotary wheel concept is of interest, due to its possibility to singulize nuclear spheres.

This concept could work to index spheres into the sphere transportation pipe. However, it would require bearing points that must be sealed to protect the bearing against helium where cold welding can occur. Furthermore, the sealing points must prevent a break in the helium

pressure boundary. This concept will work with usable spheres. However, when pieces of broken spheres are loaded and a piece of broken sphere and a usable sphere are in one loading segment, the rotating mechanism will be blocked.

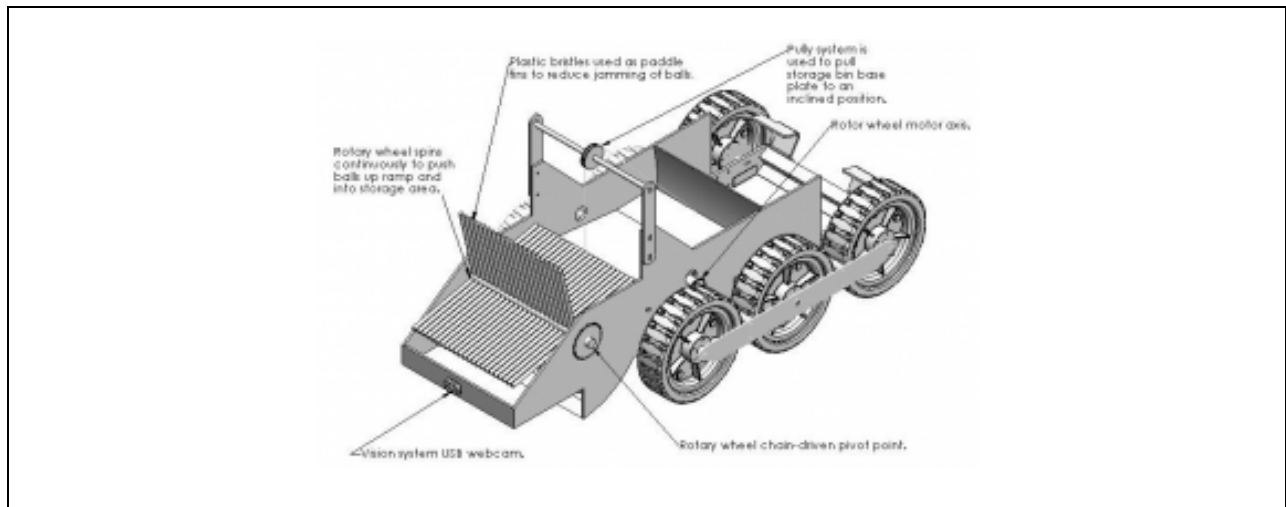


Figure 10: Sphere pick-up mechanism, [18]

2.7.3 BALL LOADER

A ball loader concept is shown in Figure 11. It consists of a feeding ramp where balls are rolled onto the two rotating wheels. The wheels move the balls further when the ball moves between them. Sphere indexing can be done by using this concept. The spheres are rolled into position against the wheels, now not rotating. When a sphere must be indexed the wheels can be rotated enough to feed one sphere through to the next section, or into the sphere transport pipe.

A usable, round sphere will be gripped between the two wheels and taken through by the rotating wheels for indexing, while a piece of broken sphere will not be gripped because it is too small. The indexing wheels are spaced for 60 mm fuel spheres. A broken sphere that is smaller than 60 mm will cause a blockage between the two wheels because the wheels will not be able to grip the piece of broken sphere and move it along.

This concept also has moving parts that will require maintenance actions in the helium environment to prevent cold welding. The wheels must be positioned correctly to be able to feed the spheres through. The spheres will roll down under gravity onto the wheels. Therefore the wheels must be positioned below the sphere bed, thus in the storage tank filled with helium.

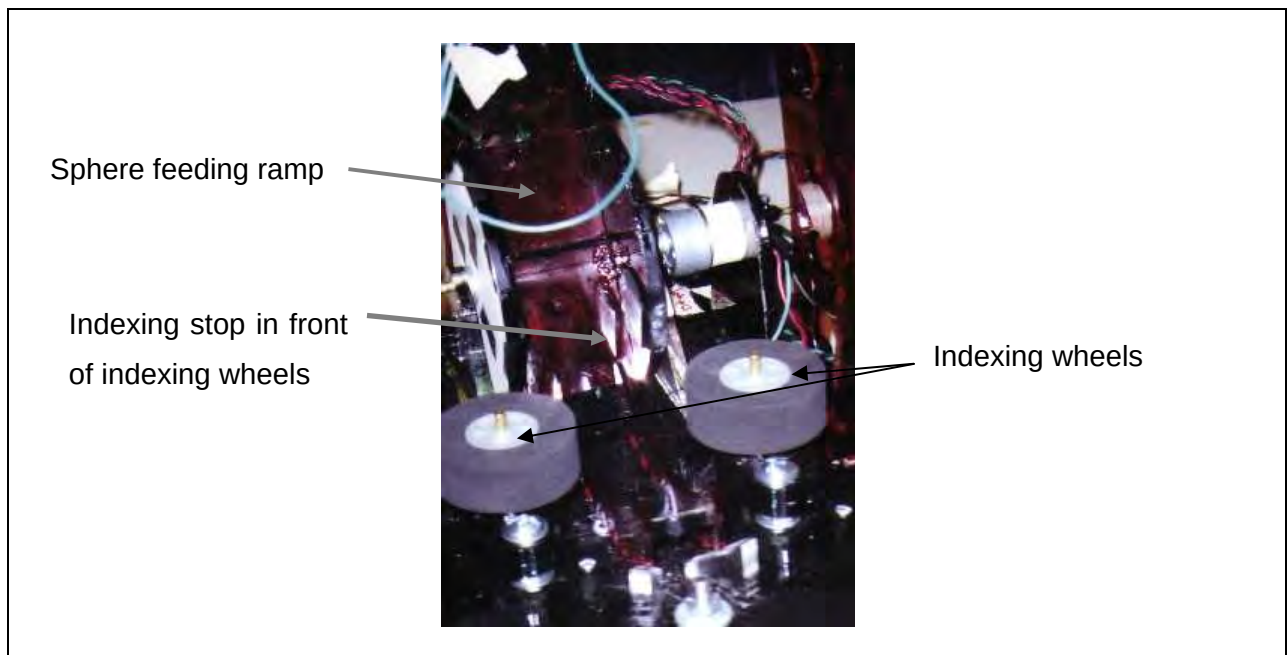


Figure 11: Ball Loader

2.7.4 PAINTBALL GUN LOADER

A further typical ball loader mechanism is found in paintball guns. Figure 12 indicates the head where the ball is removed from the bed of balls and guided towards the loading port of the gun. The mechanism is provided with flexible feeders (item 32 in Figure 12), which moves the ball forward without damaging the ball. When the opening to the gun is open again, the string of balls is moved forward again to drop another ball into the barrel.

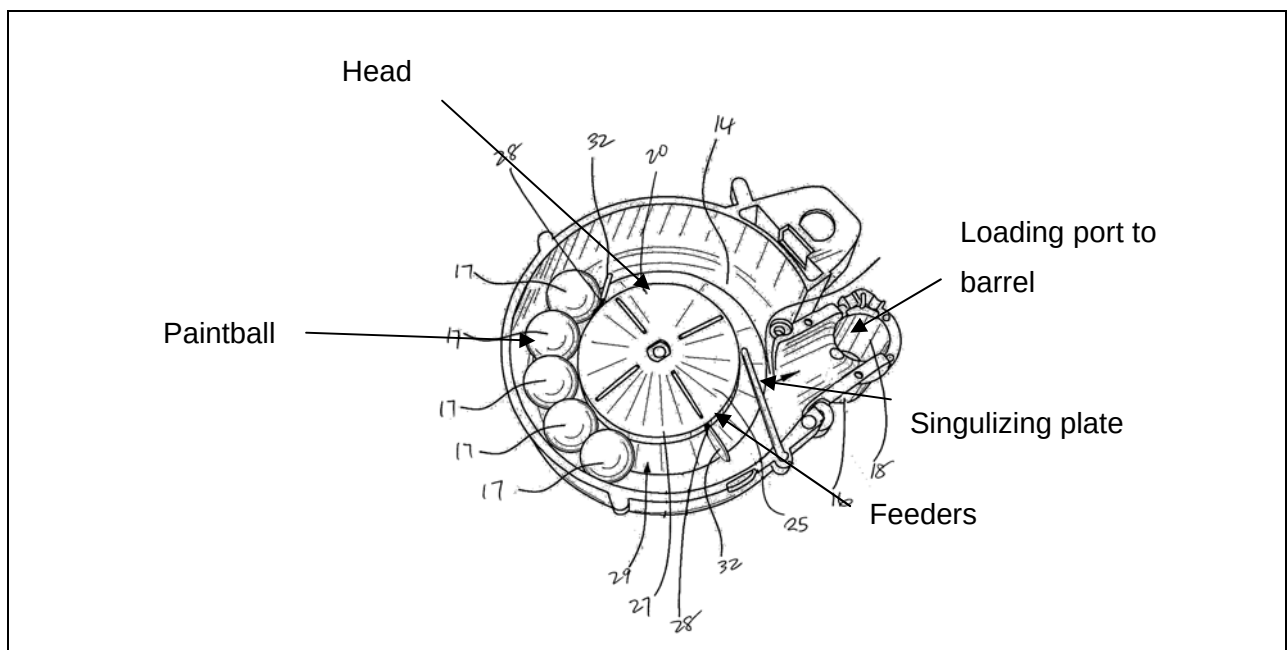


Figure 12: Paintball gun loader

Figure 13 indicates the indexing action of a typical paintball gun loader. The balls are indexed between the upper and lower sprockets into the barrel of the gun.

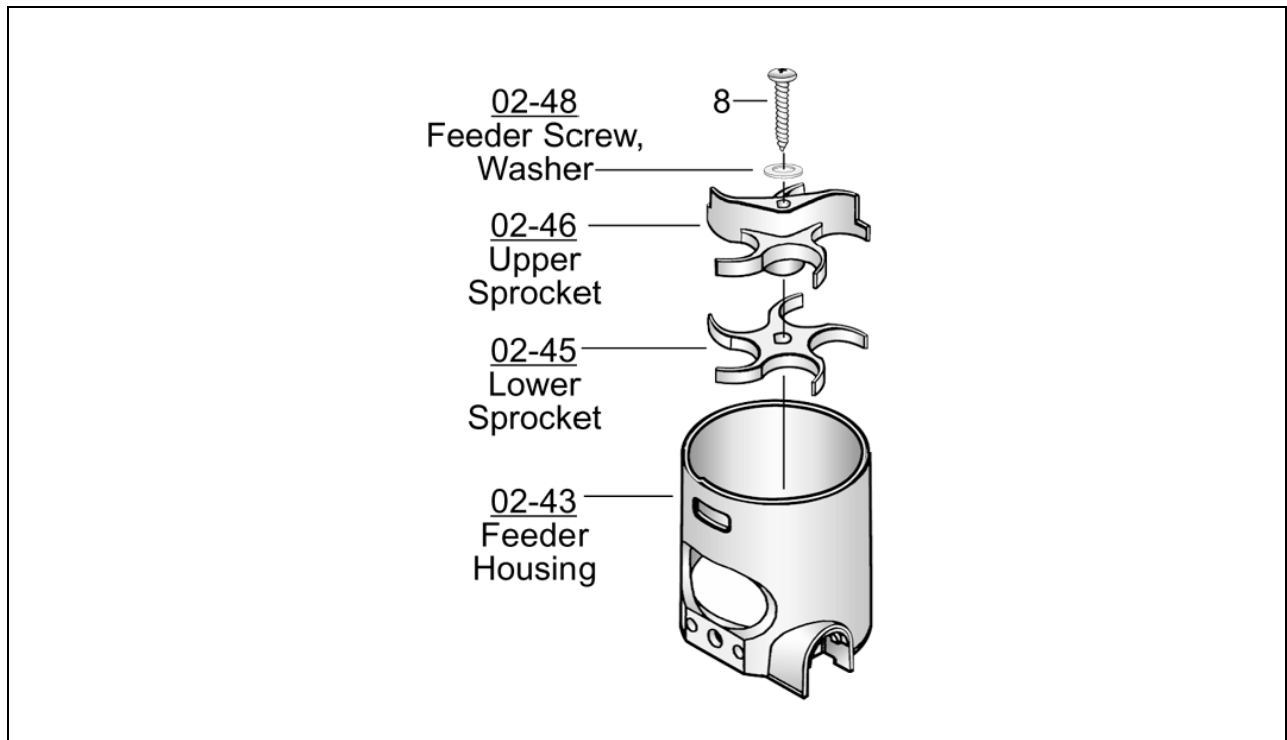


Figure 13: Paintball gun singulizer, [19]

These loaders are reliable in loading balls due to the good quality of the balls. The balls are standard in size and no broken balls are present. When broken balls are introduced into the mechanism, blockages will occur causing no feed to the gun because these mechanisms are not designed to operate with broken paintballs.

2.7.5 LEVITATION STIR BALL LOADER

Figure 15 is a machine that is used to load plenums with balls. Figure 16(b) indicates these plenums. Figure 16(a) indicates the main components of the loader section. The plenum (item 21 in Figure 16) is filled with spheres by the rake, item 10 in Figure 16(a), which is moved linearly by a pneumatically operated piston. When the plenum is full it is taken out and replaced with an empty plenum.

This concept will not be successful for the present study application. This design works only with usable spheres. When pieces of broken spheres are present, a sphere location hole can be filled with a piece of broken sphere and a usable sphere, causing the plenum to be stuck because the sphere is protruding from the plenum into the section above the plenum. Figure 14 shows how a piece of sphere will block this machine.

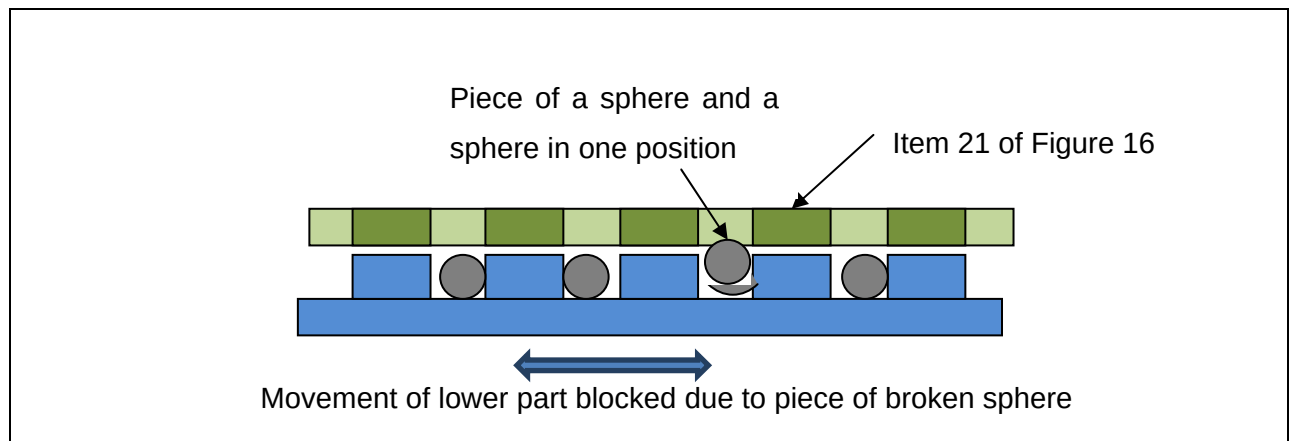


Figure 14: Diagram showing blocked movement

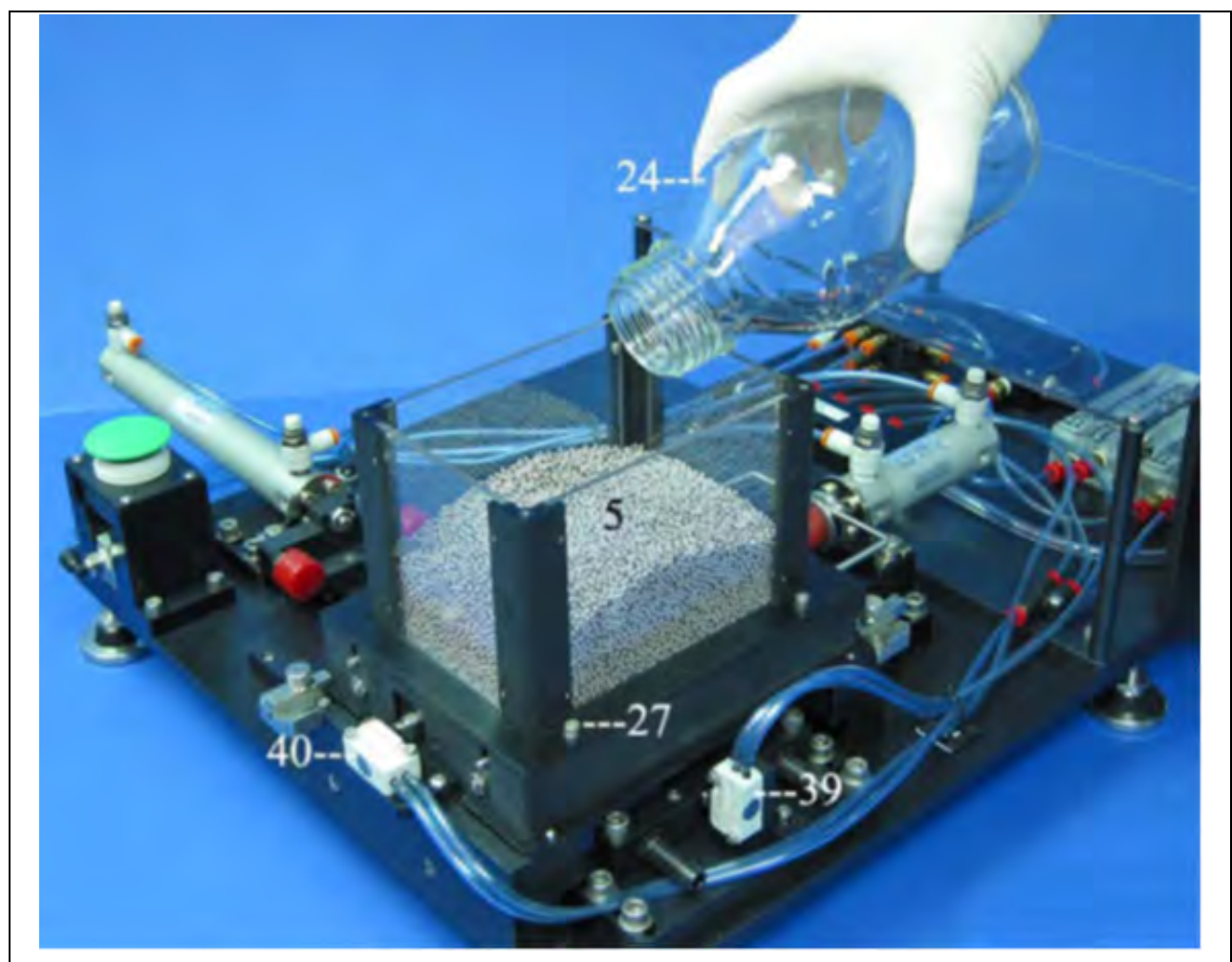


Figure 15: Levitation stir ball loader, [20]

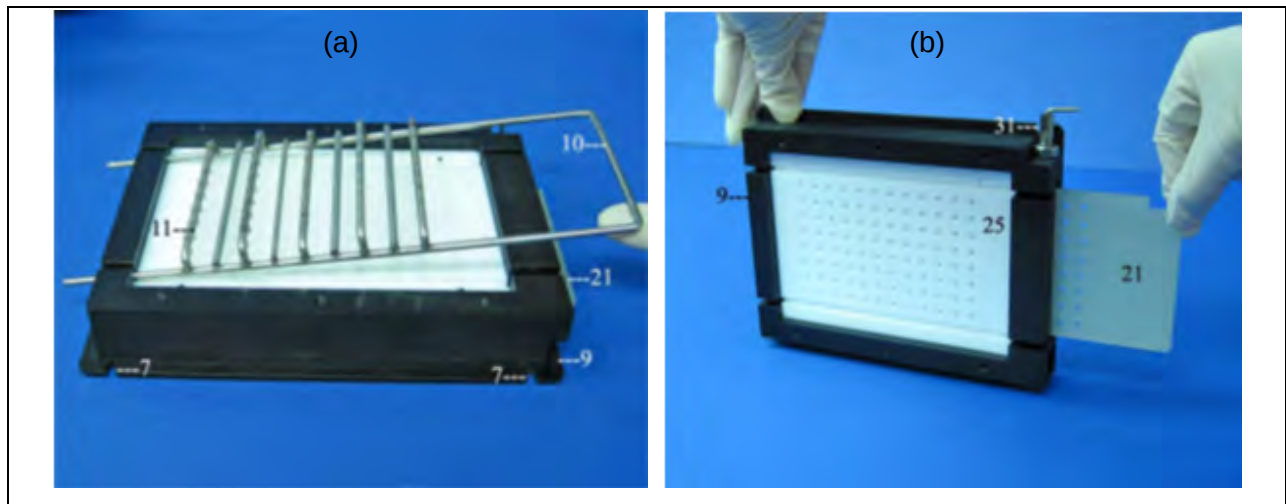


Figure 16: Main components and loading plenum, [20]

2.7.6 NUCLEAR FUEL SPHERE UNLOADING DEVICES

There are other fuel unloading device concepts from earlier concept designs of pebble bed reactors. Figure 17 shows some conceptual drawings of unloading devices that were done in 1962 for a conceptual pebble bed design, [21].

These designs are all unloading fuel through the bottom of the tank, while it is required for the present study to unload fuel only through the top of the storage tank. These concepts will be investigated further during the research study.

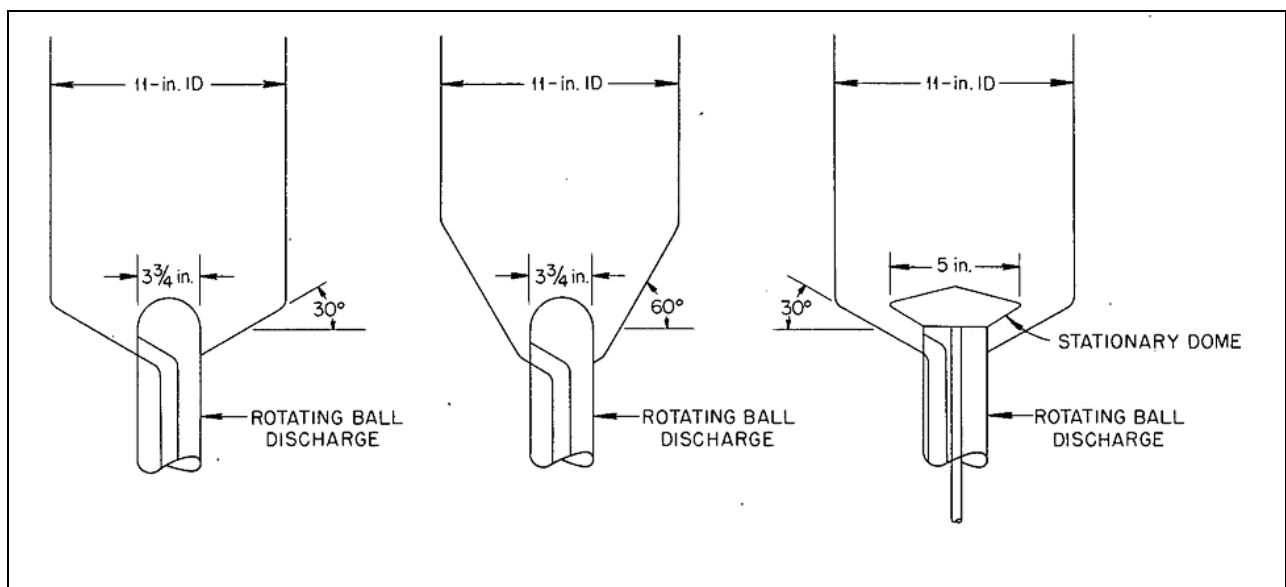


Figure 17: Gravity fuel unloading devices, [21]

Figure 18 shows an unloading machine concept for the unloading of an ordered bed core, [22]. This unloading machine is a vacuum concept opposed to the gravity concept as indicated in Figure 17. An ordered bed core is when the spheres are packed in the core in guides to create

an ordered packaging, as shown in Figure 19. With an ordered bed core the machine can be stepped down according to the core layer distances to unload the core. However, with a bed such as the one used in the PBMR where the spheres are loaded in a large open core cavity without guides, it will be difficult to control the suction nozzle towards the sphere bed, without forcing the suction head into the sphere bed. When the head is forced into the sphere bed the entrance to the suction nozzle will be blocked and the spheres will be damaged.

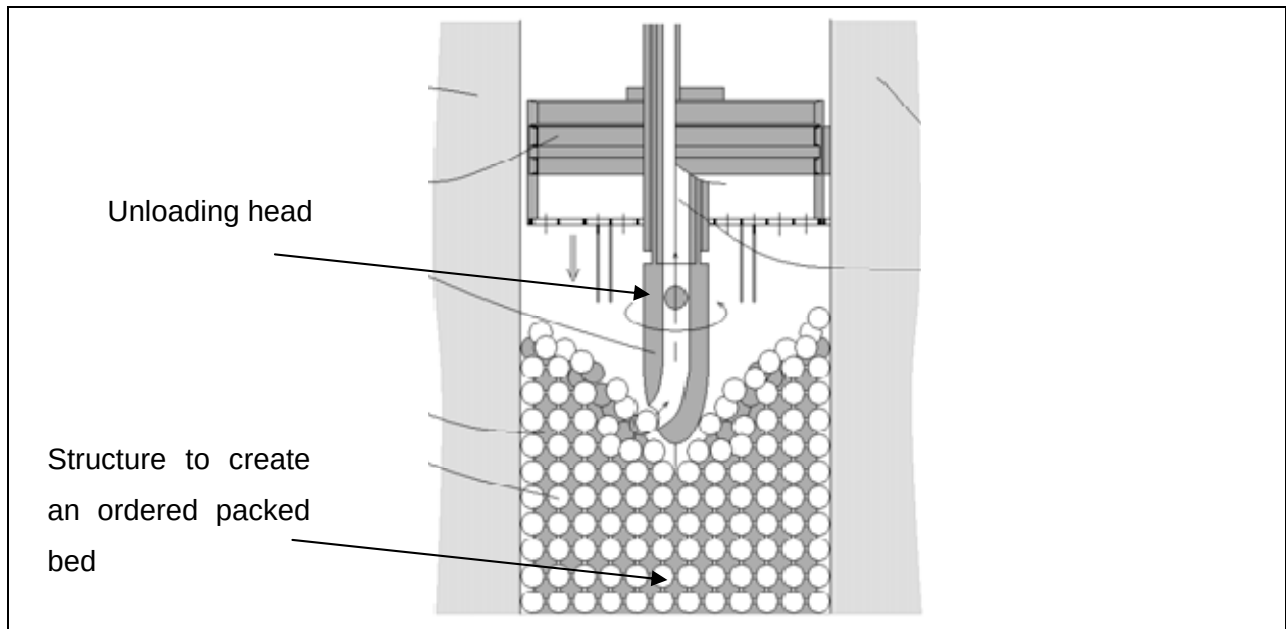


Figure 18: Vacuum fuel unloading of an ordered bed core, [22]

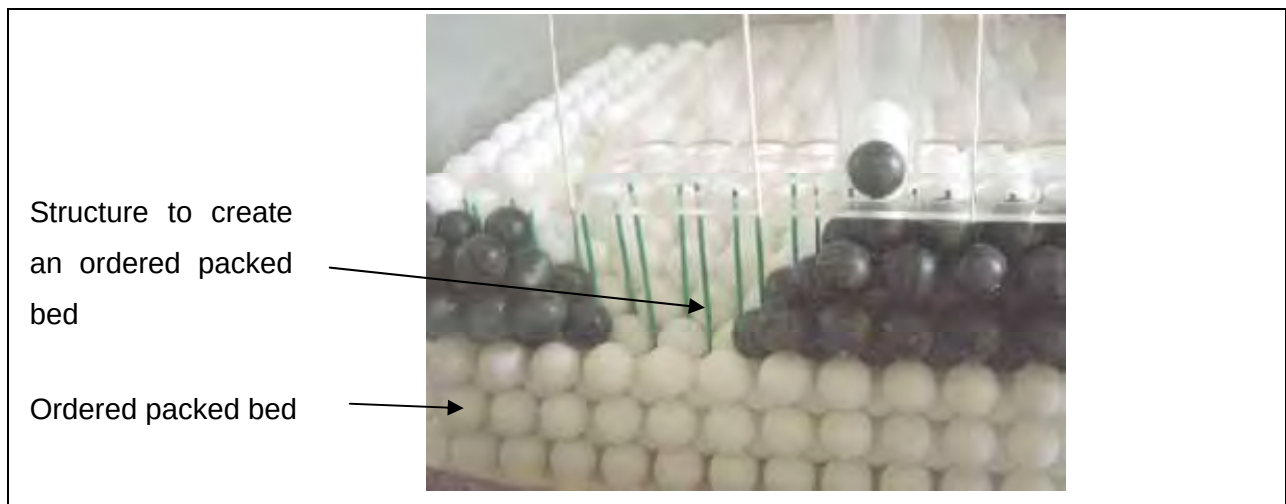


Figure 19: Ordered packed bed, [22]

2.8 CONCLUSIONS

The design of components that are used in the reactor system must take the dust in the system into account and the design must include contingencies for dust removal, [5]. Furthermore the research on helium technology showed that a tank unloading device with the minimum of friction

components must be designed or selected. The components subjected to friction must be designed to withstand the highly frictional environment of dust and helium.

It can also be concluded that a safe design, from a maintenance point of view, will be constructed from metallic items rather than silicone type of materials because metallic items are more resistant to radioactivity.

The maintenance requirements of the tank unloading device must take in consideration that the tank unloading device will be in a radioactive environment. Based on the research regarding nuclear safety principles, care must be taken when opening a radioactive containment system or area.

The sphere handling concepts that were researched and described will not succeed in its present designed layout as a tank unloading device. All these concepts have too many friction points that can fail due to wear of the bearing interfaces. To prevent failure of these components due to wear, regular maintenance actions will be required during the TUD's lifetime. These unloading concepts have not been designed for easy and safe maintenance in the radioactive environment.

3. CONCEPTUAL DESIGN

3.1 INTRODUCTION

This section presents the concepts that were developed for the tank unloading device. A short description will be given of the concepts. Thereafter a concept selection will be done and then future development will be pointed out.

3.2 DESIGN REQUIREMENTS AND DESIGN SPECIFICATIONS

The following functional requirements are directly applicable to the TUD and summarised as follows:

- Unloading performance: amount of spheres per hour;
- Operating medium: helium and nitrogen;
- Maintenance: time required for maintenance in radioactive environment;
- Broken pieces separation: description of identification of broken pieces to follow.

The main performance specification that the tank unloading machine must comply with is:

- Deliver a minimum of 333 spheres/hour;
- Indexing must be at regular intervals.

Safety and maintenance requirements are generally applicable to equipment in the nuclear industry. Safety requirements must be adhered to in the nuclear industry. The drive of the tank unloading device will penetrate through the pressure boundary into the radioactive environment, requiring that this concept must be designed to be functional in the radioactive environment filled with dust and helium. A robust and maintainable tank unloading device must be designed.

Maintenance procedures must be developed for each maintenance task in the nuclear industry to ensure personnel safety due to the high risk of radiation. Therefore equipment is designed for low maintenance requirements to reduce cost on elaborative maintenance preparation and tasks.

3.3 CONCEPT GENERATION

Two concepts are being investigated for this study. Both these concepts utilize an unloading head in the storage tank with a drive unit outside the storage tank with a penetration item to penetrate safely through the containment barrier into the tank where radioactive fuel is stored. The penetration item will not be part of the study because it has already been tested for reliability in the nuclear industry.

The first concept utilizes pneumatic suction to lift the sphere into the sphere transportation pipe, while the second concept uses gravity and mechanical equipment to unload the spheres from the storage tank.

To remove spheres from a storage tank into a transportation pipe, the spheres must be loosened from the interlocking force caused by the weight of the spheres on top of them. Then the spheres must be aligned for indexed removal, where after they can be released in a separately into the transportation pipe.

It was decided to build concept-demonstrating models. The purpose of these models was to demonstrate the concept of suction or gravity removal. The one model used suction to remove spheres, while the other concept used rotational movement to remove the spheres. Figure 20 shows the suction model and Figure 21 the gravity model that were built.

Both concepts were functional in unloading the spheres. However, the suction option required a high suction rate to lift the sphere and suck it into the lifting pipe. Once the sphere was in the suction pipe the sphere accelerated to a very high velocity due to the high suction rate required to lift the sphere from its resting position. Therefore the suction concept was not developed any further.

3.3.1 CONCEPT 1: PNEUMATIC SUCTION

This concept uses pneumatic suction to lift the fuel sphere from its position into the suction nozzle of the head into the suction pipe in the unloading head. Figure 20(a) is a picture of the scale model that was built to do the suction tests. Figure 20(b) is a cross section of the head with the conveying pipe inside, the bottom of the tank and some spheres.

The spheres in the tank roll down to the lowest point in the tank. There is a base fixed in the middle of the tank bottom where the head tank unloading device interfaces for rotational bearing support. The space between the head and the tank bottom is approximately 65 mm to allow a ring of spheres to form around the base underneath the head. The head of the unloading device rotates just above the spheres while enough air to lift a sphere is sucked into the suction nozzle of the head. When the suction nozzle in the head is moved over a sphere the sphere is sucked into the suction nozzle of the head, from where the sphere is transported the top. Meanwhile the head rotates further to the next sphere lying against the base which is then sucked in. Singulising of spheres are achieved when the head rotates and spheres lying in a ring around the base are sucked in as the nozzle pass over them. The unloading intervals between the spheres can be adjusted by rotating the head faster for shorter intervals, or slower for longer intervals.

This concept was able to unload spheres. Furthermore, the delivery of spheres can be reduced by decreasing the head rotational speed, with zero delivery at zero speed. Thus with better control of the blower performance and the rotational speed of the head it should be possible to deliver the required 333 spheres/hour. Due to the limited selection of spheres and blowers available these tests were performed with lighter sphere masses and a blower capacity that is to

powerful. However, the purpose of the test was not to verify the design, but only to confirm that suction could be used as an unloading possibility.

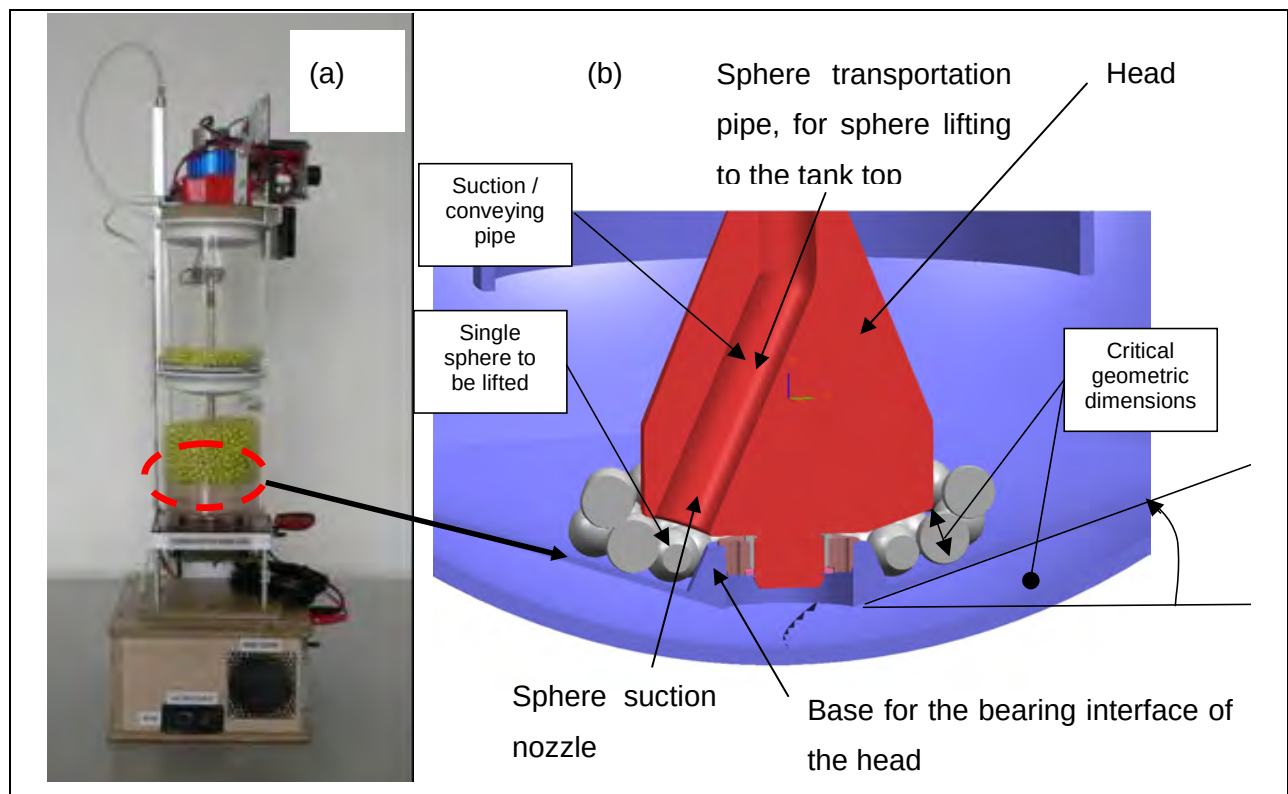


Figure 20: Vacuum concept (a) test model; (b) the detail of the unloading head

A disadvantage of this concept is that high sphere velocities are created. When a sphere is lying in the open, thus not in a pipe, a high gas velocity is required from a nozzle to lift the sphere and suck the sphere into the nozzle. The gas velocity over the sphere causes friction over the sphere to lift it. Once the sphere is lifted a pressure differential exists over the sphere causing it to be lifted into the lower pressure region. Once the sphere enters the suction pipe it is transported due to a pressure differential over the sphere due to the moving transportation gas. To create this pressure differential over the sphere less gas flow velocity is required as when a sphere is to be lifted from a flat area. A further disadvantage of the high suction velocity is that it is now possible to lift more spheres into the suction pipe due to this high gas velocity. When there is too much spheres in the vertical sphere transport pipe the spheres will hover in midair due to blower capacity that cannot lift the spheres further. The string of spheres will then hover in the suction pipe until the suction is removed and the spheres are allowed to roll down.

To make this concept more acceptable it would be required to decrease the gas velocity once the sphere enters the conveying nozzle in the head, and then to increase it again when the sphere is at the top of the tank so that the next sphere can be sucked into the conveying pipe. The gas velocity can be controlled with by-pass pipes between the suction nozzle and the top exit. When a sphere is in the lifting section between the nozzle and the top exit, some of the

gases will by-pass the sphere pipe in the parallel by-pass pipe. This by-pass pipe must be smaller in diameter than the sphere pipe to maintain some gas-flow through the sphere pipe for sphere lifting. There is however a possibility that the smaller diameter by-pass pipe can be clogged with graphite dust.

It will be possible to do maintenance on the vacuum concept by removing the head with its driveshaft at the top of the tank. This process will require opening of the pressure boundary which will require special tools in the nuclear environment.

3.3.2 CONCEPT 2: GRAVITY UNLOADING

Figure 21(a) is a scale model that was built to test the gravity unloading concept. Concept 2 uses gravity to drop spheres down into a sphere transport line into a constant gas flow rate.

Figure 21(b) indicates a single sphere in the groove between the tank bottom and the head of the tank unloading device. The groove has the width of a sphere and the purpose is to form a single row or string of spheres in the groove.

Spheres are taken from the sphere bed into the groove. While the head of the unloading device rotates a sphere rolls down through the head into the sphere pipe below. Spheres can be indexed faster by rotating the head faster, and vice versa.

The length of the vertical pipe segment in the head can influence the amount of spheres unloaded into the gas stream. The pipe fills up with spheres and the bottom sphere is taken by the gas stream. A short pipe segment will prevent too many spheres from flowing into the gas stream.

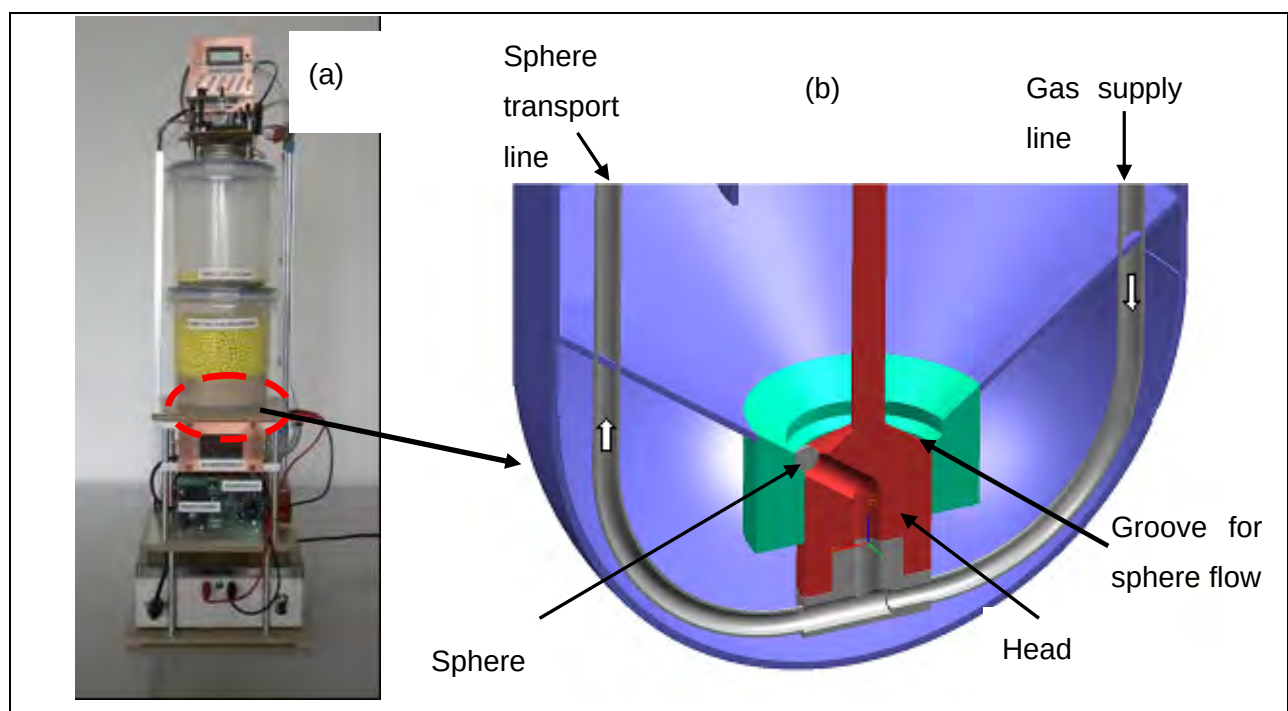


Figure 21: Combination of gravity concept

Spheres were unloaded in strings by the gravity concept. By decreasing the rotational speed delivery had a tendency to decrease until zero spheres were delivered when the head was not rotating. When there are too many spheres in the pipe the given blower does not have the capacity to lift the mass of spheres. Again the purpose of this test was only to show that the gravity unloading concept was possible.

It was concluded that better control had to be incorporated into the design for better singulization of the spheres because the gravity concept worked well to unload spheres only when the head rotated slowly. When the head was rotating too fast a string of spheres fell through into the sphere transport pipe. It is therefore recommended that a more effective singulizing action must be developed for this concept to increase timely indexing in the unloading sequence.

3.4 CONCEPT SELECTION

For the concept selection the criteria in Table 2 will be used. The weights scaling is indicated with 1 as the most important weight.

Table 2: Concept selection criteria

Criteria	Criteria weight	Description	
Control of indexing	4	Indicate how accurate the indexing of spheres can be controlled.	1: no control 10: accurate control
Control of sphere speed	5	Is the sphere speed within transportation speed limits, with a maximum of 10 m/s?	1: speed not in limits 10: speed within limits
Sphere string forming	3	Is there a tendency to load strings of spheres into the sphere transport pipe?	1: yes 10: No strings
Dust environment	2	Are there changes that dust will cause failure of the concept?	1: yes 10: no
Helium environment	1	Are there frictional components other than <i>vespel</i> ?	1: yes there are 10: no, nothing
Construction simplicity	7	Constructability of the concept	1: complex 10: easy
Maintenance	6	Is maintenance possible?	1: difficult 10: easy
Development status	8	Is further development required?	1: yes 10: no

Table 3: Concept selection

Criteria	Vacuum	Gravity	Selection
Control of indexing	4/10	5/10	Gravity
Control of sphere speed	1/10	7/10	Gravity
Sphere string forming	8/10	8/10	Any
Dust environment	3/10	8/10	Gravity
Helium environment	8/10	8/10	Any
Construction simplicity	8/10	7/10	Vacuum
Maintenance possible	8/10	8/10	Any
Further development	3/10	5/10	Gravity
Totals	43	56	

From Table 3 the gravity option seems to be the better option. Therefore the gravity option has been selected. In both cases improvement is required for the concepts to comply with the design requirements. There is documented evidence available from the gas reactor history on gravity unloading which added to the choice of selecting the gravity concept for improvement. These concepts were shortly described in the literature study.

This gravity concept has a slightly different geometry than those given in the literature study. It was done because it was reasoned that the sphere will line up in the groove for single unloading.

3.5 CONCLUSION

The principle of vacuum lifting and gravity unloading are both workable solutions. It is therefore envisaged that both concepts can be further developed into working concepts. With the limited time frame the gravity concept was selected because of better compliance with the requirements.

It is however evident that the concept needs to be developed further to allow more reliable indexing.

4. DETAIL DESIGN

4.1 INTRODUCTION

The gravity concept was selected, but with the recommendation of further development of the indexing mechanism. The detail design will include the improvement of the concept as well as the calculations to ensure that the tank unloading design will comply with the design specifications.

4.2 DETAIL DESIGN

4.2.1 CONCEPT IMPROVEMENT

The gravity concept has been improved. A mechanical separation action is now included in the gravity concept to be able to separate and index spheres reliably. The main principle of this improvement is that the spheres that are strung in the groove are now prevented by a mechanical separation action from rolling freely into the sphere transportation pipe. As shown in Figure 22 only a single sphere can enter the head until the sphere reaches the housing. The housing is a stationary item while the head rotates. The rotating head moves the sphere until the sphere reaches the exit hole in the housing and then falls through into the sphere transportation pipe. The next sphere cannot enter behind the first sphere because the head has already passed the housing exit hole.

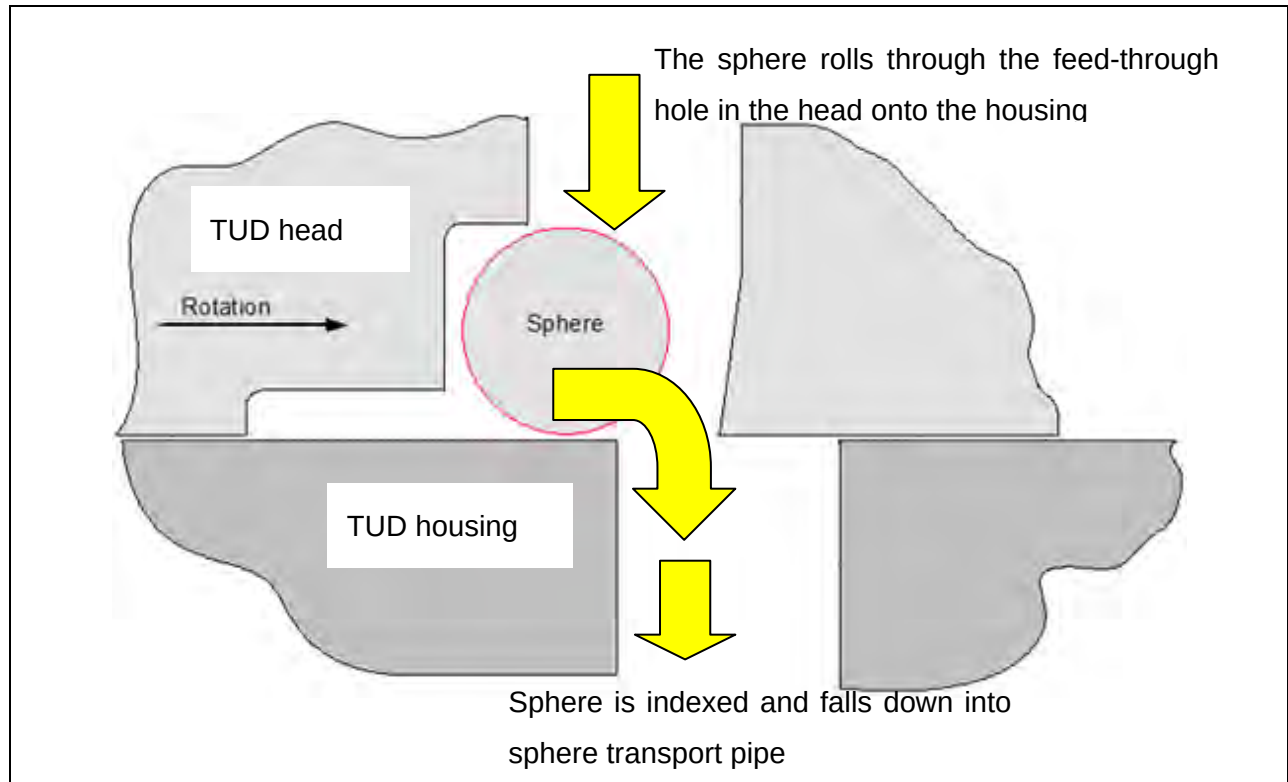


Figure 22: Mechanical sphere indexing

Figure 23 shows the head which is provided with feed-through holes spaced 180° apart. The function of the head is to retrieve a fuel sphere from the sphere bed and then to transfer this sphere to the housing, Figure 24. The head has a diameter specified to prevent bridge-forming. Bridge-forming happens when the spheres interlock with each other in the surrounding geometry to form an interlocked sphere mass which prevents further sphere flow, [8] and [10].

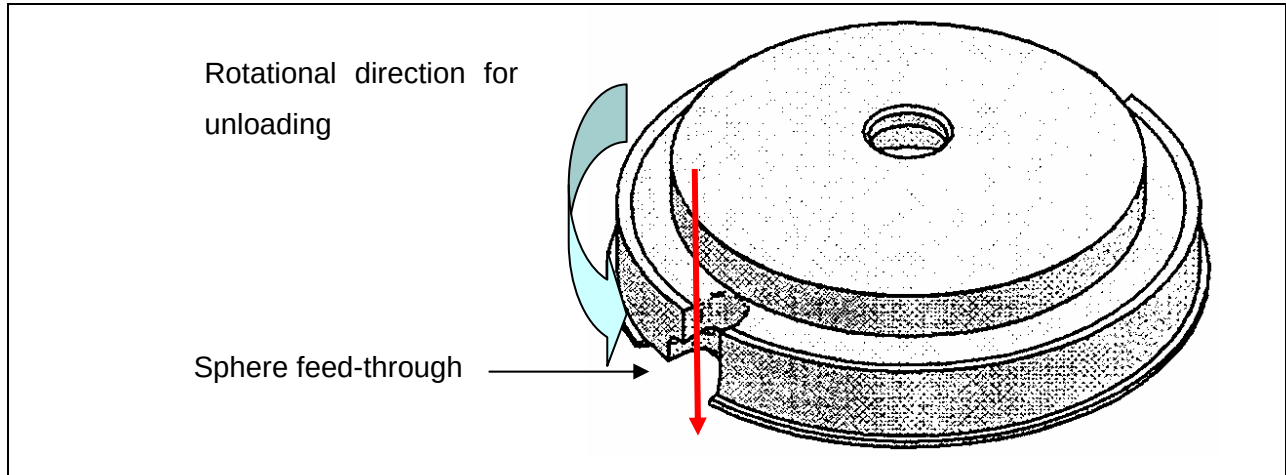


Figure 23: TUD head (first iteration)

The head has the following properties:

- Head outer diameter (D_{osp}) of 720 mm;
- Pitch circle diameter of sphere exit (D_{sp}) of 640 mm;
- Shaft diameter (D_{sh}) of 80 mm;
- Mass of 224 kg for the head (m_{sp});
- Mass of 37 kg for the shaft (m_{sh}).

Figure 24 is a three-dimensional model of the housing of the tank unloading device. The spheres that are located in the feed-through holes of the head are taken with the rotating head until the sphere is situated over one of the holes in the housing where the sphere will fall through the housing into a sphere transport pipe.

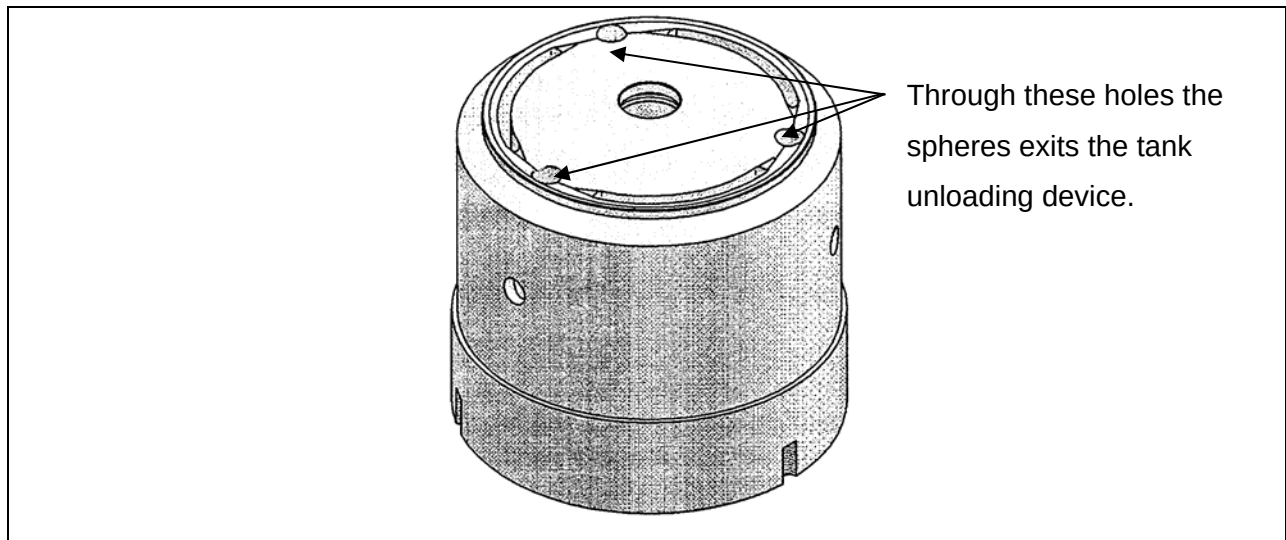


Figure 24: TUD housing

The disadvantage of this concept is that fuel spheres can be damaged due to spheres being caught between the separating (or singularization) mechanisms. Mechanical separation acting in the form of a scissor is used to separate spheres and this scissor action will damage the fuel spheres when a sphere is caught between the separating mechanisms.

Striking of spheres in the scissor mechanism can occur when the head rotates too fast and the sphere cannot pass through in time before the gap closes, or when the head rotates too slowly and more than one sphere fall through, causing the last sphere to be clamped between the head and the housing, as indicated in Figure 25. With broken spheres in the tank that will also pass through the tank unloading device the risk of blockages increases.

The timing of the unloading machine is based on a sphere that must be delivered into the sphere transportation pipe every 10.8 seconds. It is required to deliver a minimum of 333 spheres per hour which is one sphere every 10.8 seconds:

- $3600 \text{ s} / 333 \text{ spheres per hour} = 10.8 \text{ s}$ (1)

The housing in Figure 24 shows three holes, where a sphere must be delivered every 10.8 seconds per hole. (During redistribution only one hole will be used and during reactor refilling all three holes will be used. The required rotational speed is:

- $60 \text{ s} \cdot \text{min}^{-1} / 10.8 \text{ s} = 5.6 \text{ min}^{-1}$ (2)

The unloading device will be developed for a speed of 10 min^{-1} to be able to deliver more than the required minimum of 333 spheres per hour. When fewer spheres are required the tank unloading device can be operated in a stop-start operation to fill and maintain a sphere buffer in the fuel handling system.

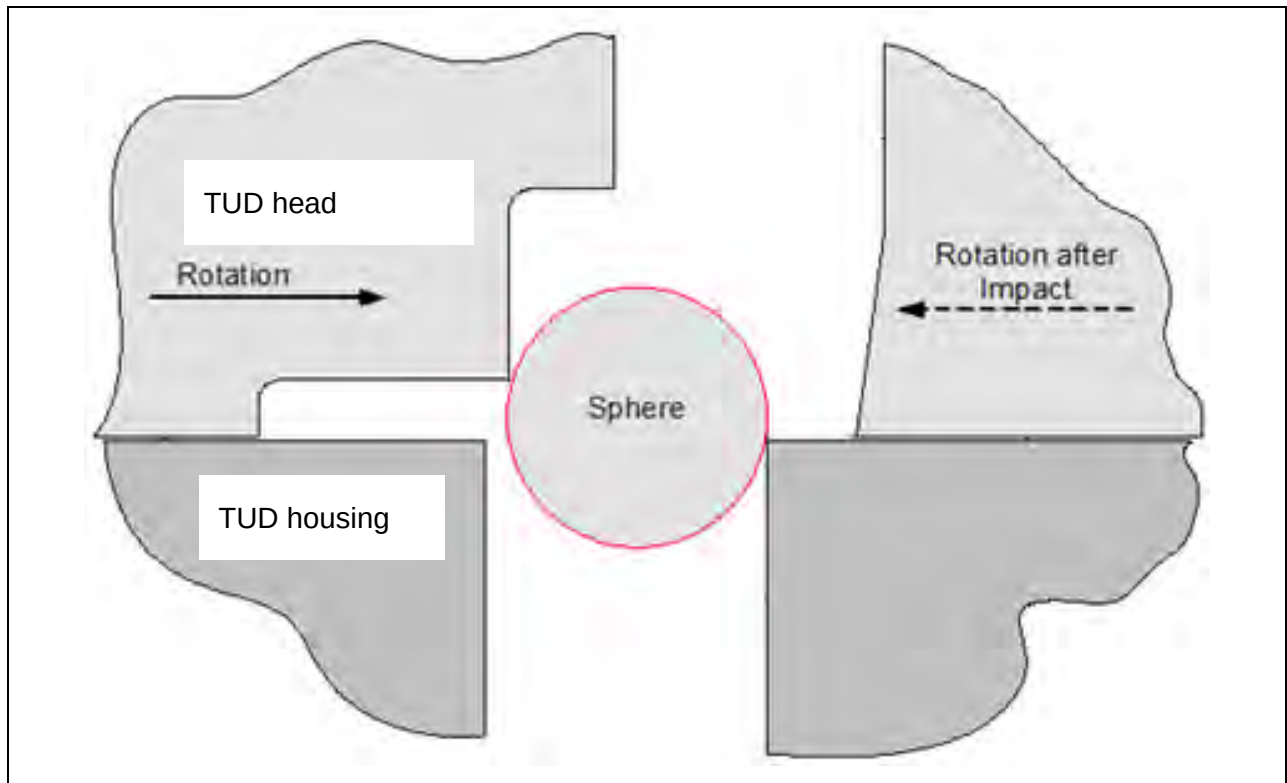


Figure 25: Clamped sphere

4.2.2 DETAIL DESIGN CALCULATIONS

Tests performed with a scale model during a previous studie indicated that the present head design must rotate approximately 10 min^{-1} over the housing exit for the unloading of spheres without damage to the spheres, Figure 24. Should the head rotate slower, there is a chance that more spheres would exit while the head hole is aligned with the housing hole. If the head rotates faster, the single sphere that is in the process of being unloaded would be caught between the head and the housing.

The TUD needs to start at a certain point to allow for acceleration to correct min^{-1} before passing the exit hole. This is required to prevent unsuccessful sphere unloading or even damage to the spheres.

Figure 26 depicts the positions of these unloading holes. The darker rings, spaced at 120° , are the exit holes in the housing, while the two lighter rings, spaced at 180° , are the holes in the head with the head in the parked position. The lighter broken-lined rings indicate the position of the head, rotated 19° clockwise from the start position, where the head must have reached the required rotational speed of 10 min^{-1} before the head reaches the exit hole of the housing, to ensure successful sphere unloading.

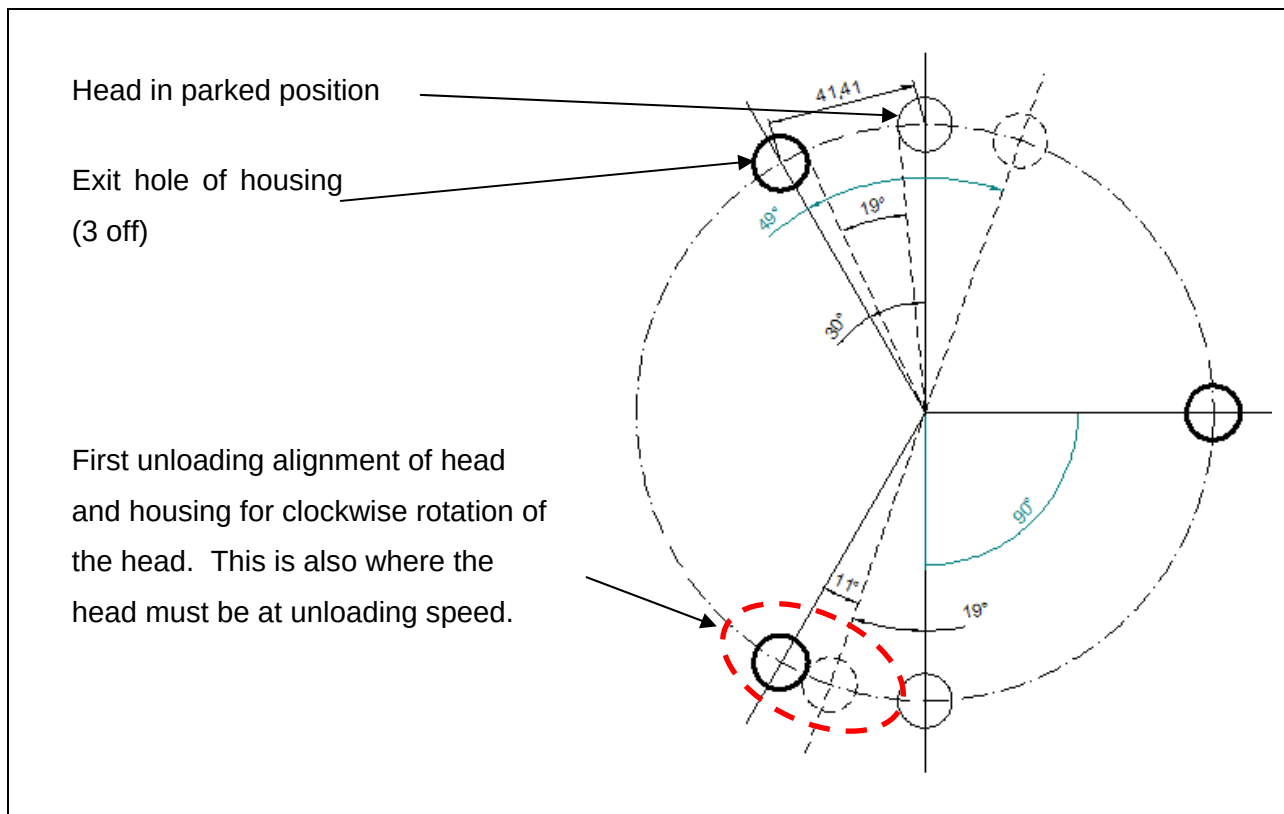


Figure 26: Head and housing park alignment

A drive was developed which provides enough torque to overcome the friction of the spheres on the head and to accelerate the head from the indicated position in Figure 26 to the required speed of 10 min^{-1} .

The first step is to calculate the moment of inertia of the head. The head is divided in two sections, as indicated in Figure 27.

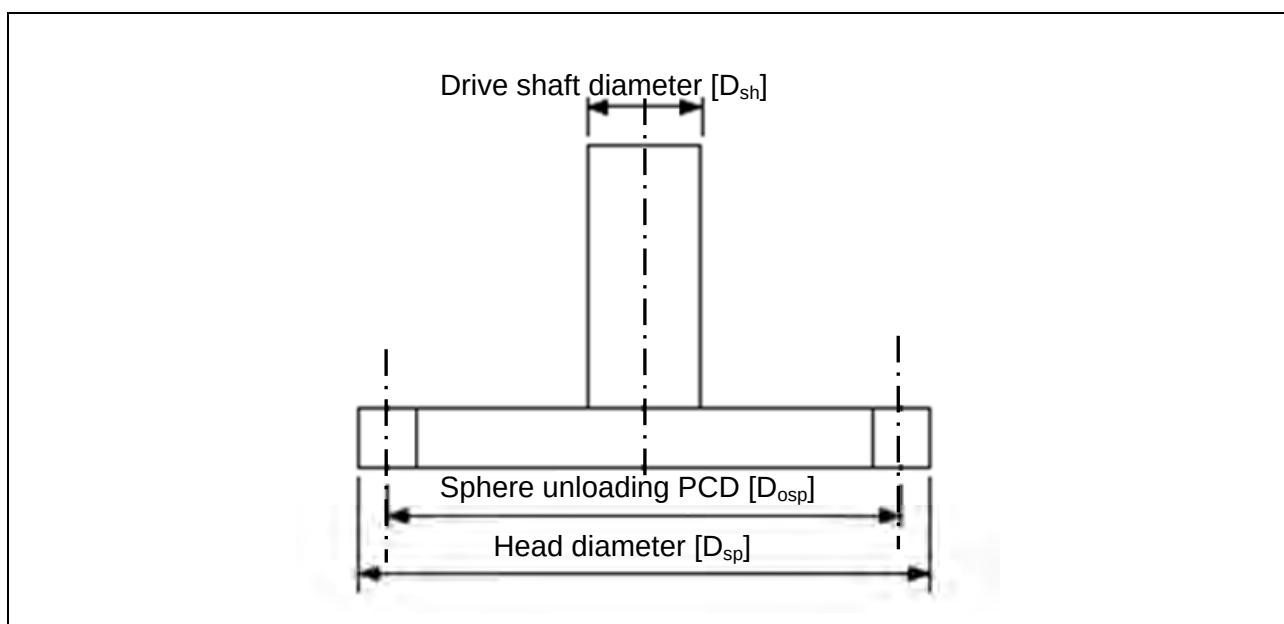


Figure 27: Head dimensions for calculations

The moment of inertia of the head is then calculated [28]:

- The equation for the shaft [28]: $J_{sh} = 0.5 \cdot m_{sh} \cdot (D_{sh}/2)^2$ (3)

- The equation for the head disk [28]: $J_{sp} = 0.5 \cdot m_{sp} \cdot (D_{osp}/2)^2$ (4)

- With the head rotating at 10 min^{-1} (or 1.047 rad/s) the head has a total moment of inertia (J_t) of $14.545 \text{ kg} \cdot \text{m}^2$.

The second step is to determine the torque required to accelerate the head against the friction of the spheres on the head. The head must be accelerated from standstill to 10 min^{-1} within 19° , refer to Figure 26.

With $\omega_{sp} = 1.047 \text{ rad/s}$ the acceleration is calculated as 1.66 rad/s^2 with the equation, [28]:

- $\alpha_D = (\omega_{sp}^2 - \omega_0^2) / (2 \cdot \theta_D)$ (5)

The torque required for accelerating the head from the stationary position to 1.047 rad/s is 24 Nm . This value excludes the friction of the fuel spheres on the head. The torque is calculated from [28]:

- $T_D = J_t \cdot \alpha_D$ (6)

The maintenance pipe above the head prevents spheres the overload the head. Refer to Figure 1 for the tank layout. However for the calculations a conservative height of 1 m will be used to calculate the friction on the head.

This column is however not a solid column because the spheres have open spaces between them. The column weight is calculated by a packaging factor. The packaging factor is a function of the volume in which the spheres are. If the spheres are packed in a cube the packaging factor can be calculated by using the 60° angles in which they will stack themselves. The packaging factor can be calculated using equation 7, although it is based on standard shapes [29]:

- $\eta = \frac{\pi}{3\sqrt{2}} \approx 0.74$ (7)

In this case there is the round tank shell with the inner tube and an inner support ring which changes the packaging factor. With the use of Figure 28, a scale drawing of the tank filled with scaled spheres, the packaging factor was calculated. On this drawing the spheres were counted and their total volume calculated. The filled volume of the tank was then calculated. Only a quarter of the tank was used and then multiplied by four to get the total volume. This was done for both a top view and a bottom view. For the top view a packaging factor of 0.49 was calculated and a packaging factor of 0.51 was calculated for the side view, thus approximately 0.5 .

A packaging factor of 0.5 means that half the tank is filled with spheres and the remainder filled with gas. It has however been assumed that this value is very low against the given value of

0.74 by Wolfram Mathworld, reference [29]. It has thus been decided to use a value of 0.6 for the packaging factor, which will also cause a heavier mass on the head for more conservative calculations than a packaging factor of 0.5.

Due to the geometry of the vessel the packaging factor of the spheres is assumed to be approximately 0.6 on the head. Thus, the volume of the spheres positioned on the head is calculated by multiplying the volume of the solid graphite column by 0.6 to get the mass of the graphite sphere column. The weight on the head is then the volume times the density of graphite (1740 kg/m^3) times the force of gravity ($g=9.81 \text{ m/s}^2$), which gives the weight (W) as 3294 N on the head.

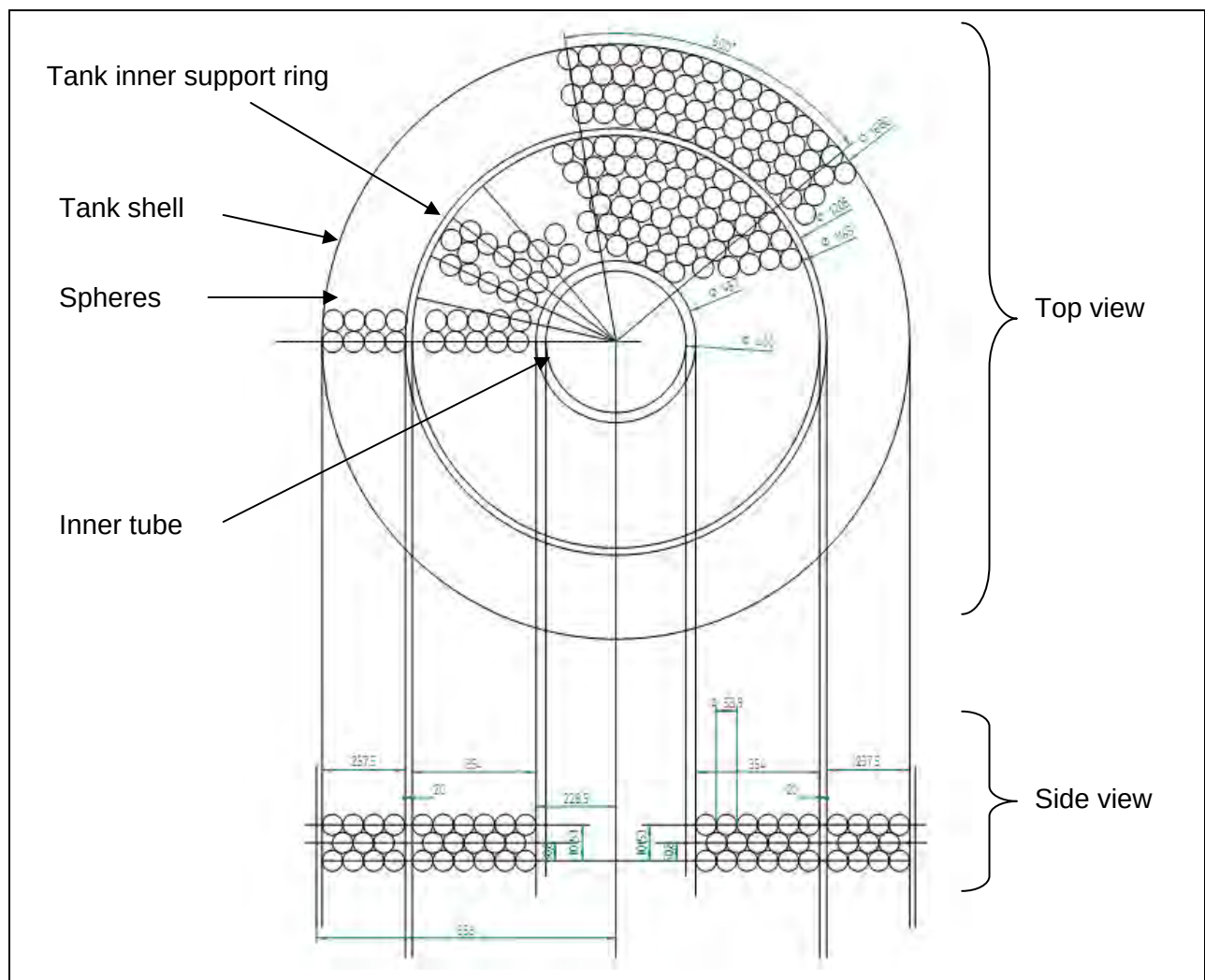


Figure 28: Scale drawing of tank filled with spheres

The torque required to accelerate the head due to the friction of the spheres on the head is calculated with the force of the spheres on the head. The force is calculated with the formula, [28]:

- $F\mu = W \times \mu$.

(8)

The pressure (p_μ) on the head is a function of the force on the head and the area of the head where the force is applied. The torque required to turn the head with the spheres on the head is then calculated with equation 9, [28]:

- $T_\mu = r \times p_\mu$. (9)

However, the since the radius of the head is increasing, starting at the head centre and then increases to 320 mm, the torque can be calculated by integrating over the radius of the head top area, as indicated in Figure 29.

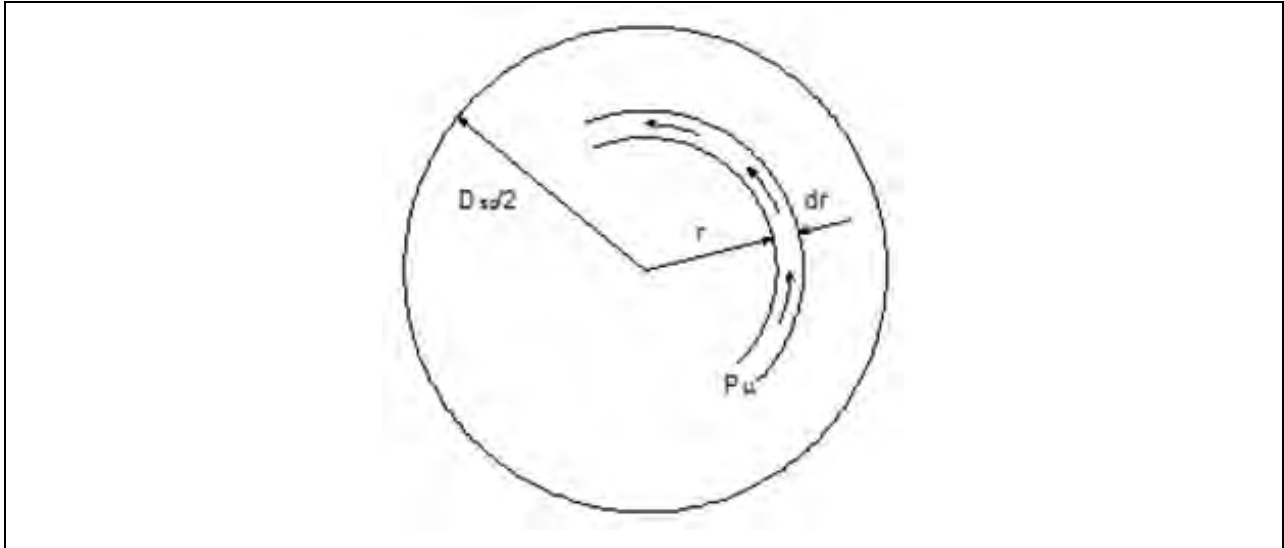


Figure 29: Integrating over the radius of the head

The equation that is used to calculate the torque is:

$$T_\mu := \int_0^{D_{sp} \cdot 0.5} r \cdot (\pi \cdot r \cdot p_\mu) dr \quad (10)$$

The torque required against the friction is calculated as 211 Nm by equation (10).

- $\alpha = \frac{\omega_{sp}^2 - \omega_0^2}{2 \cdot \theta}$; where $\alpha = 1.65 \text{ m/s}^2$ [28] (11)

- $J = 0.5 \cdot m \cdot \left(\frac{D}{2}\right)^2$; where $J = 14.545 \text{ m}^2 \cdot \text{kg}$ [28] (12)

- $T = J \cdot \alpha$; where $T = 24 \text{ Nm}$ [28] (13)

The total torque required for accelerating the head can be found by adding the accelerating torque and friction torque values: $24 \text{ Nm} + 211 \text{ Nm} = 235 \text{ Nm}$. (14)

The next step is to determine what the impact force of the rotating head is on the sphere if a sphere is clamped. There are three conditions applicable to the head:

- First is the torque required to accelerate the head;
- Second, to maintain head optimum rotational speed;
- Third is the rotating head striking a sphere.

The force on the sphere that is now clamped between the head and the housing, as indicated in Figure 25, is a function of the momentum of the head as well as the stall torque force of the actuator applied to the rotation of the head. A conservative approach is to do this calculation without spheres on the head, because the spheres will have a dampening effect on the impact of the head on the sphere, due to the friction brake between the rotating head and the spheres on top of the head.

It is required to rotate the head with only enough torque to overcome the friction in the rotating parts of the unloading mechanism. However, to accelerate the head from its parked position a high torque is required. Therefore the head is driven through a variable electromechanical torque limiter. When the head is to be accelerated the torque limiter is set to a high setting enough to overcome the full friction of spheres on the head together with the torque required to accelerate the head.

When the head speed is reached the torque is reduced to a torque setting just above the torque required to overcome friction on the head since acceleration is not required. According to calculations a torque setting of 211 Nm corresponds with the friction, therefore a torque limiter with a maximum torque setting of 313 Nm has been selected. This selection is the closest standard for off-the-shelf torque limiters to the required minimum of 235 Nm, thus the torque limiter will be operated at a maximum of 235 Nm at acceleration of the head, and then reduced to 211 Nm after acceleration.

When the head strikes a miss fed sphere, the force of the actuator is limited to a minimum required force for rotating the head. However, the impact has been simulated mathematically to predict what is going to happen at impact. In paragraph 8 of Appendix A it was calculated that an impact occurring in a time duration equal to or less than 0.003 s will result in a force higher than 18 kN, enough to break the sphere. This force is the result of the momentum of the rotating head.

It can be concluded that the momentum of the head alone is enough to cut the sphere at impact. The conservative assumption is that there are only a few spheres on the head, thus friction between the head and the spheres can be ignored at the time of impact, which causes the highest force on the sphere due to the absence of the friction brake between spheres on the head and head itself. It has therefore been decided to introduce an impact limiter to the design to protect the sphere.

a. Main actuator components

The characteristic of the actuator that rotates the head is required to design the impact limiter. Based on the 235 Nm, as calculated above, and 10 min^{-1} rotational speed, an actuator drive was selected. The following components are required for the actuator:

- Actuator drive;

- Reduction gearbox;
- Angle sensor;
- Torque limiter.

Table 4 contains the torque requirements, as calculated, for the functions to be performed by the actuator. Approximate values were calculated and based on these values an actuator was selected, as given below.

The actuator must be able to turn in both clockwise and anti-clockwise directions to be able to release a sphere.

Table 4: Torque requirements

Function	Torque required	Value
Start-up of head rotation	Start rotational movement and overcome friction	235 Nm
Operational head rotation	Overcome friction	211 Nm

Figure 30 shows a cutaway diagram of a hollow shaft actuator. The actuator was selected from the *Harmonic Drive* catalogue as an actuator-gearbox combination [30]. The hollow shaft will be used to drive the angle sensor for the head.

An actuator with a 1:160 harmonic drive reduction gearbox was selected from the FHA-40B series catalogue. The 1:160 gearbox could be an option, but the standard 1:100 gearbox could not deliver the rated torque of 210 Nm. If the smaller reduction ratio gives 69 Nm and the higher ratio gives 137, the 1:160 would give 219 Nm by interpolation, as given in Table 5. This would give 219.3 Nm for operational rotation and a 1200 Nm for start-up. When the gearbox rotates at 10 min^{-1} the actuator motor would be running at 1600 min^{-1} .

Table 5: Gearbox ratio (Information obtained from the *Harmonic Drive* catalogue, [30])

	UNIT	FHA-40B-5036	FHA-40B-2536	FHA-40B-XX36
Rated output power	W	360	360	360
Rated output torque	Nm	69	137	219.2
Max. output torque	Nm	402	750	1200
Max. continued stall torque	Nm	78	157	251.2
Rated output speed	min^{-1}	50	25	15.625
Max. output speed	min^{-1}	70	35	21.875
Speed ratio		50	100	160
Rated motor speed	min^{-1}	2500	2500	2500
Max. motor speed	min^{-1}	3500	3500	3500

The actuator is a 24 V(AC) servomotor with controller, also supplied by *Harmonic Drive*.

The encoder build into the actuator assembly will be used as verification for the slip of the torque limiter. The angle sensor that is fixed to the head and the encoder that is on the actuator assembly must give the same rotation speed. A difference in their rotation speed would indicate 'slip' in the torque limiter.

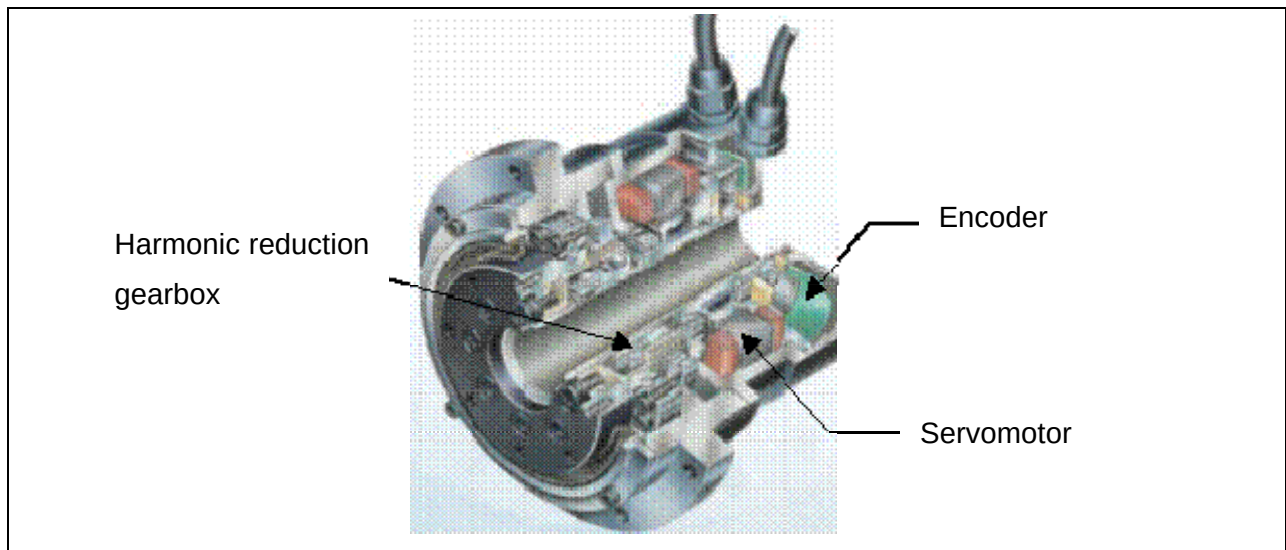


Figure 30: FHA-40B actuator (from the *Harmonic Drive Catalogue*, [30])

There are various options available for the torque limiter. The mechanical torque limiters are pre-set at a fixed setting. There are the electromagnetic clutches where the setting can be altered during operation. The option that the torque limit can be changed during operation makes this clutch the preferred choice because the torque limiter can be adjusted to a high-torque setting for start-up and then the torque setting can be lowered during operation to overcome the friction of the system.

Table 5 gives the maximum torque of the actuator as 1200 Nm. The torque limiter must have a higher torque setting to be able to use the full potential of the actuator.

From the *Heid Antriebstechnik Catalogue* [32], an electromagnetic multi-disc clutch was selected. It is possible to operate the clutch at various torque settings by controlling the present supply to the clutch coil. The coil of the clutch is operated at 24 V(DC). The FM 80 clutch from the FM series has a maximum torque setting of 1250 Nm.

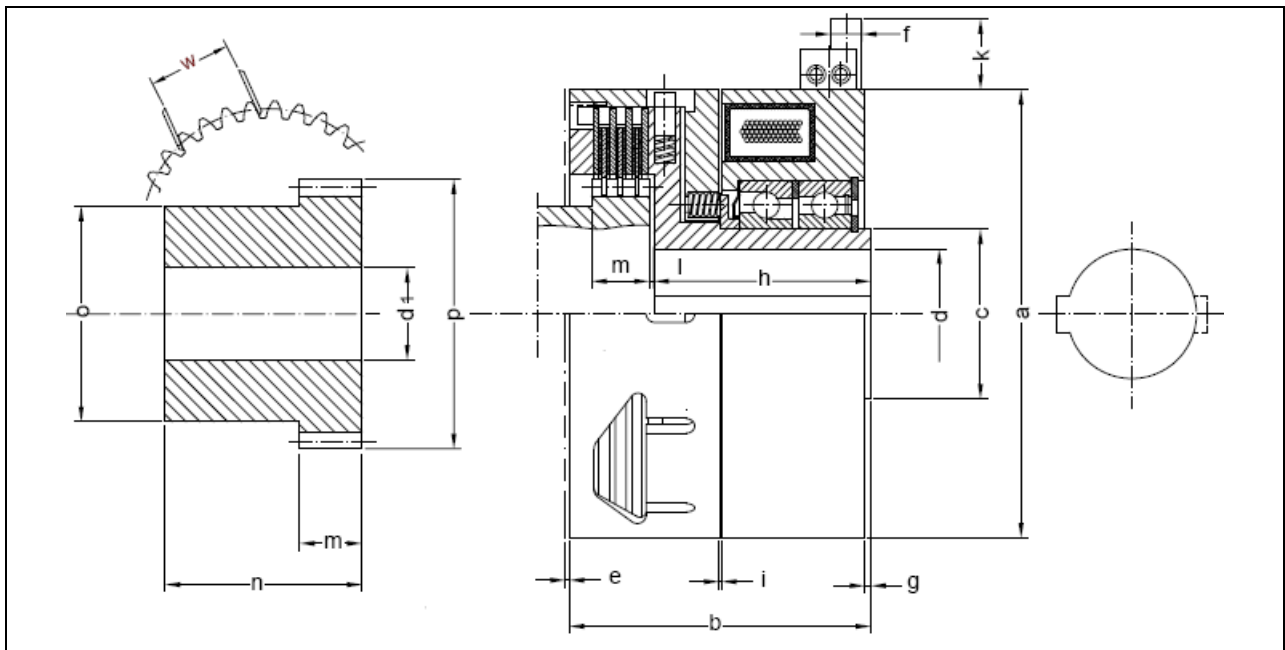


Figure 31: Torque limiter (Obtained from the *Heid Antriebstechnik* Catalogue, [32])

In order to stop the head at a specific position the position of the head must be known, therefore the need for an angle sensor. It is proposed that the angle sensor should be mounted at the bottom of the actuator. The drive shaft of the angle sensor goes through the hollow shaft of the actuator and is mounted in the hollow shaft of the encoder.

A single-turn encoder was chosen because the position of the head is important for every rotation. The angle can be zeroed after installation. Figure 32 is a representation of the angle encoder.

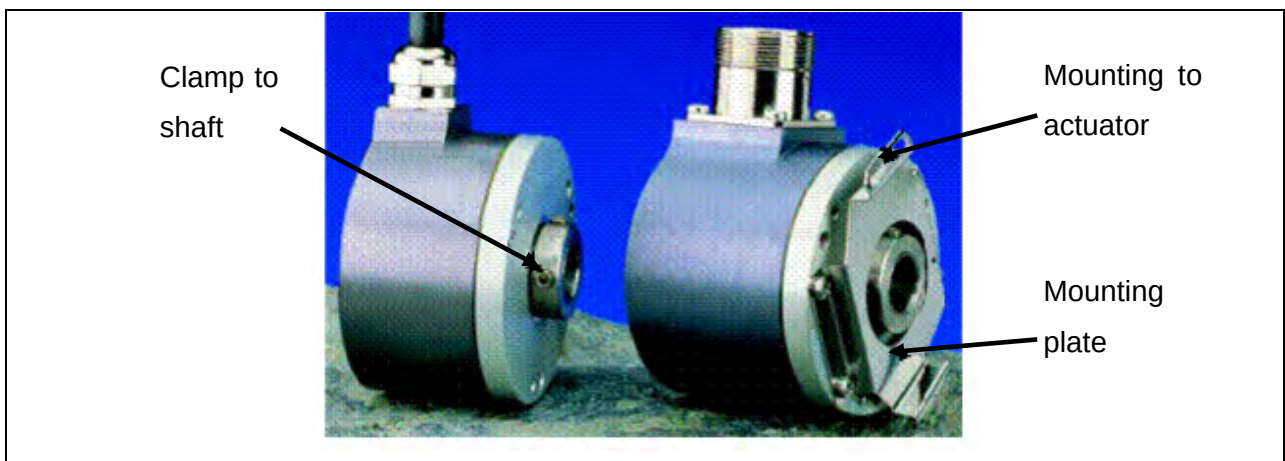


Figure 32: *Leine & Linde* 567 Angle Encoder (from the *Leine & Linde* Catalogue, [31])

b. Impact-limiter: metal spring

A very dynamic situation will develop when the head hits a miss fed sphere. The force on the sphere is a function of the impact time. When the head strike the sphere the drive

torque from the actuator is increased until a maximum torque of 1200 Nm (from Table 5) where after the actuator drive torque falls to the continuous torque setting of 251 Nm (from Table 5). The actuator is thus still driving the head into the sphere when the head has come to a standstill after full depression of the spring. To demonstrate what was happening a mathematical model was developed, Appendix A, and is described in the next section.

Since the impact on the sphere is a function of the deceleration of the head, the algorithm calculates the incremental speed according to the deceleration that occurs. The velocity of the head is calculated with equation 15:

$$\bullet \quad \omega_i = \omega_{i-1} + \alpha_{i-1} \cdot \Delta t \quad [28] \quad (15)$$

With the velocity after time increment 1, the new position of the head is calculated with equation 16:

$$\bullet \quad x_i = x_{i-1} + \omega_i \cdot \left(\frac{D^{0.5}}{m}\right) \cdot \Delta t \quad [28] \quad (16)$$

The new position of equation (16) is then used to calculate the force on the sphere with a spring k-value 1.

The algorithm methodology with a spring is described in paragraph 9.2 of Appendix A. Each time the model was run, a different k-value for the spring was used to determine the effect of the spring k-value on the system under impact. Table 6 shows that by increasing the k-value of the spring, the force on the sphere increases, which corresponds with the spring equation:

$$\bullet \quad F = k \times \Delta x. \quad (17)$$

The lower k-value provides better protection for the sphere because it increases the time of impact. The resulting force, together with the spring displacement, is then used as a direct input for the spring design.

Table 6: Simulation results of head reactions after impact with different k-values

Spring	Maximum spring compression		Maximum head return		Maximum force on sphere	
k-value [kN/m]	Time [s]	Distance [mm]	Time [s]	Distance [mm]	Time [s]	Force [N]
2000	0.01361	2.9634	0.05127	-2.5382	0.01364	5.9268
4000	0.00951	2.0665	0.0446	-3.2845	0.00957	8.2663
6000	0.00775	1.6769	0.04156	-3.608	0.00775	10.061
8000	0.00675	1.4468	0.03991	-3.799	0.00669	11.575
10000	0.00597	1.2908	0.03895	-3.9284	0.00601	12.908
12000	0.00545	1.1761	0.03811	-4.0235	0.00547	14.114
14000	0.00505	1.0873	0.03741	-4.0971	0.00507	15.222
16000	0.00474	1.0159	0.03684	-4.1562	0.00473	16.254
18000	0.00442	0.95674	0.03651	-4.2052	0.00448	17.222

Coil springs have to be held in position by a mechanical fixture, while a plate spring can be machined as part of the mechanical fixture and will therefore not be removable from its position like a coil spring. Therefore a plate spring design was selected to limit the possibility of failed or dislodged items between the fuel spheres. A recovery plan must now be developed for the replacement of the failed spring.

However, calculations indicated that the plate spring could not provide the impact limitation function without being pre-loaded. The plate was then pre-loaded in series with the parallel-configured coil spring assembly. A graphical diagram was drawn where the variables for the available space was identified, as indicated in Figure 33. The spring assembly is fixed to the housing while the head rotates above the spring assembly.

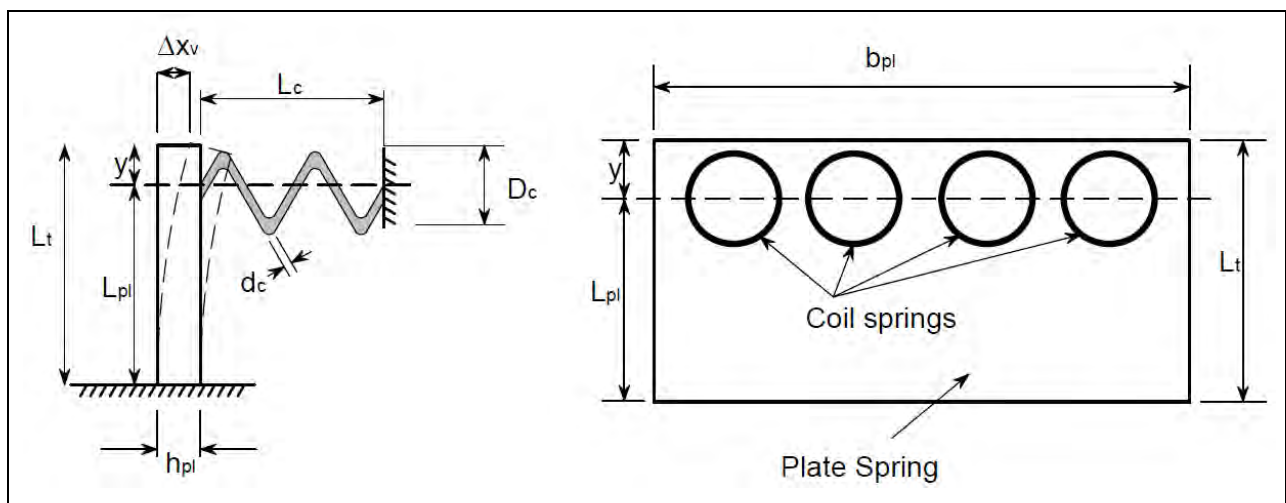


Figure 33: Coil spring configuration

At this stage a three-dimensional model of the spring had to be developed to ensure that there was enough space to fit the spring assembly. The result of the spring design is shown in Figure 34. This geometry is the maximum size that can be fitted into the present unloading concept.

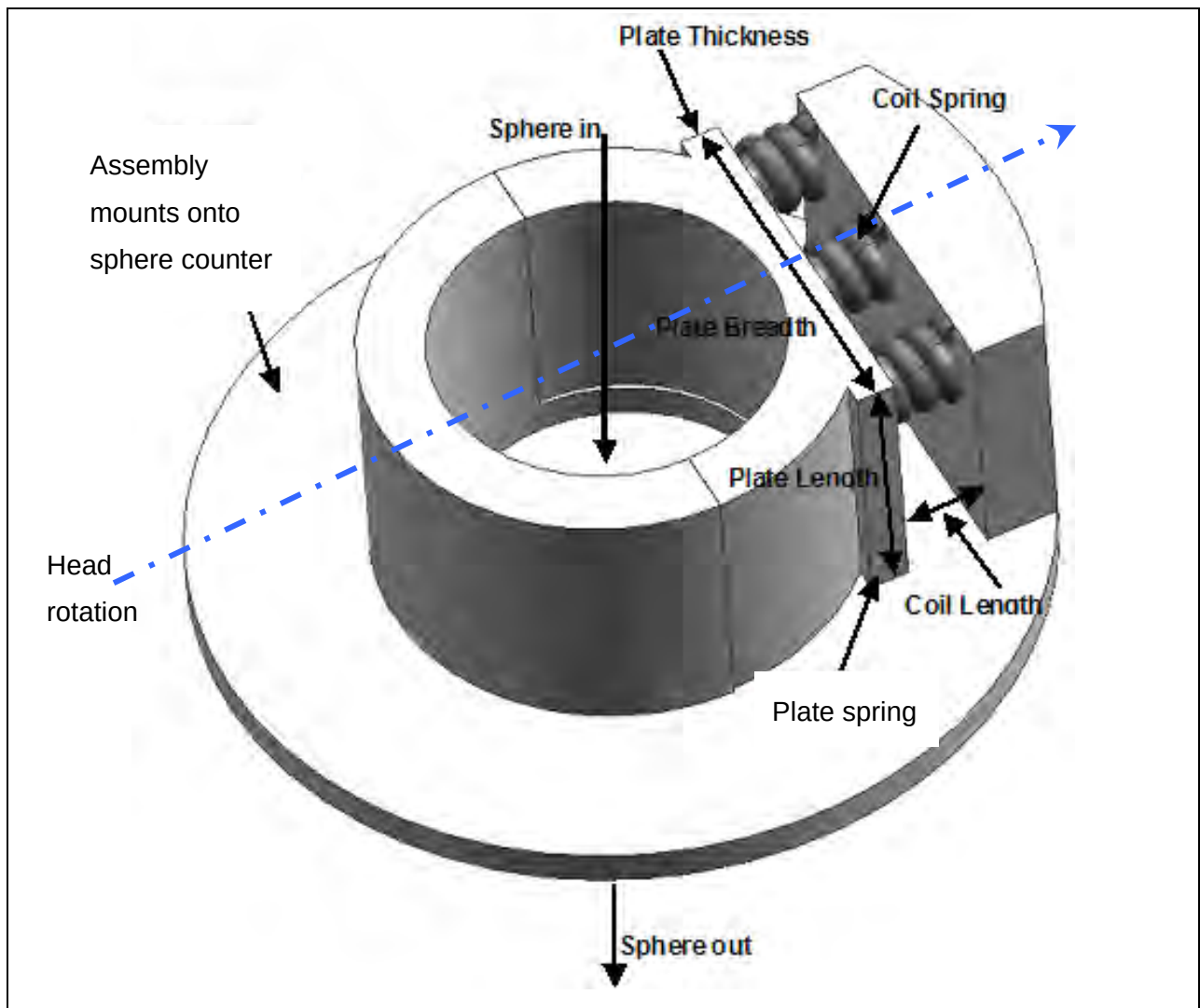


Figure 34: Three-dimensional model of the metal spring

Table 7 lists the characteristics of the spring that is fixed due to the layout of the unloading machine and coil springs.

Table 7: Fixed plate spring measurements

Variable	Value
Maximum plate height	50 mm
Maximum plate thickness	Function of coil spring length
Maximum plate width	90 mm

Relationships are used to position the springs in the highest possible position on the plate spring. The relations in Table 8 are used to create the maximum plate length.

Table 8: Coil spring measurements

Relation	To achieve
Assume spring diameter	e.g. $D = 12$
$y = D/2$	Coil radius
$L_{pl} = L_t - y$	Coils positioned at plate spring top:
Coil diameter: function of plate width and number of coils	To automatically space coils equally
Minimum spring index C	$C = 6$ (reference [28] page 360)
Spring wire diameter	$d = D/C$, select first smaller wire size
Adjust spring diameter D	With selected wire thickness d
Get number of coil springs in available space	Number = plate width / coil diameter

The dimensional relations were used to calculate the k-value of each spring. From the algorithm a k-value for the spring assembly is obtained. The spring assembly can now be designed to give this k-value. The spring must be designed to lower the possibility of fatigue failure because it is a major maintenance task to replace a failed spring and a failed spring can cause damage to the spheres.

The spring assembly will now be developed followed by the fatigue calculations as a check on the design. Should fatigue be possible according to the calculations the design will be changed again.

A k-value ratio exists between the coil spring assembly and the plate spring. The plate spring and coil springs group is arranged as a parallel spring assembly. The ratio between the two is according to equation 18:

$$\bullet \quad k_{as} = k_{pl} + k_{ca} \quad [28] \quad (18)$$

The deflection and force results of the algorithm are then used to test the different spring geometries. For each k-value for the algorithm, a k-value ratio adjustment can be used to optimize the spring assembly. Loop 2 of Figure 35 is used to adjust the k-value of the spring assembly Figure 34 as input to the algorithm. Loop 1 of Figure 35 adjusts the ratio between plate spring and coil spring.

The conditional statement for Loop 1 is to balance the k-value between the plate spring and coil spring group to obtain an optimum balance between the maximum bending stress in the plate spring (Paragraph A9.3.3.1 of Appendix A) and the coil spring failure (Paragraph A9.3.4.1 of Appendix A). The conditional statement for Loop 2 is to adjust the spring assembly k-value until the conditions of Loop 1 can be met.

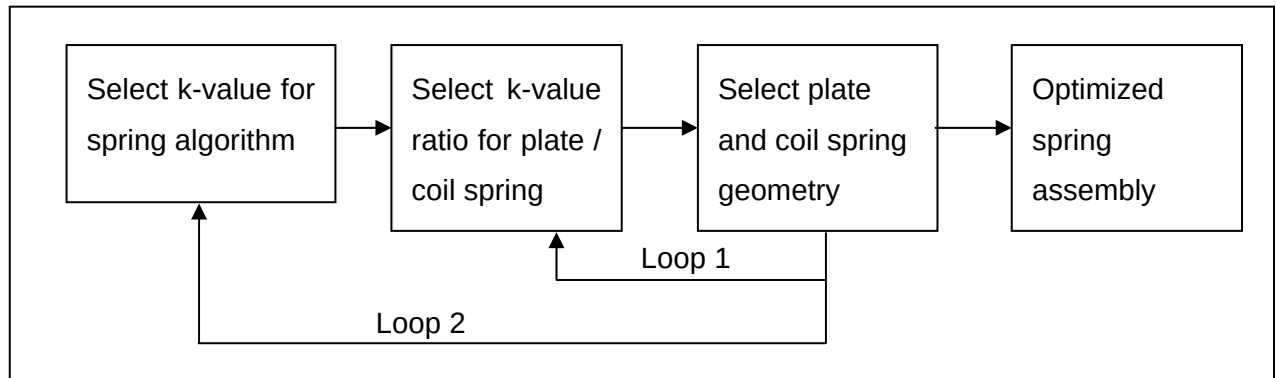


Figure 35: Variable selection options

Table 9 lists the findings when the ratios were adjusted.

Table 9: Results on variable settings for fatigue calculations

Variable	Effect	
	Higher	Lower
k-value input to algorithm	Higher force Lower travel	Lower force Higher travel
Plate height, L_t	Higher deflection	Stiffer spring
Plate thickness, h_{pl}	Less deflection Higher stress	Lower stress More deflection
Coil spring diameter, D	Less movement space	More movement space
k-value ratio: plate / coil spring		

The mathematical model described in Appendix A was used to capture data on spring performance. The results of these tests are displayed in Table 10 and Table 11 below. The coil spring was fixed in the design at a diameter of 12 mm, wire diameter of 2 mm, and 7 springs with 3 coils per spring. The k-value for the spring pack is 215.02 kN/m, with 30.72 kN/m for each coil. The k-value for the plate is calculated as 2285 kN/m.

Table 10: Spring simulation results

Spring	Max spring compression		Max head return		Max force on sphere	
	Time (s)	Distance (mm)	Time (s)	Distance (mm)	Time (s)	Force (kN)
k-value						
500	0.028	6.21	0.081	0.18	0.028	3.10
1000	0.020	4.27	0.062	-1.45	0.020	4.27
1500	0.016	3.45	0.055	-2.14	0.016	5.17
2000	0.014	2.96	0.051	-2.54	0.014	5.93
4000	0.010	2.07	0.045	-3.28	0.010	8.27
6000	0.008	1.68	0.042	-3.61	0.008	10.06
8000	0.007	1.45	0.040	-3.80	0.007	11.58
10000	0.006	1.29	0.039	-3.93	0.006	12.91
12000	0.005	1.18	0.038	-4.02	0.005	14.11

Table 11: Spring simulation results (continued)

Spring	Coil spring failure		Plate bending stress	Plate thickness	Spring material space filling	Total space required
k-value	t = s	tm=Sy	MPa	mm	mm	mm
500	yes	no	1533	1.96	7.96	14.17
1000	no	no	1480	2.75	8.75	13.02
1500	n	no	1407	2.34	8.34	11.79
2000	no	no	1350	3.61	9.61	12.57
4000	no	no	1209	4.64	10.64	12.71
6000	no	no	1130	5.35	11.35	13.03
8000	no	no	1077	5.91	11.91	13.36
10000	no	no	1067	6.37	12.37	13.66
12000	no	no	1005	6.78	12.78	13.96

Table 12 gives the results where the coil spring was fixed at a diameter of 8 mm, wire diameter of 1.3 mm, and at 11 springs of 3 coils each. The k-value for the test spring pack is 219.63 kN/m, and 19.066 kN/m for each coil.

Table 12: Spring simulation results (continued)

Spring	Coil spring failure		Plate bending stress	Plate thickness	Spring material space filling	Total space required
k-value	t = s	tm=Sy	MPa	mm	mm	mm
500	yes	yes	1525	1.95	5.85	12.06
1000	yes	no	1477	2.74	6.64	10.91
1500	no	no	1405	3.24	7.14	10.59
2000	no	no	1349	3.61	7.51	10.47
4000	no	no	1209	4.64	8.54	10.61
6000	no	no	1130	5.35	9.25	10.93
8000	no	no	1076	5.9	9.80	11.25
10000	no	no	1036	6.37	10.27	11.56
12000	no	no	1005	6.78	10.68	11.86

The smallest spring assembly space requirement in Table 12 is where the spring assembly has a k-value of 2000 kN/m, where 10.47 mm is required to position the spring assembly. Of this 10.47 mm the spring plate takes up 3.6 mm (plate thickness h_{pl} as indicated in Figure 36) and the compressed coil a length of $3 \times 1.3 = 3.9$ mm, totalling hardware that takes up 7.5 mm of the 10.47 mm. According to fatigue calculations the coil springs will not fail, but the bending stress on the plate spring is very high. The high bending stress will require

special materials, for example the yield stress of EN24 steel is only 650 MPa according to *Shelley Steels* [33].

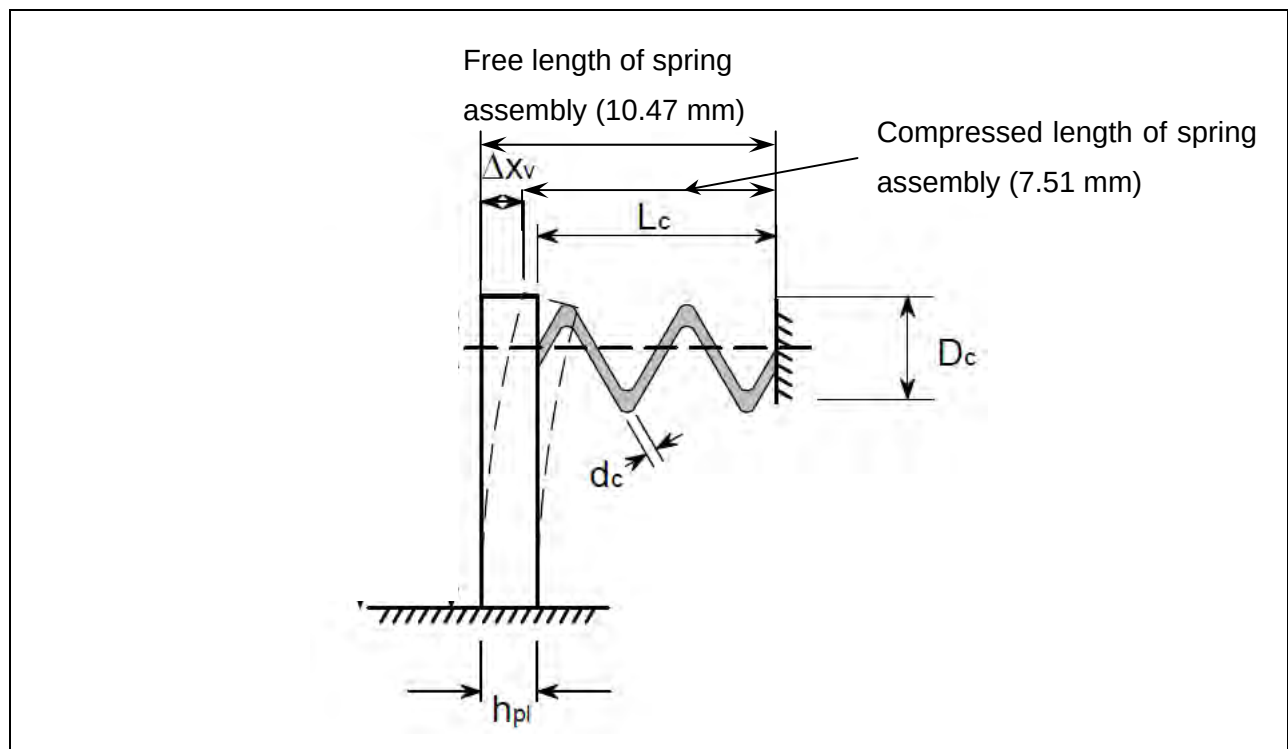


Figure 36: Spring installation space requirement

The metal spring works, but the bending stresses in the materials are high. The high stresses make the design a marginal design and definitely not robust. Regular maintenance will be required which is expensive in the nuclear industry. It has therefore been decided to discontinue further development work on the metal spring concept.

c. Impact-limiter: silicone spring

To expand on the gravity concept the possibility of using a silicone spring was investigated. The silicone spring was placed in the same position as the metal spring. Figure 37 shows the three dimensional model of the silicone spring where the silicone is housed in a metal ring.

The silicone spring will not suddenly collapse as happened with the failure of the metal spring, but the silicone will degrade over time due to the radioactive environment, [23], [24]. Therefore the silicone must be replaced at certain maintenance intervals.

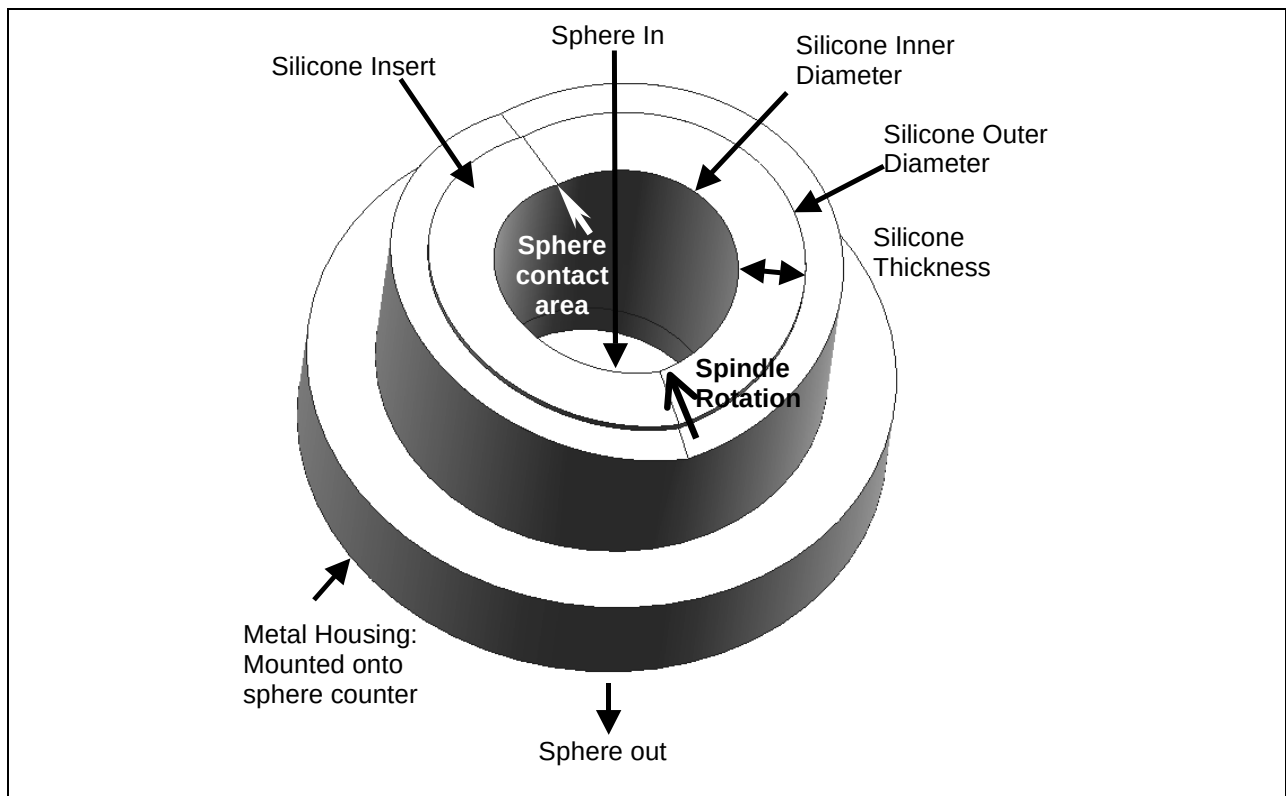


Figure 37: Three dimensional model for the silicone spring

The silicone spring increases pressure on the sphere as the silicone is being compressed, just as with the metal spring. The force on the sphere increases more with compression of the silicone than the metal spring. This is due to the increasing area of compression, as indicated in Figure 38, where the silicone takes on the form of the sphere and the contact area is increased.

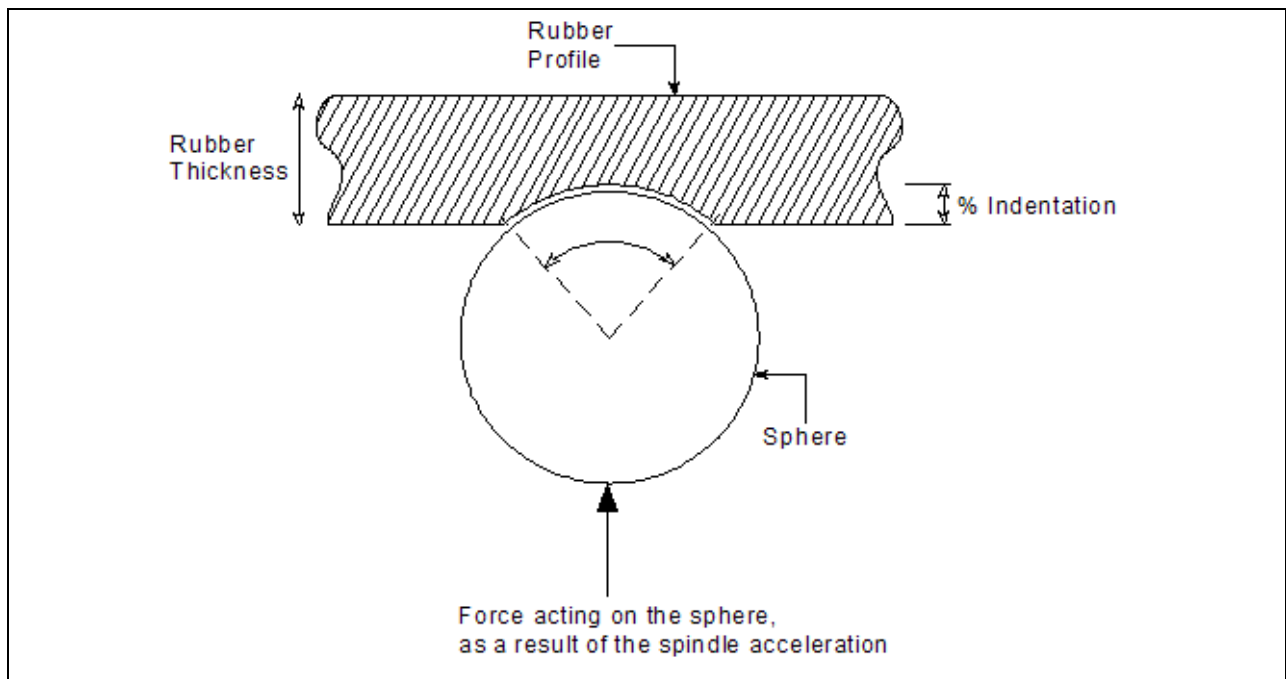


Figure 38: Sphere deforms silicone

The same algorithm that was used for the metal spring calculations was used for the silicone spring. However, in this case the formula was altered to make provision for the silicone properties and the increasing force on the spring due to the increasing contact area between the silicone and the sphere.

To calculate the increasing force on the sphere due to the increasing area of contact, a relation was required in the algorithm to take this phenomenon into account.

With reference to Figure 39, the increasing force on the sphere can be described mathematically with the following equation:

$$\bullet \quad A(\%C) = \frac{\pi \cdot \left(r \cdot \sin \left(\arccos \left(\frac{r - \left(\frac{\%C}{100} \cdot r \right)}{r} \right) \right) \right)^2}{4} \quad (19)$$

Equation 19 is derived from:

- The area of contact between the sphere and the silicone is:
 - o $A = (\pi \times d^2)/4$, (20)
 - o Where $d = 2 \times R \times \sin \theta$. (21)
- The increasing contact area is a function of the increasing indentation,
 - o $\Delta x = \text{sphere radius} - (\% \text{compression} / 100 \times \text{the rubber thickness}) \times \theta$. (22)
- The angle θ is calculated by:
 - o $\cos \theta = (\text{head radius} - \Delta x) / \text{sphere radius}$. (23)
- The resulting equation becomes equation 19
- The force on the sphere is calculated with the equation:
 - o $\text{pressure} = \text{force} / \text{area}$ (24)
- The percentage indentation is calculated as a function of Δx where the:
 - o $\text{indentation}\% = (\Delta x / \text{rubber thickness}) \times 100$ (25)
- The k-value of the silicone spring is calculated by the (force as a function of % indentation, %C, of the rubber again as a function of Δx) / Δx , or in formula format as
 - o $k(\Delta x) = \frac{F(\%C(\Delta x))}{\Delta x}$ (26)

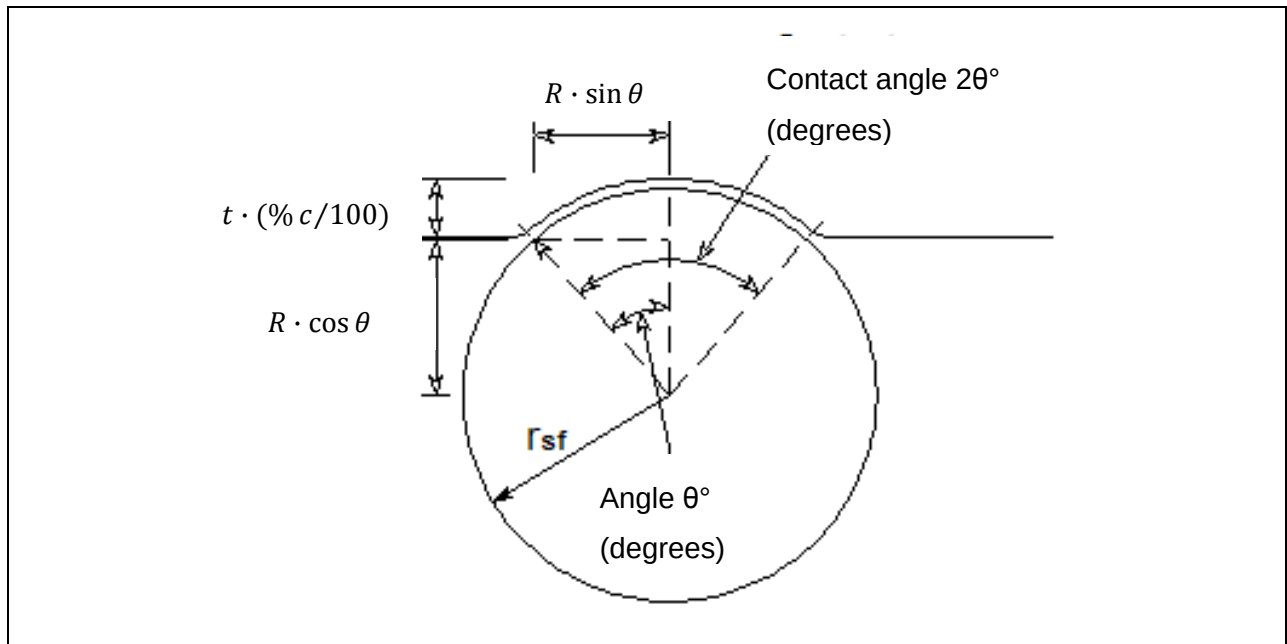


Figure 39: Mathematical representation of silicone deformation

A typical result is shown in Figure 40, which depends on the silicone hardness used.

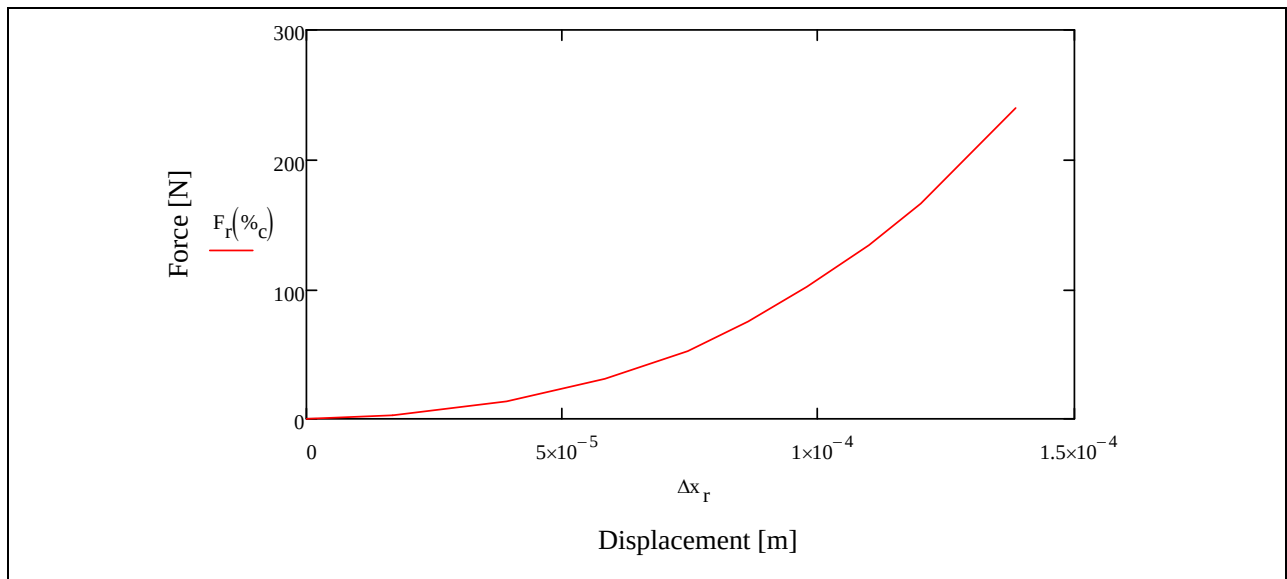


Figure 40: Silicone compression pressure increase

The algorithm was altered to calculate the k-value for the silicone spring. Figure 41 shows the improved algorithm. The mathematical calculations with this algorithm can be followed in Appendix A.

$$\begin{pmatrix} x_{sr} \\ \omega \\ \alpha \\ F_{srv} \\ Fact \end{pmatrix} := \begin{array}{l} \omega_0 \leftarrow \omega_{sp} \cdot s \\ x_{sr0} \leftarrow 0 \\ F_{srv0} \leftarrow 0 \\ F_{tl0} \leftarrow 0 \\ \alpha_0 \leftarrow 0 \\ i \leftarrow 0 \\ \text{while } i \cdot \Delta t_{step} < 0.17 \\ \quad i \leftarrow i + 1 \\ \quad \omega_i \leftarrow \omega_{i-1} + \alpha_{i-1} \cdot \Delta t_{step} \\ \quad \omega_i \leftarrow 0 \text{ if } |\omega_i| < 10^{-4} \\ \quad x_{sr_i} \leftarrow x_{sr_{i-1}} + \omega_i \cdot \frac{D_{sp} \cdot 0.5}{m} \cdot \Delta t_{step} \\ \quad F_{srv_i} \leftarrow x_{sr_i} \cdot k_r(x_{sr_i} \cdot m) \cdot \left(\frac{m}{N}\right) \cdot (x_{sr_i} \geq 0) \\ \quad F_{fric} \leftarrow \begin{cases} \frac{F_e}{N} & \text{if } \omega_i > 0 \\ -\frac{F_e}{N} & \text{otherwise} \end{cases} \\ \quad Fact_i \leftarrow \begin{cases} F_{srv_i} + F_{fric} & \text{if } \omega_i = \omega_{sp} \cdot s \\ \frac{F_{tls}}{N} & \text{otherwise} \end{cases} \\ \quad Fact_i \leftarrow \frac{F_{tls}}{N} \text{ if } Fact_i > \frac{F_{tls}}{N} \\ \quad \alpha_i \leftarrow (Fact_i - F_{srv_i} - F_{fric}) \cdot \frac{D_{sp} \cdot 0.5}{J_t} \cdot (m \cdot kg) \end{array} \\ \begin{pmatrix} x_{sr} \\ \omega \\ \alpha \\ F_{srv} \\ Fact \end{pmatrix}
\end{array}$$

Figure 41: Algorithm for silicone spring design

To determine the force on the sphere due to the silicone properties, silicones with different hardness were used as springs. The silicones in Table 13 are four silicone types that are available in the Rogers Corporation catalogue, [26]. The silicone inputs for the calculation

are given as the % indentation in the first column and then the force required giving this indentation in the second column, repeated for the different silicones. When the same force is applied to a thinner and a thicker silicone strip, the indentation will be more in the thicker strip. Therefore the same thickness will be used in all the tests, in this case silicone rings of 20 mm thickness.

Table 13: Silicone properties from available silicone

HT-1500		HT-1270		HT-1260		HT-1451	
Indent	Δp	Indent	Δp	Indent	Δp	Indent	Δp
%	[kPa]	%	[kPa]	%	[kPa]	%	[kPa]
0	0	0	0	0	0	0	0
2.1	344.7379	3.9	344.7379	5.4	344.7379	7	344.7379
4.9	689.4758	7.5	689.4758	10	689.4758	13.7	689.4758
7.3	1034.214	10.7	1034.214	14.6	1034.214	21.1	1034.214
9.3	1378.952	14.1	1378.952	18.7	1378.952	27.3	1378.952
10.8	1723.690	17.05	1723.690	22.8	1723.690	32.3	1723.690
12.2	2068.427	20.05	2068.427	27	2068.427	37.4	2068.427
13.75	2413.165	22.8	2413.165	31.6	2413.165	42	2413.165
15	2757.903	25	2757.903	35.5	2757.903	46.4	2757.903
16.1	3102.641	27.3	3102.641	38.8	3102.641	50.3	3102.641
17.3	3447.379	29.3	3447.379	41.6	3447.379	54.8	3447.379

When the head collides with the sphere, energy from the moving head is transferred into the spring until the head stops. Then the energy in the spring is released and pushes the head back in a anti-clockwise rotation. When the head is returned anti-clockwise past the point of impact, the sphere between the head and the spring is released.

The results for the silicone rings of 20 mm thickness are given in Table 14. Table 14 gives the silicone type with calculated characteristics of silicone indentation, then the distance of anti-clockwise rotation of the head and then the maximum force applied to the silicone, each at the time the action value occurred. The results, Table 14, show that it is possible to obtain impact duration of longer than the required 0.003 seconds to protect the sphere. All the silicones in Table 13 give the longer impact time, which will protect the sphere against damage at impact.

An anti-clockwise travel of more than zero is required to release the sphere. If the anti-clockwise rotation of the head is still positive, the head does not release tension on the sphere. In such a case the sphere is not released and the head will remain in a “stuck” position. Thus silicones with properties such as HT-1451 will not work. Silicone properties from type HT-1260 result in an anti-clockwise rotation of the head, but it is not enough to release the sphere and the time of pressure release on the sphere might not be enough to release the sphere completely.

Table 14: Results for different silicones

Silicone type	Indent		Anti-clocks travel		Maximum force	
	mm	sec	mm	sec	kN	sec
HT-1500	3.879	0.01532	-1.6946	0.05183	7.8479	0.01537
HT-1270	4.753	0.01912	-0.94345	0.05801	6.0811	0.01915
HT-1260	5.3099	0.0214	-0.4607	0.06178	5.377	0.0214
HT-1451	6.0796	0.02445	0.19669	0.06744	4.8205	0.02448

The author added some arbitrary silicone properties to the list to investigate what would be required to exceed the 18 kN sphere break force and what would be required to stay just below the sphere break force. Those properties are listed in Table 15 and the results are listed in Table 16.

Table 15: Required silicone properties

Required-1		Required-2	
Indent	Δp	Indent	Δp
%	[Pa]	%	[Pa]
0	0	0	0
0.084	344737.9	0.175	344739.9
0.196	689475.8	0.408333	689479.8
0.292	1034214	0.608333	1034220
0.372	1378952	0.775	1378960
0.432	1723690	0.9	1723700
0.488	2068427	1.016667	2068439
0.55	2413165	1.145833	2413179
0.6	2757903	1.25	2757919
0.644	3102641	1.341667	3102659
0.692	3447379	1.441667	3447399

Table 16: Results for different silicones

Silicone type	Indent		Back travel		Maximum force	
	Distance (mm)	Time (s)	Distance (mm)	Time (s)	Force (kN)	Time (s)
Required 1	1.1335	0.00466	-4.0424	0.03635	22.287	0.00468
Required 2	1.4702	0.00602	-3.7548	0.03846	17.501	0.00625

The ideal silicone would be type “Required 2” of Table 15 and Table 16. This silicone is represented in Figure 42 where the indentation is given as a function the pressure applied to the silicone.

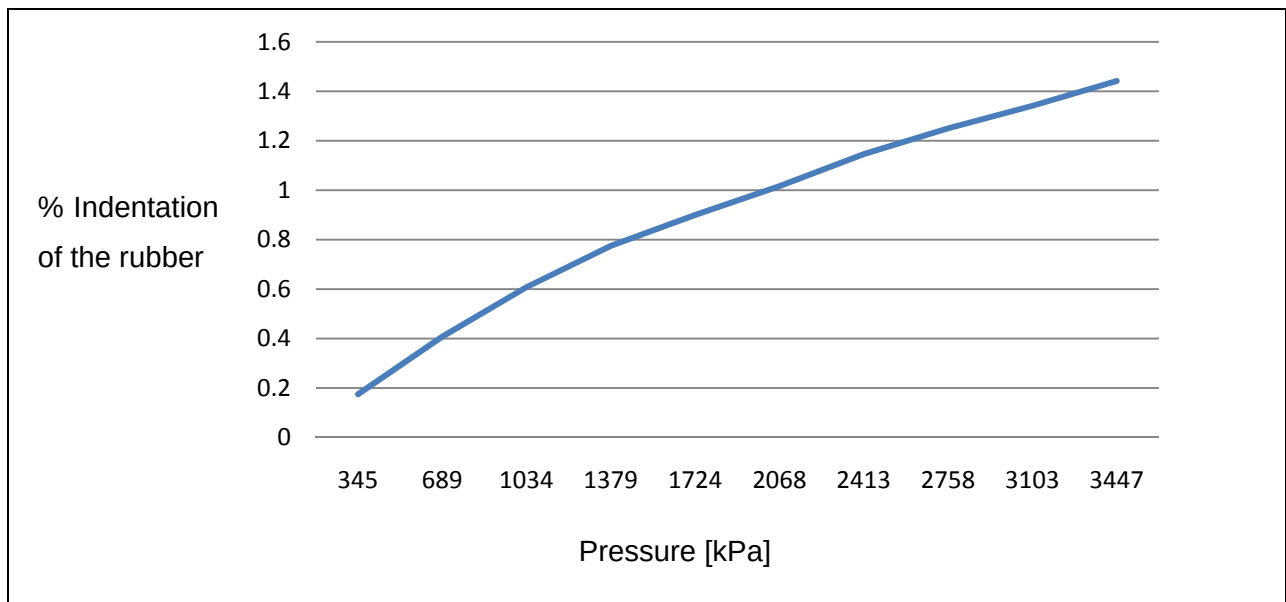


Figure 42: Properties for silicone "Required 2"

Some typical results from the algorithm in Figure 41 are shown in Figure 43 to Figure 45. The results are for the silicone type "Required 1".

Figure 43 indicates the velocity of the head against the time after impact, thus at time = 0 seconds the impact occurs at a speed of 1.047 rad/s. Then the head returns in the opposite direction until a speed of approximately 1 rad/s is reached. Thereafter the head returns to the drive direction, driven by the actuator, to again hit the sphere. After approximately 0.15 seconds the head stops against the sphere.

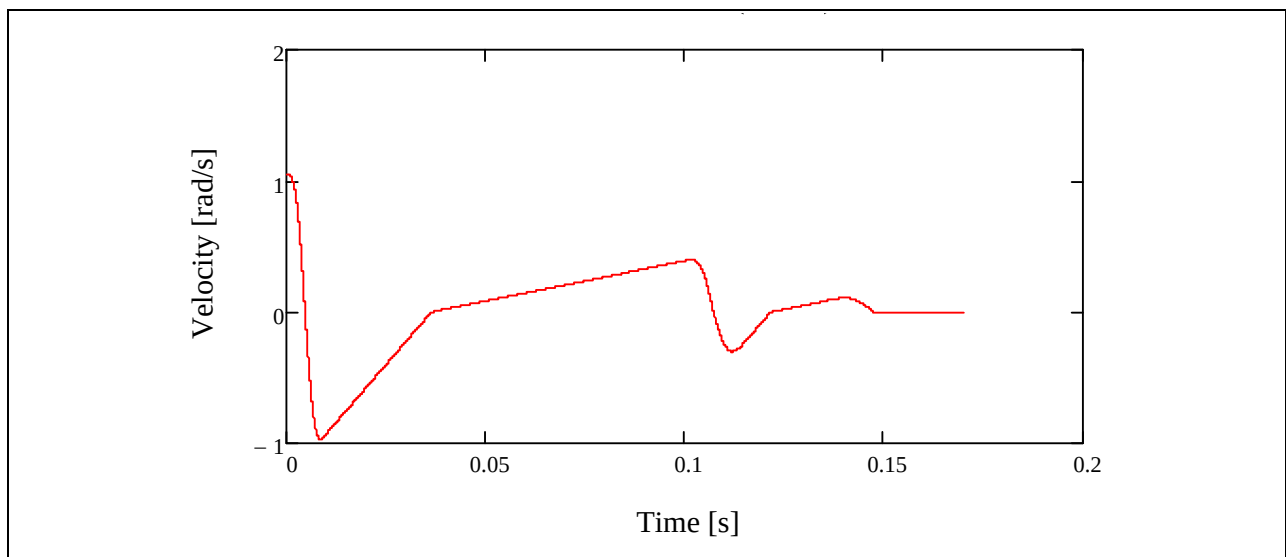


Figure 43: Typical head velocity after impact with the silicone spring of type "Required 1" silicone

Figure 44 indicates the head displacement for the silicone type "Required 1". The indentation into the silicone after impact is approximately 1 mm where after the head turns in

the opposite direction until a distance of 4 mm from the impact position has been reached, representing a back travel of approximately 5 mm. After 0.15 seconds the head stops against the sphere with a slight indentation into the silicone. This correlates with Figure 30 where it is indicated that the head stopped at 0.15 seconds.

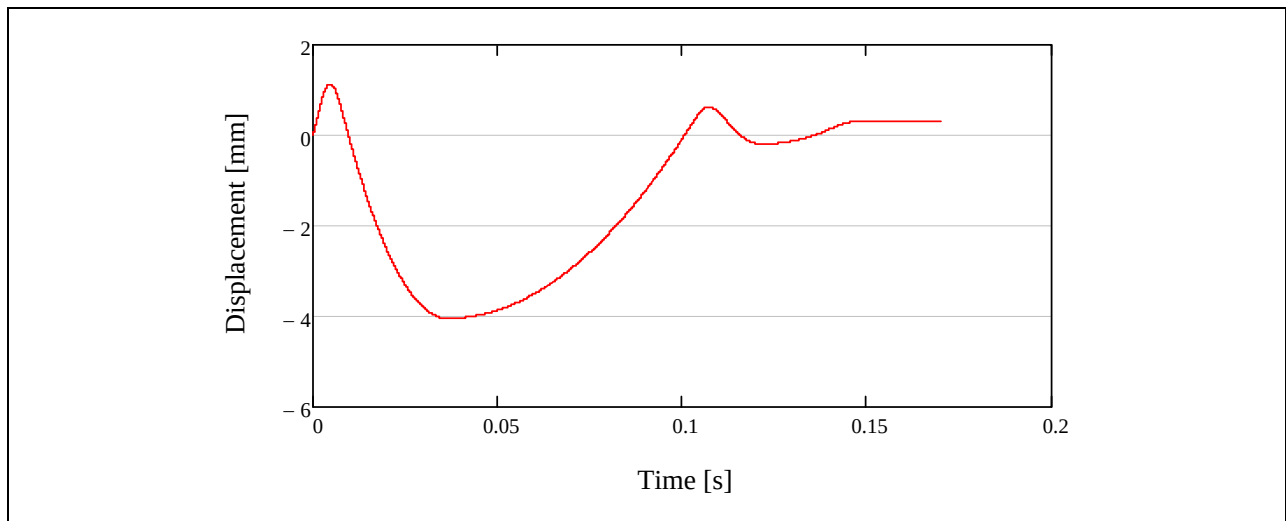


Figure 44: Head displacement after impact with type "Required 1" silicone

Figure 45 is the force diagram of this impact. The initial force on the sphere was over 20 kN after the impact. Take note that the force increases with time after impact, and it is this time interval that must be extended to decrease the impact force. The force on the sphere increases due to the kinetic energy of the head that is transferred to the spring, but also due to the actuator that builds its torque limit up until its continuous stall torque is reached. The increasing torque of the actuator can be seen in Figure 45.

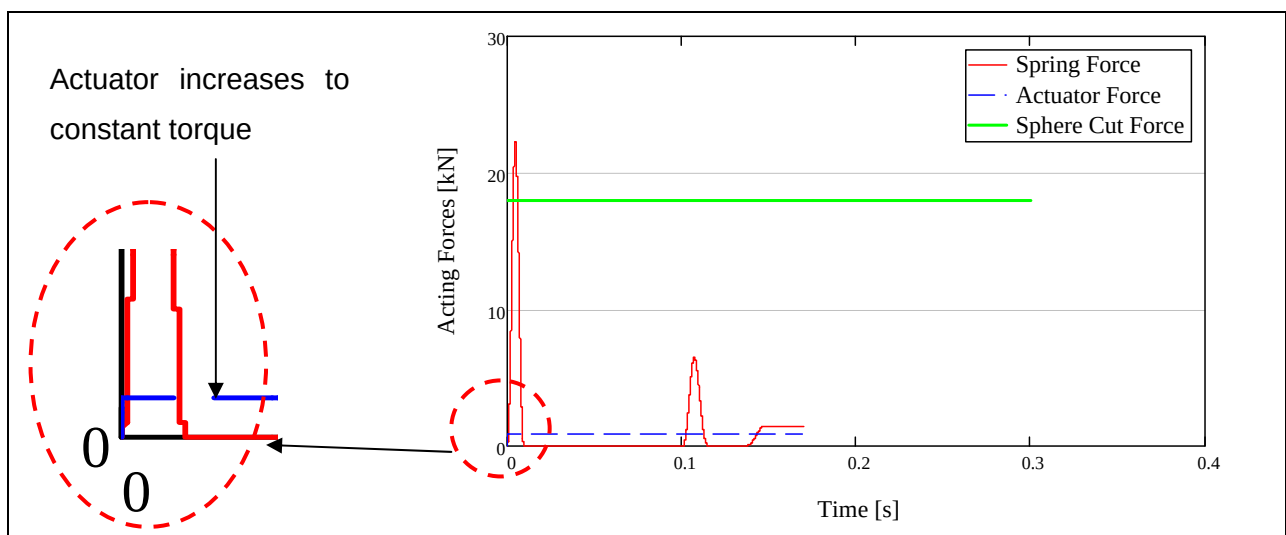


Figure 45: Forces on the sphere after impact

Silicone apparently provides good sphere protection properties and silicones with the required properties are available. The preferred silicones must have the same properties as that provided by HT-1270 and HT-1500 provided by the *Rogers Corporation*.

4.2.3 CONCEPT DESIGN STATUS AND COMPLIANCE VERIFICATION

Spheres are removed from the sphere bed at the required indexing sequence. Both the silicone and metal spring protect the sphere against damage. However, the springs will require inspections and maintenance during the operational lifetime of the tank unloading device. Table 17 lists the criteria and the evaluation of the existing concept status against the criteria.

Table 17: Evaluating the concept status

Criteria	Advantage	Disadvantage
Sphere removal from sphere bed	333 spheres / hr	
Unload rhythm	Very good, every 7 seconds	
Sphere damage		Spheres could be damaged by scissor action between head and housing
Broken sphere separation		Only dust, not larger pieces
Maintenance requirement		Regular inspections will be required and most likely maintenance to replace the springs.
Maintenance possible		No, head diameter too big for removal
Dust generation	Dust generation is limited due to slow rotational speed	
Nuclear safety		When a sphere is broken a DiD barrier is broken

4.2.4 FURTHER GRAVITY CONCEPT DEVELOPMENT

Due to the inspection and maintenance requirements the head must be removable. The head is however not removable through the maintenance pipe. The maintenance pipe has an inner diameter of 400 mm as indicated in Figure 28. Therefore the head diameter must be reduced to less than 400 mm. This concept will be developed further to limit the maintenance needs, rather than to increase maintainability. The spring assembly is the part of the tank unloading device that will need inspection and maintenance. If the need for the spring can be removed, the need for spring maintenance will also be removed.

It has however still been decided to decrease the head diameter to make the head removable through the maintenance pipe. Due to the smaller head diameter bridge-forming can possibly occur. Bridge-forming can be limited or even prevented by the maintenance pipe that can be positioned in such a way that it prevents large sphere loads in the head, as indicated in Figure 46, but this must be verified through testing. This is the same principle that is used in bird

feeders where a container is filled with seeds and the seeds do not empty through the opening at the foot of the feeder, shown in Figure 47.

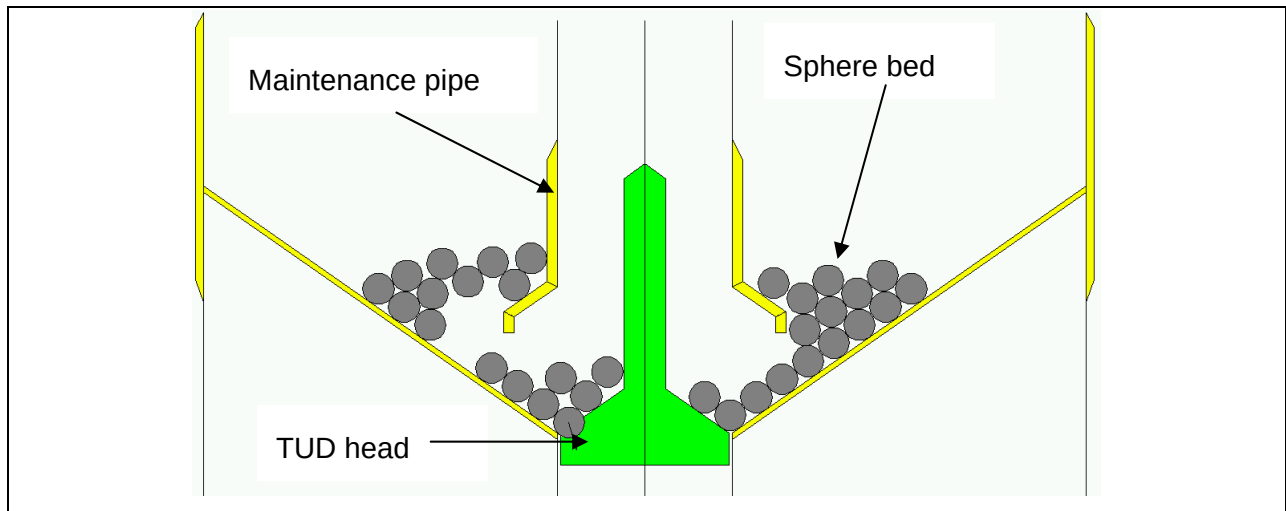


Figure 46: Maintenance pipe used to limit amount of spheres on the head



Figure 47: Bird feeder

The need for the spring can also be removed by removing the scissor action between the head and the housing which causes the potential sphere damage. The scissor action between the head and the housing can only be removed when the sphere passes through the centre of rotation. To remove spheres through the centre of rotation and still mechanically index the spheres, the castle separation, indicated in Figure 48, was developed. However, this concept could potentially cause a blockage at the entrance to the sphere pipe. This blockage is due to the funnel shape of the sphere catcher at the entrance of the sphere pipe. Two spheres cannot enter the transport pipe together and then might cause a blockage at the entrance due to bridge-forming.

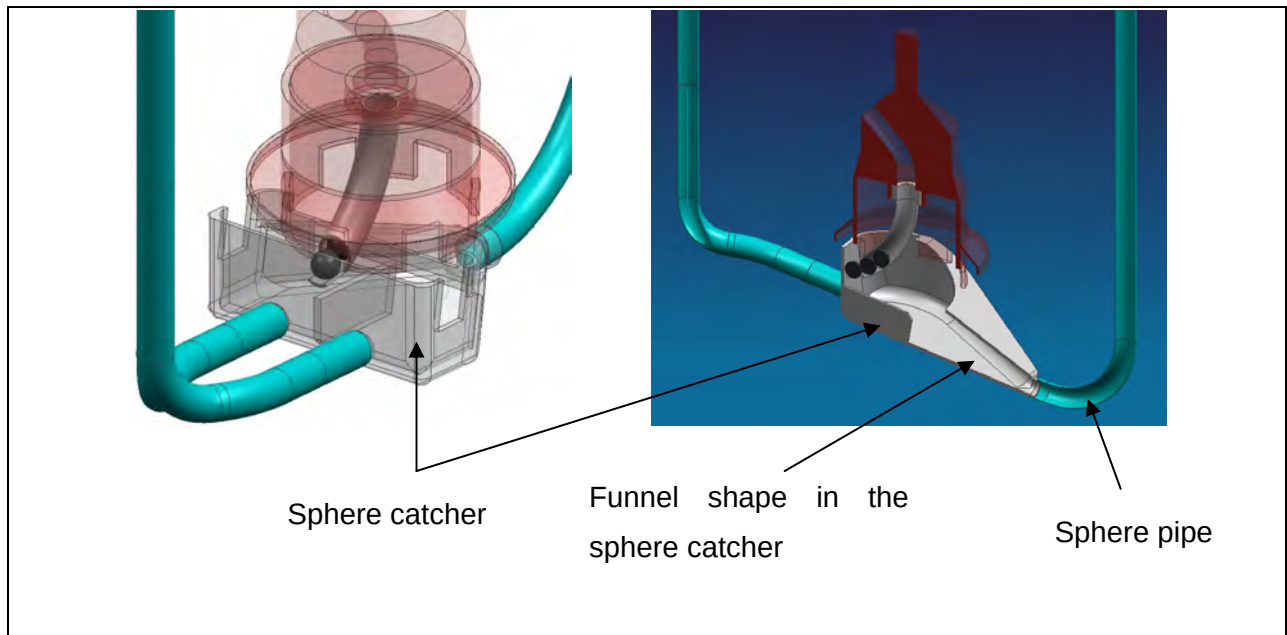


Figure 48: Castle indexing

To remove the scissor action, the head was divided into two levels. Previously the head had to remove a sphere from the sphere bed and index the sphere through the scissor action as indicated in Figure 22. Due to the separation of these two functions they will now be performed in two separate actions on two levels. Thus the spheres are not indexed into the sphere pipe directly from the top of the head, but from the second layer of the head.

This improvement area is indicated in Figure 49.

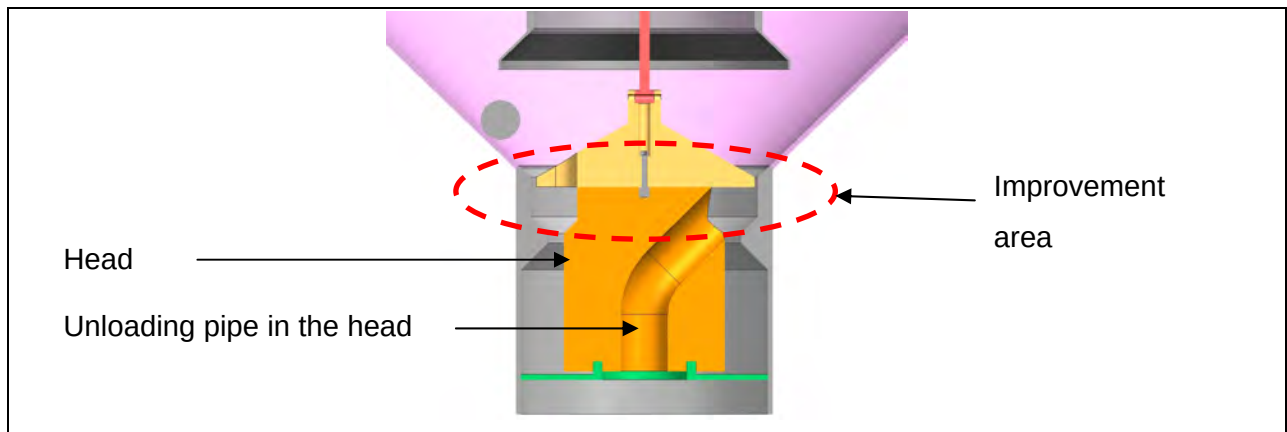


Figure 49: Separate functions between the separated levels

With the improvements of the gravity concept, the concept can be visualised as shown in Figure 50. The unloading functional levels can be defined as follows:

- Level 1: Remove spheres from the sphere bed;
- Level 2: Line spheres up for single unloading;
- Level 3: Single unloading.

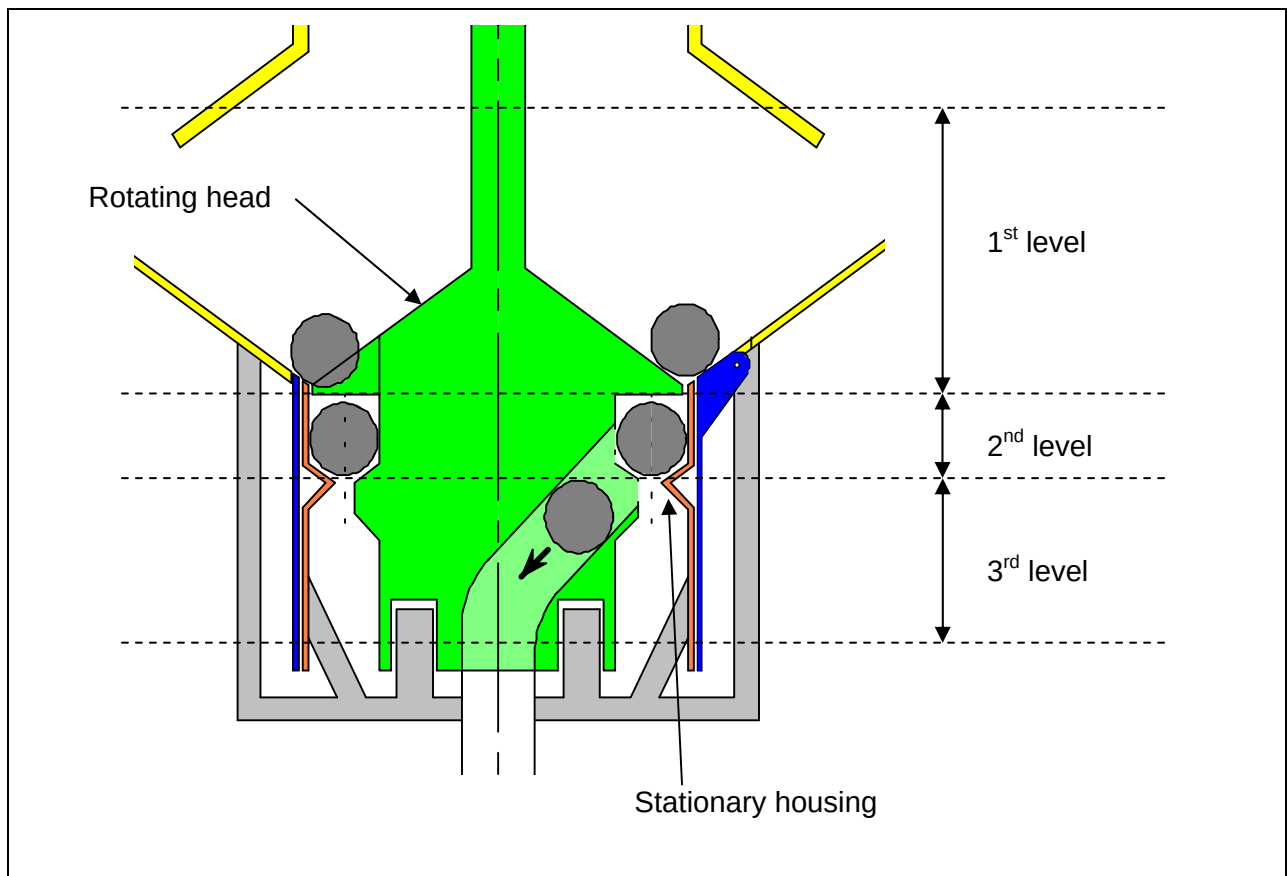


Figure 50: Current gravity concept improvement

The functions of the levels are described as follows:

- a. Level 1: Remove spheres from sphere bed

The function of the first level is to remove the spheres from the sphere bed. This is done by the opening between the maintenance pipe and the cone which limits the amount of spheres that can flow between the maintenance pipe and the tank cone. By increasing the opening between the two components more spheres will be allowed to flow through, while by decreasing the opening fewer spheres will flow through. This width of the opening is not known and must therefore be determined by a full-scale test. The opening is indicated in Figure 51.

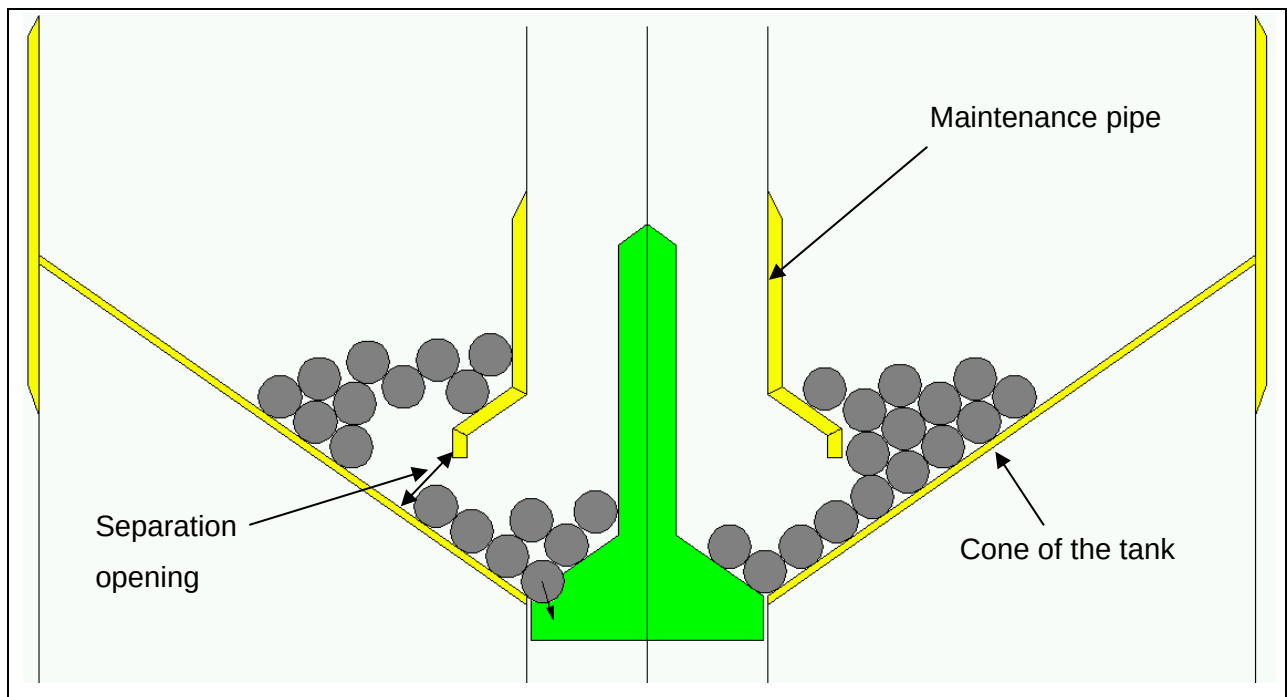


Figure 51: Level 1 operation: remove spheres from sphere bed

b. Level 2: Line spheres up for single unloading

On the second level (indicated in Figure 52) the spheres are aligned for entrance to the third level. The spheres settle at the bottom of the angle between the cone and the head on the second level, while the head rotates. During rotation of the head the exit of the second level reaches the next sphere to fall through onto the third level.

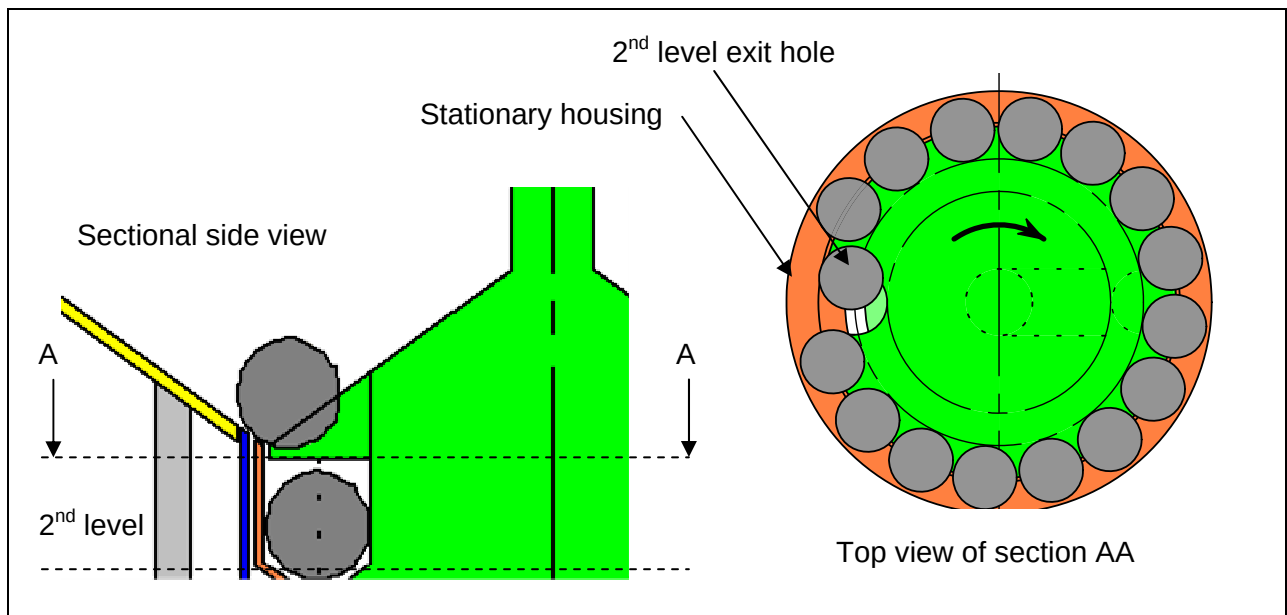


Figure 52: Level 2 operation: align spheres for single unloading

c. Level 3: Single unloading

The third level is the last level of sphere unloading. The spheres on the second level are moving in the same direction as the rotation of the head, but at half the speed, as shown in Figure 53. This is because the spheres are resting both on the rotating head and on the stationary housing.

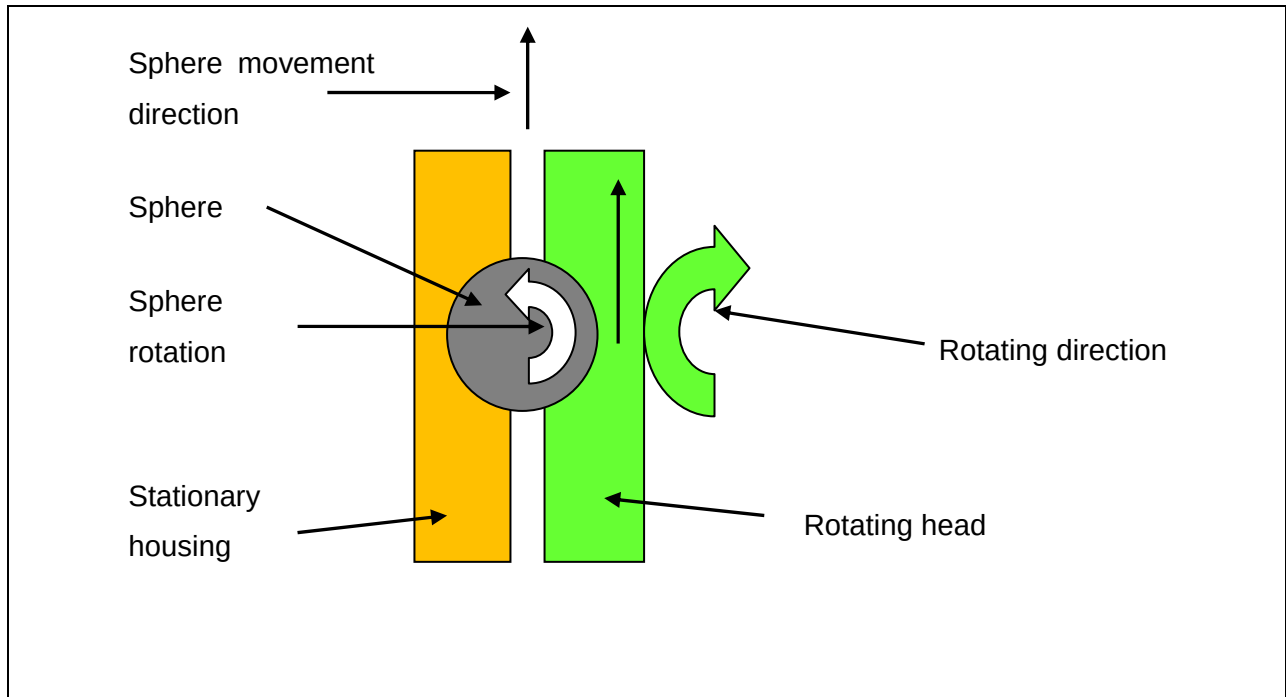


Figure 53: Sphere speed relative to head and housing

As the access hole of the head rotates, it captures the last sphere in the row on the second level, as shown in Figure 54.

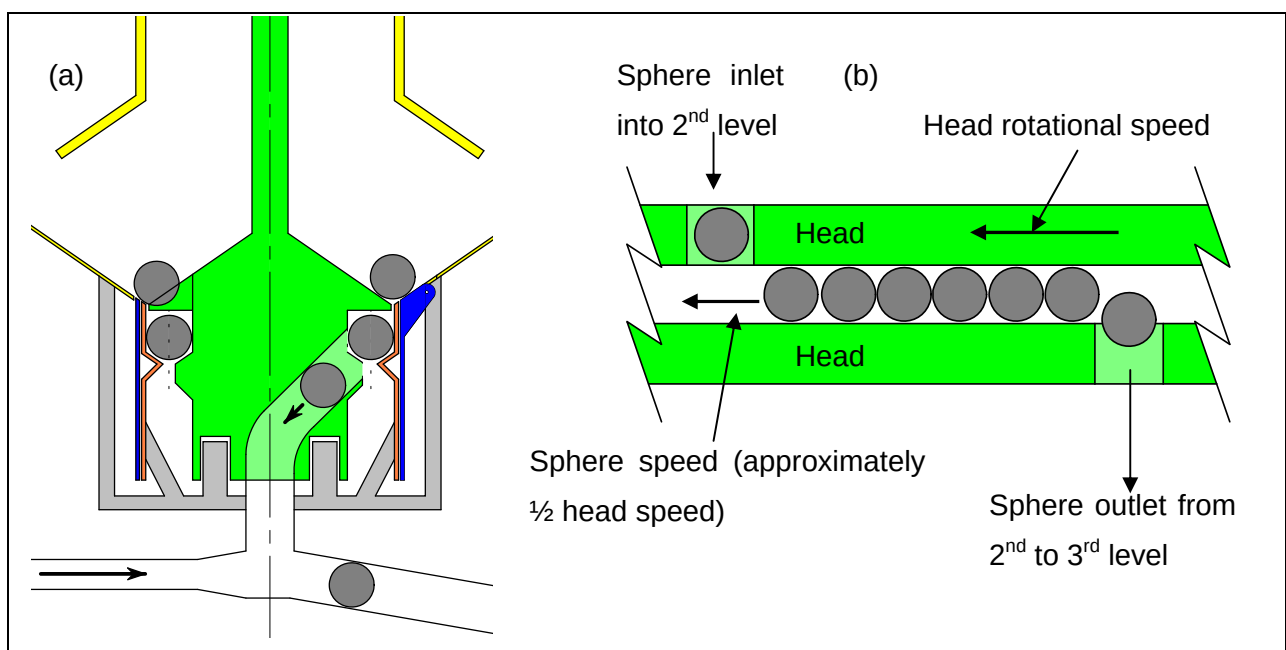


Figure 54: Third level operation: single unloading or indexing

The intake of spheres into the exit hole, or in other words the rate of unloading, is a function of the rotational speed of the head. The faster the head rotates, the faster the unloading of spheres takes place.

4.2.5 CONCEPT IMPROVEMENT FOR BROKEN SPHERE PIECES

Before a concept for the removal of pieces of broken spheres can be developed it is important to understand what the unusable spheres are. There is no data available on sizes of broken fuel spheres. Therefore this paragraph discusses the sizes of broken spheres that can be expected.

The fuel sphere specification requires the following from the sphere integrity, [34]:

- At least 99.5 % of the fuel spheres shall, at a confidence level of 95 %, survive 50 drops from a height of 4 m onto the graphite sphere bed;
- At least 90 % of the fuel-spheres shall, at a confidence level of 95 %, have a crushing strength of at least 18 kN when compressed between parallel flat steel plates.

To be able to design for pieces of broken spheres, there must be an understanding of the geometry of the pieces of broken spheres and the effect of these pieces on the sphere transport. Based on the picture of a sphere given in Figure 3, it is assumed that the sphere can delaminate, or break into two parts (or more) or even be pulverized into chunks. (Naturally, the following description is based on assumptions of how the fuel sphere geometry can fail, but it is at least a starting point for defining the size for the separation gap.)

When a sphere delaminates, the outer 5 mm graphite skin delaminates from the 50 mm sphere core, just like an orange skin when the orange is peeled. The geometry of the remaining sphere will typically be as shown in Figure 55. Such a delaminated graphite skin is assumed to be not thicker than one-third of the sphere's shell thickness. (This assumption is based on the friction between the sphere core and the skin, which will render it difficult to remove a half section of the skin.)

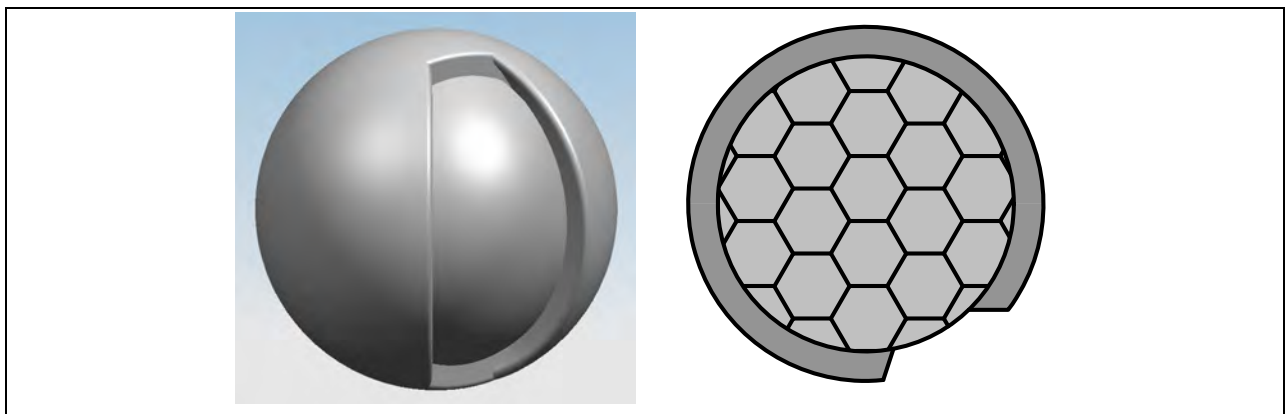


Figure 55: Damaged sphere

Sphere fracture must occur within the 50 mm diameter of the sphere core; otherwise it will just represent delamination of the graphite skin, Figure 56 (b). According to this reasoning it can be assumed that a broken sphere will be less than 40 mm thick with a flat side, as in Figure 56 (c).

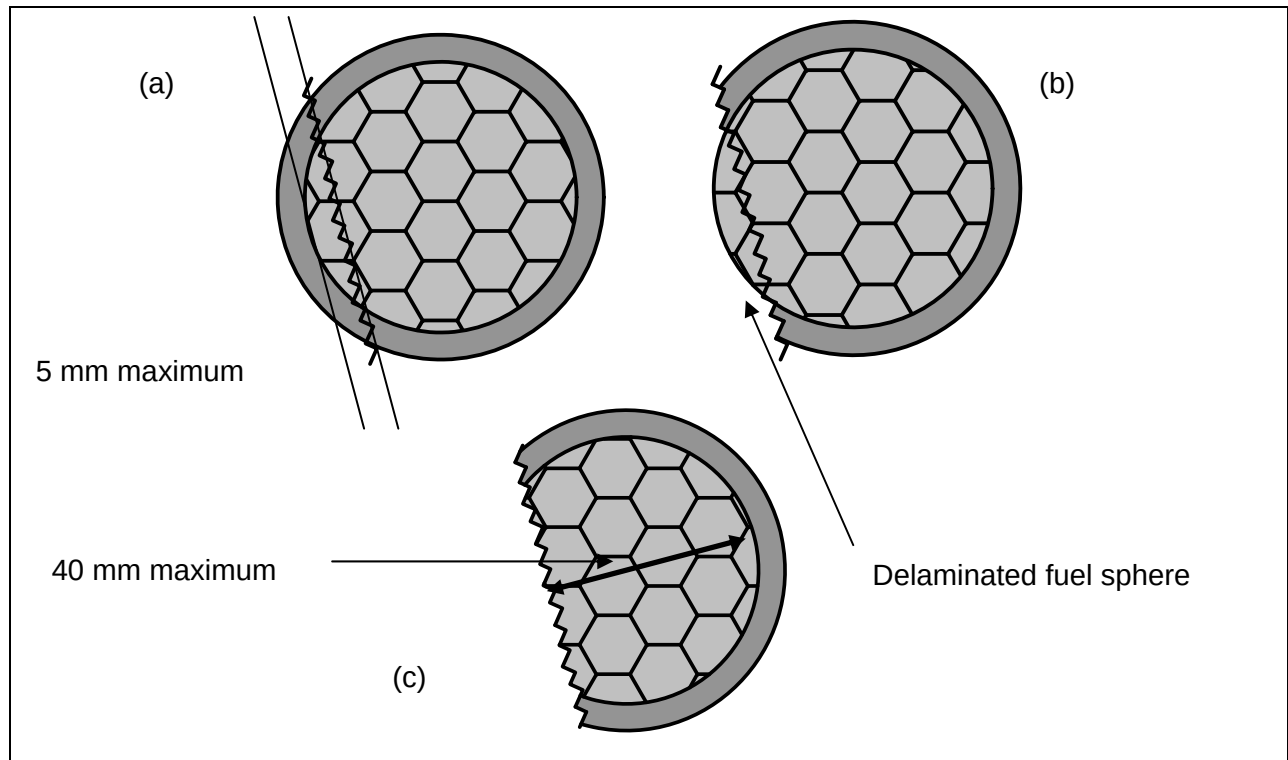


Figure 56: Possible fuel sphere fracture

The geometry of the pieces must be able to block and wedge a sphere in a sphere pipe. Figure 58 indicates that the sphere pipe has a diameter of 7 mm larger than the sphere, where the sphere is supported on the pipe ribs creating a gap of between 5 and 6 mm. A piece of broken sphere smaller than 5 mm would thus not be able to wedge the sphere, but a larger piece can cause a wedged sphere, as shown in Figure 57.

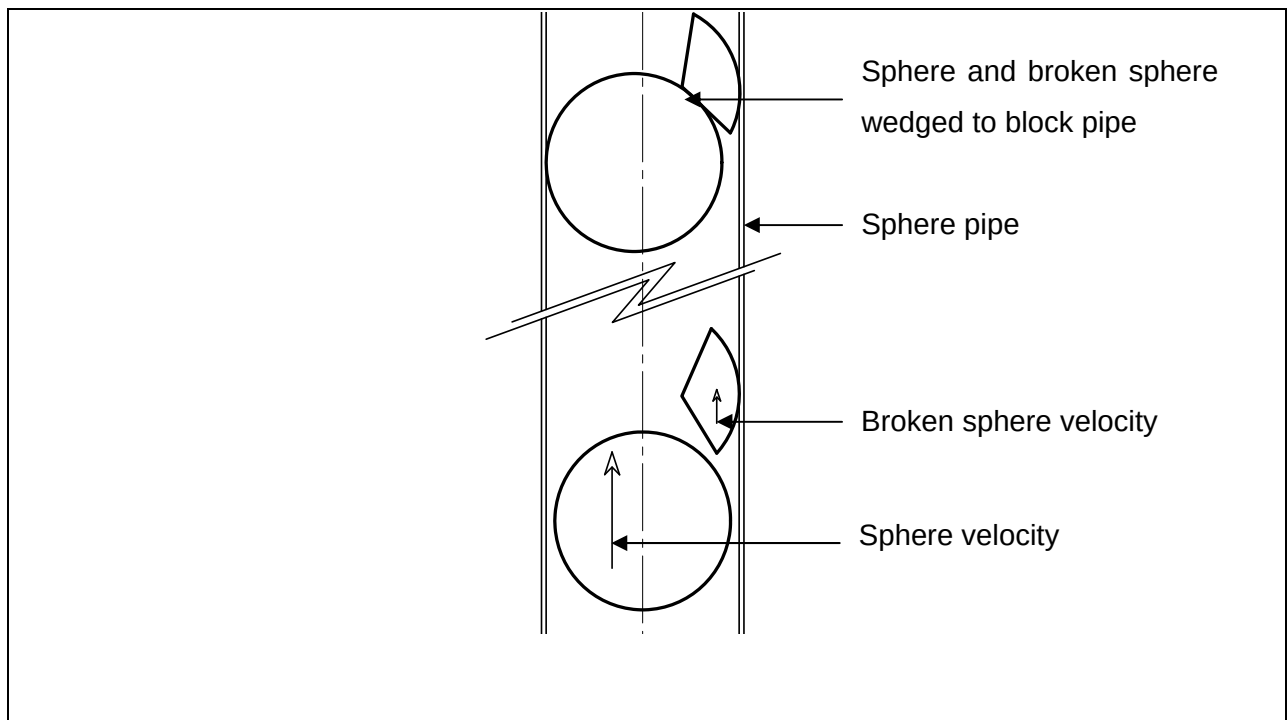


Figure 57: Sphere blockage in sphere pipe

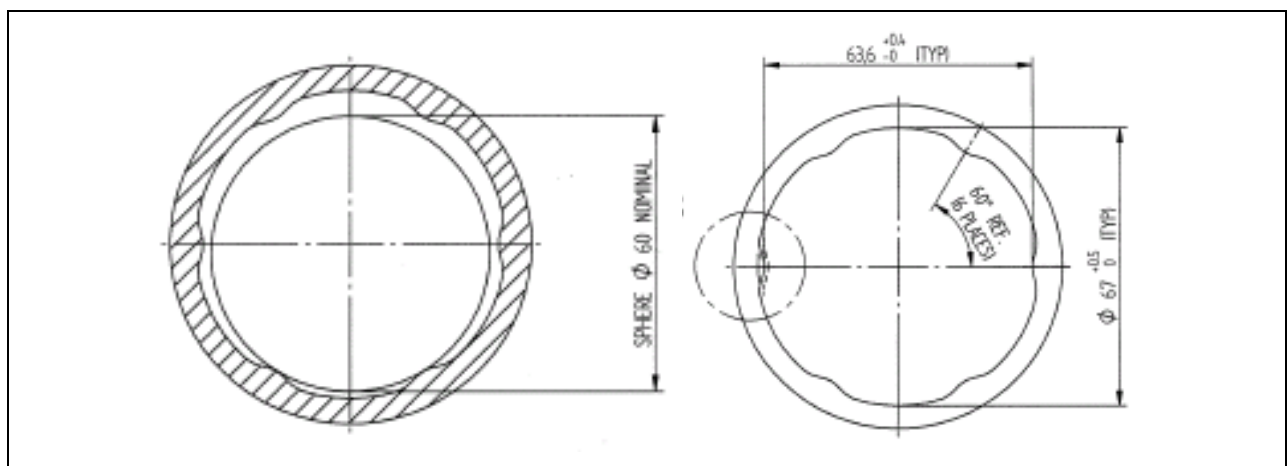


Figure 58: Sphere pipe dimensions

Pieces of broken spheres will be transported with the spheres when they are large enough. Based on Figure 56 it is assumed that a sphere of 50 mm, (Figure 56 (b)), can still roll in a sphere-pipe, but a sphere of 40 mm, on a flat side, (Figure 56(c)), will not be able to roll in a sphere pipe without transportation gas.

Figure 59 (a) and (b) indicate that a broken sphere piece must have an edge of less than 6 mm to be able to wedge between the sphere and the sphere pipe. Figure 59 (c) shows a piece of a sphere that is too large to wedge the sphere, and in this case the piece of sphere will be transported with the sphere.

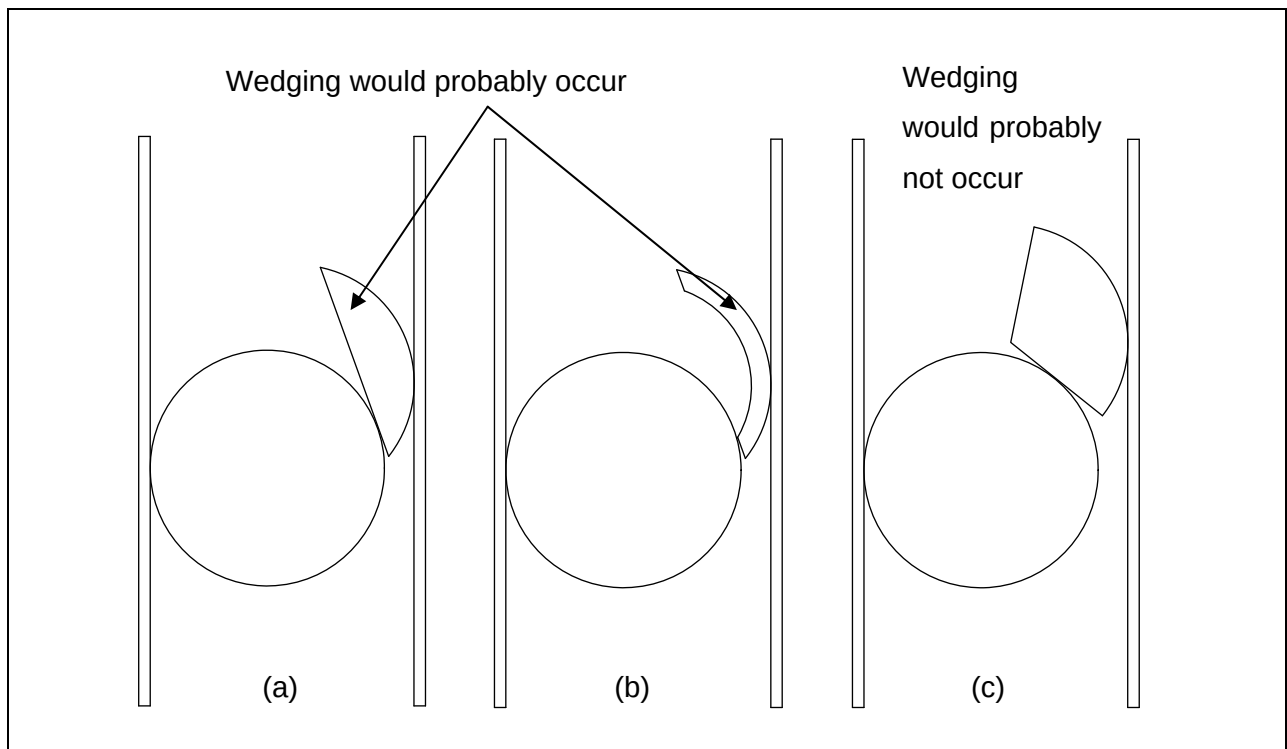


Figure 59: Broken sphere geometry that could cause blockage

Based on the discussion about sphere breakage it can be concluded that the cut off size can be assumed to be between the 40 mm and 50 mm. (Anything less than 40 mm is a broken sphere and anything more than 50 mm is a usable sphere). According to the German experts, unusable spheres are regarded as smaller than 50 % of the volume of the sphere.

The requirement is to separate all the pieces of broken spheres from usable spheres and store the broken spheres in the bottom of the tank. The best location to separate the broken spheres is therefore at the last stage of unloading where the pieces can fall through to the bottom of the tank as shown in Figure 60.

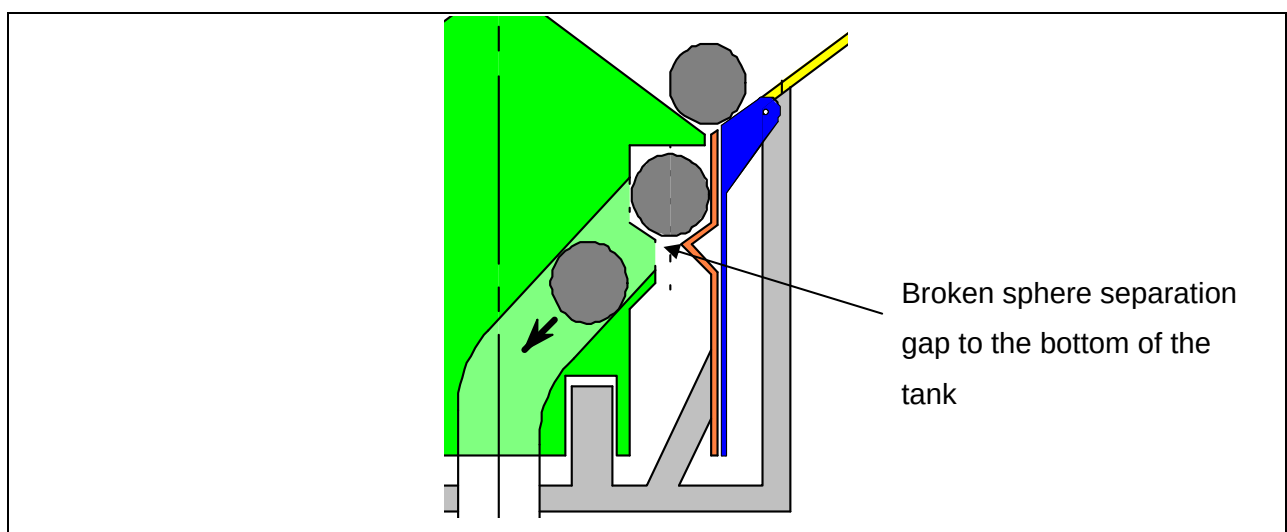


Figure 60: Removal of pieces of broken spheres

Movement restriction must be provided to support the head against horizontal movement when larger pieces from damaged spheres are wedged through the separating opening. The movement restriction must also comply with maintenance requirements in the nuclear industry. Therefore it has been decided to use Vespel as the material for a bearing. Figure 61 shows how the bearing will be installed.

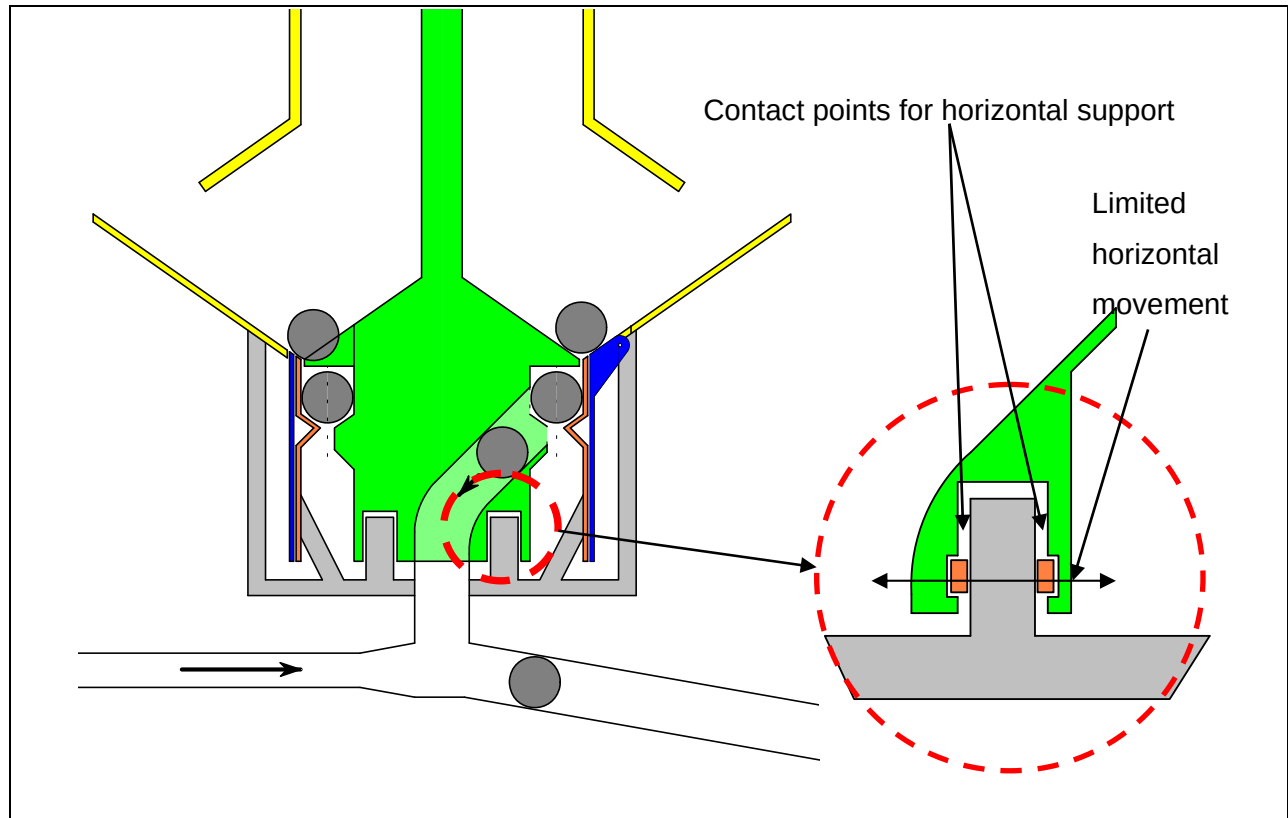


Figure 61: Bearing support

The gravity concept can be re-evaluated to verify the design status against the requirements. Table 18 indicates that the requirements have been achieved theoretically, but are to be verified through tests.

Table 18: Re-evaluating the concept status

Criteria	Advantage	Disadvantage
Sphere removal from sphere bed	333 spheres / hr	To be tested
Unload rhythm	Theoretically every 7 seconds	To be tested
Sphere damage	No sphere damage foreseen	
Broken sphere separation	Broke sphere separation is theoretically possible	To be tested
Maintenance need	Measure the torque on actuator. If the torque increases there could be possible bearing damage.	
Maintenance possible	Yes, head can be removed	
Dust generation	Dust generation is limited due to slow rotation speed	
Nuclear safety	No sphere damage foreseen	

4.3 CONCLUSION

It can be concluded that the gravity concept complies theoretically with the requirements that are placed on the tank unloading device. The bending stress of the metal spring assembly is very high and will fail due to metal fatigue. In the case of the silicone spring, the life expectancy of the silicone is not long enough to cover the required life expectancy of the unloading machine. Thus, although calculations indicate that both the springs could work, a maintenance plan is to be provided to prevent spring failure.

The latest improvements present a better solution for the gravity concept, not needing an intensive maintenance plan due to the absence of the springs.

5. TESTING

5.1 INTRODUCTION

An examination of Table 18 confirms that the requirements have theoretically been complied with. This section describes the testing of the gravity unloading machine developed for the tank unloading device.

5.2 TEST METHODOLOGY

A full scale test unit was built to confirm the geometrical assumptions and to verify the TUD's capability to satisfactorily perform the functions that were allocated to the TUD. The functions are as follows:

- Make provision for maintenance;
- Lift single sphere;
- Count spheres that pass through the unit;
- Separate pieces of broken spheres;
- Empty tank for final decommissioning.

Geometry that has to be confirmed:

- TUD maintenance pipe height;
- Angle of cone ring.

Figure 62 shows the three-dimensional model of the tank unloading device test unit, which has been designed as a full-scale model. The tank unloading device is connected to a full-scale model of the bottom section of the storage tank in order to simulate the sphere flow through the tank into the tank unloading device. The maintenance pipe is adjustable because the optimum height for the opening for spheres to flow through onto the tank unloading device head (Figure 51) is to be determined through testing. For this test unit the spheres are returned to the same tank to be able to run a continuous test.

The tests can be divided into two sets. The first set of tests is required to complete the design (i.e. parameters that must be verified with tested settings). The second set of tests is to verify the performance of the tank unloading device.

The first test is to verify the capability of the tank unloading device to unload a sphere and then a string of spheres. This test will confirm the functional operation of unloading.

The second test is to determine the exact position (height) of the maintenance pipe. When the test unit adjustment is complete, the test unit is to be filled with more spheres which will enable to run the test continuously.

The rotating speed must be increased and decreased to determine the minimum, maximum and optimum unloading speeds.

Broken spheres will be added to the sphere bed to test the capability of the tank unloading device to separate and remove broken spheres from the usable spheres. The final test is to empty the test unit of all usable and broken spheres.

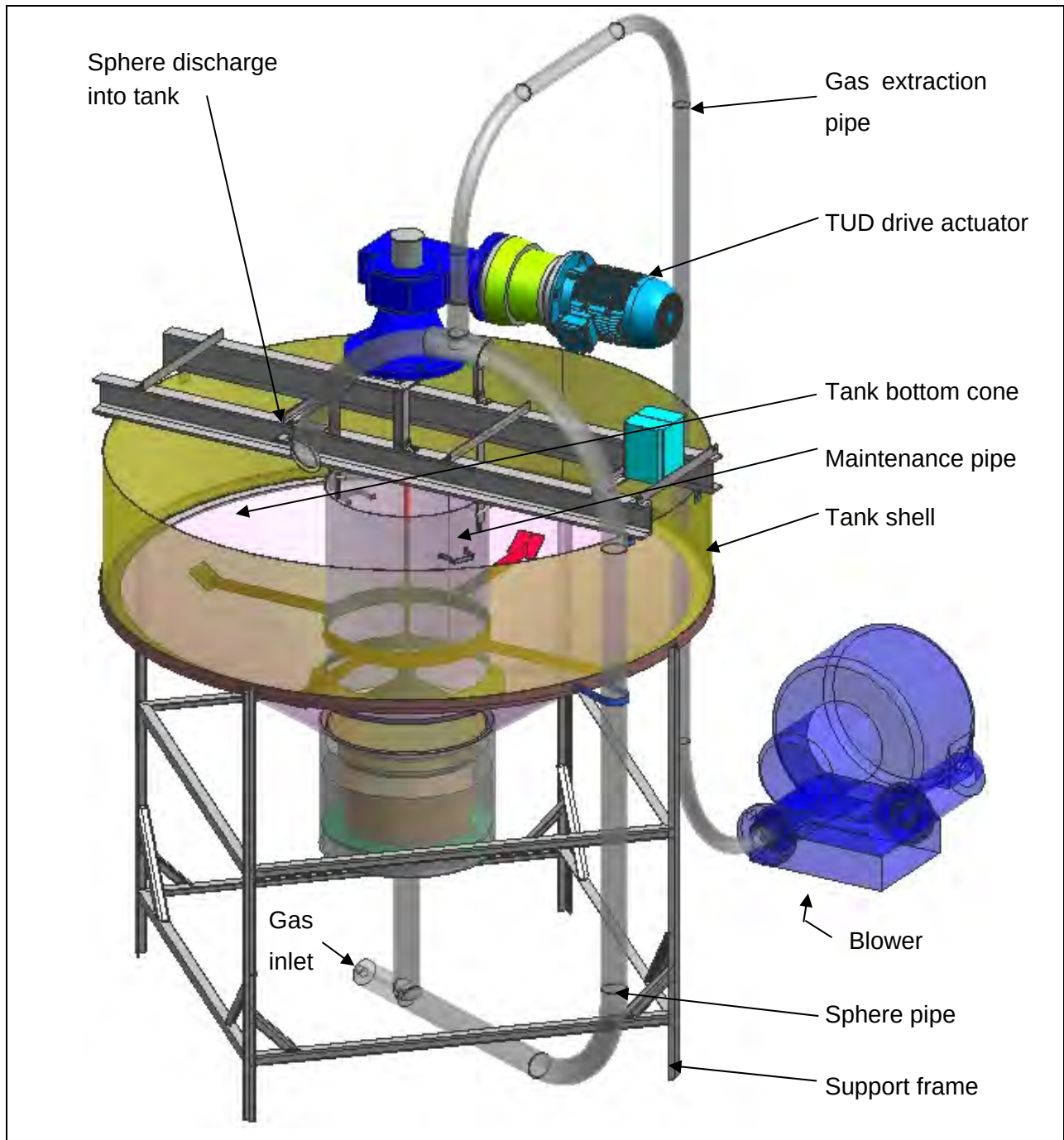


Figure 62: Full scale tank unloading device test unit

The test unit, shown in Figure 63, has been built according to the three dimensional model indicated in Figure 62.

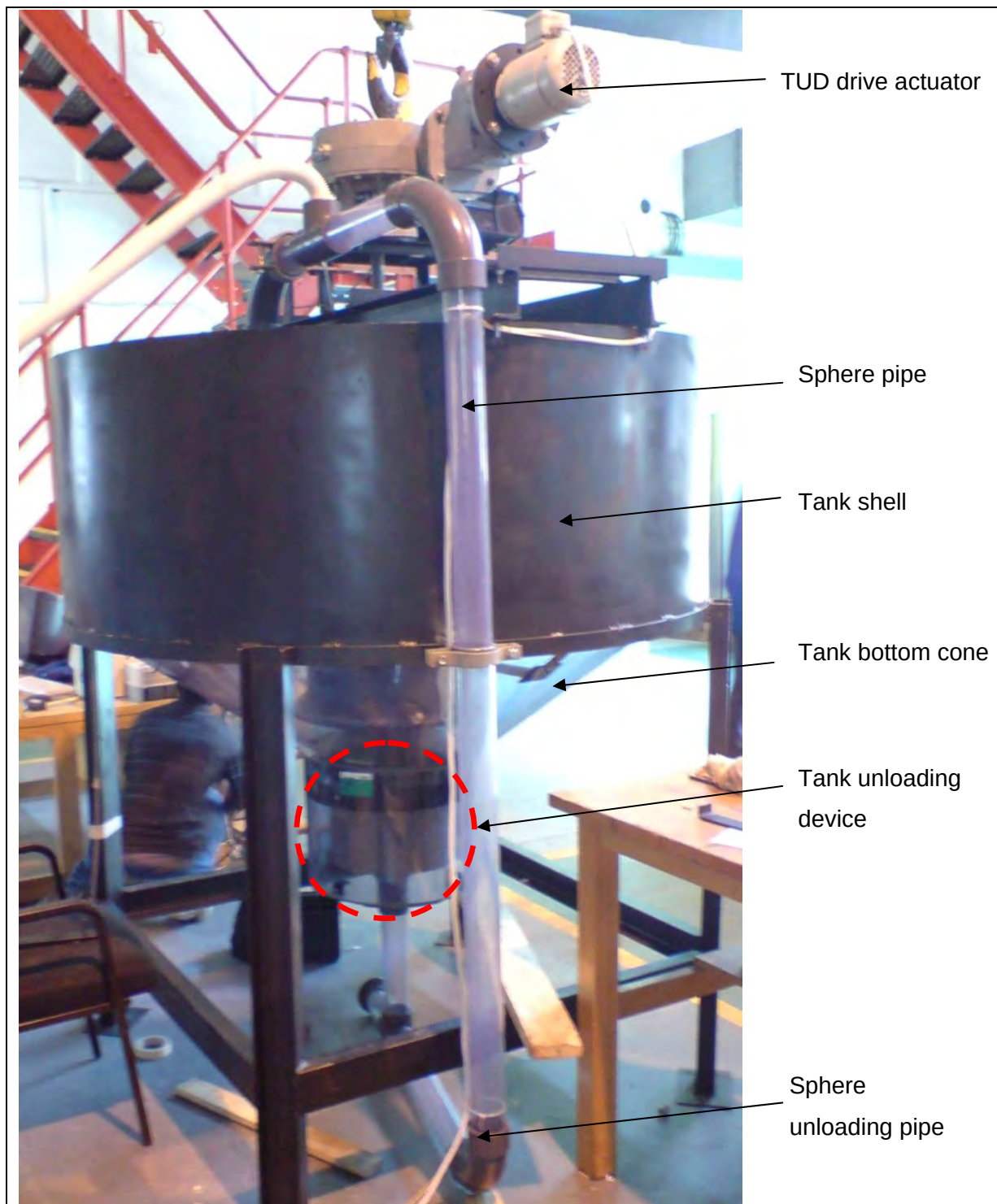


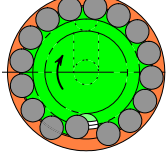
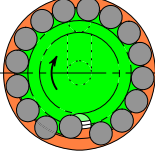
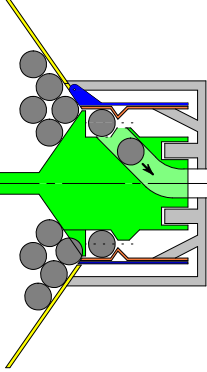
Figure 63: Constructed test unit

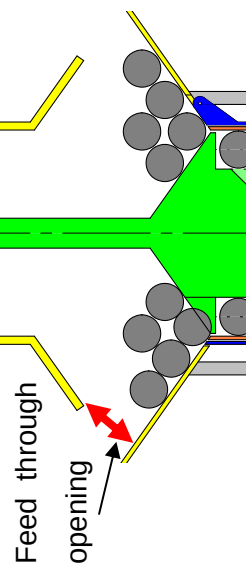
5.3 TESTING

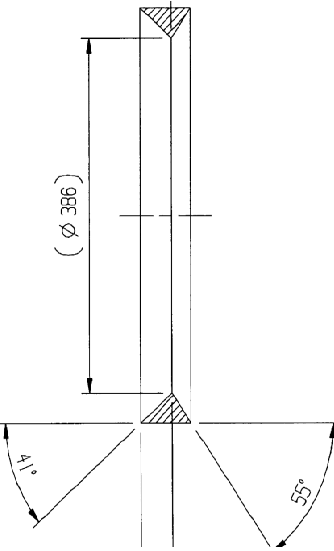
5.3.1 TEST SHEET AND RESULTS

Table 19 is the test sheet which was completed during the test. This sheet lists the tests that were performed, in sequence. The order of the test sheet has been developed to ensure that the functionality tests and geometrical adjustments are done in the correct order.

Table 19: Tank unloading device result sheet

Test	Test	Method	Result	Indication
1A	Circulate enough spheres through the TUD until sphere flow reliability is verified	Servitude pipe lifted (200 mm) Fill unit with 15 spheres Rotate head slowly and observe sphere flow through the head	Continuous sphere flow achieved, however test unit horizontal alignment sensitive	
1B	Determine optimal rotational speed for the TUD head	Keep adding spheres manually Adjust the rotational speed until optimum speed is achieved	Reliable for the full speed range available, up to 10 min ⁻¹	
1C	Determine optimum amount of spheres on the TUD head to ensure continuous sphere flow	Rotate head at optimum speed Add more spheres onto the head and determine the maximum number of sphere rows on the head that still allow sequential unloading	4 rows	

Test	Test	Method	Result	Indication
2	Determine optimum feed through opening	Adjust servitude pipe to allow a 90 mm gap between pipe and cone Rotate head and monitor for continuous sphere flow Slowly add more spheres to the unit Adjust the servitude pipe to ensure that the level of spheres on the head is limited to optimum number of spheres Adjust pipe lower if too many spheres are on the head	85 mm	
3A	Determine the lower speed limit	With the test unit running smoothly, lower the rotational speed of the head until the unloading becomes unreliable and note the speed limit Repeat again to ensure that the lower speed limit is repeatable	No minimum rotational speed	
3B	Determine the higher speed limit	With the test unit running smoothly, increase the rotational speed of the head until the unloading becomes unreliable and note the speed limit Repeat again to ensure that the higher speed limit is repeatable	Max speed available was 1.4 min^{-1}	
3C	Theoretical calculations indicated that a rotational speed of approximately 0.39 min^{-1} is required to unload 333 spheres/hr. Verify the rotational speed required to unload 333 spheres/hr	With the test unit running smoothly, adjust the rotating speed of the unloading head until the required 333 spheres/hr is reached Repeat the test to ensure repeatability of unloading at this speed At this stage the test set-up should be adjusted for operation without pieces of broken spheres	Approximately 0.3 min^{-1}	

Test	Test	Method	Result	Indication
Pieces of broken spheres added to the contents of the test unit				
4	Determine how the test unit will operate with pieces of broken spheres between undamaged spheres	Add unusable spheres and pieces of broken spheres to the sphere bed Catch the pieces below test unit in a bucket Run test unit again Note blockages (if occurring) and describe the blockage Make recommendations to prevent blockage	Number of blockages: 1 The blockage was test unit related and not design issue	
2	Verify that the locked position of the maintenance pipe is still the optimum position	Check that the maintenance pipe position is still maintaining the optimum amount of spheres on the TUD head	Successful	
5	Verify the test unit can be emptied of spheres and pieces of broken spheres	Remove the sphere return pipe from the test unit and divert the sphere return pipe to appropriate storage bin Run the test until the test unit is empty	Successful	
2	Verify that the locked position of the maintenance pipe is still the optimum position	Check that the maintenance pipe position is still maintaining the optimum amount of spheres on the TUD head. (This should still be functional with or without pieces of broken spheres)	Successful	
Sensitivity Tests				
6	It is assumed that the 41° side of the ring will result in clamping of the spheres, while the 55° side will carry the spheres	With the "Ring, Cone Sleeve" installed with the 41° facing up, turn it now over with the 55° facing up Continue the test with the same test set-up parameters as before	Sphere clamping did occur with the 41°	 The diagram shows a cross-section of a cone sleeve. It has a central vertical axis. Two angles are indicated: 41° on the left and 55° on the right. A horizontal line passes through the center. A dimension line indicates a diameter of (Ø 336). The 41° angle is shaded, indicating where sphere clamping occurred.

5.3.2 TEST FINDINGS

The purpose of Test 1a was to determine whether the sphere flow through the TUD head was reliable. The TUD drive unit was used to rotate the TUD head. It is important to verify how the spheres enter and exit the second level.

Spheres entered the second level of the head reliably. However, there was a tendency for the spheres to fall forward to the other side, in front of the unloading line of the second level. This caused a blockage, preventing the next sphere to enter, as indicated in Figure 64.

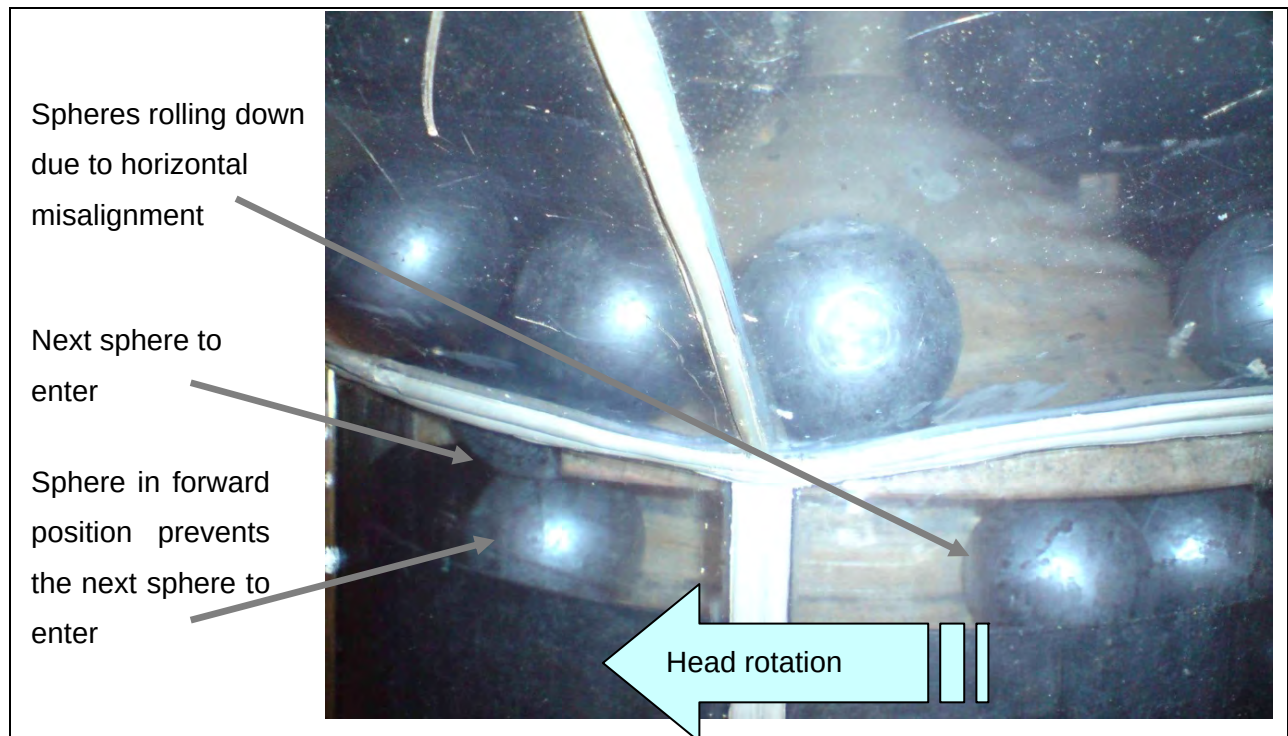


Figure 64: Spheres enter the second level of the head

At first spheres exited the second level irregularly. It was found that the ring cone was not installed horizontally, causing the spheres to roll to the lowest point, as indicated in Figure 64. When the exiting hole reached that lowest horizontal point, all the spheres just drained out through the exit hole into the unloading pipe. The unloading sequence improved when the ring cone was levelled horizontally. Finding: The TUD is very sensitive for horizontal misalignment. This means that an improvement is required to prevent the sensitivity for horizontal misalignment. (After the horizontal alignment the discharge was still not in a steady rhythm because the test set-up did not cater for horizontal alignment settings and full alignment could not be achieved. However it was sufficient for concept testing.)

It was observed that all the areas of the TUD in contact with the graphite sphere were quickly coated with graphite dust. The friction on all four contact points of the sphere became the same – graphite-to-graphite due to the contact points between sphere-to-sphere (2x), sphere-to-cone

ring and sphere-to-head. Thus there was no positive driving force from the rotating head to the sphere to force it around the TUD head towards the exit. This caused the spheres to intermittently exit the second level of the TUD from the other side of the head. This was caused the spheres rubbing against each other and causing the spheres to move in the opposite direction, thus spheres flowed in both directions around the head. The result was an irregular unloading sequence, where sometimes nothing exited and then suddenly two spheres exited the TUD (one from each side). To prevent irregular unloading from occurring, the one side of the TUD head was blocked to force the spheres to the other side of the head only.

With the one side of the TUD head blocked, the loading and unloading of the second level improved. As the spheres on the second level progressed towards the exit of the second level, a next sphere entered into the opened space in the front of the row of spheres. The spheres exited at intervals ranging from 4 seconds to 12 seconds. When the exit hole was at the highest horizontal point the unloading interval was the longest and when the exit hole was at the lowest horizontal point, the unloading was the fastest. Between the highest and lowest point the unloading rhythm was steady, depending on the rotational speed of the TUD head. To unload 333 spheres/hr a sphere had to be unloaded every 10.8 seconds, thus within the range of 4 to 12 seconds. Thus with better horizontal alignment 10.8 seconds could be achieved.

No spheres were damaged. Figure 65 is a photo of the TUD head during test operation.



Figure 65: Spheres flowing through the head

This test indicated that the TUD unloaded spheres reliably for the full speed range that could be provided by the electric motor's variable speed drive (VSD). The unloading performance of the TUD head was not sensitive to the amount of spheres present in the test unit. One sphere could be unloaded reliably, as well as a large number of spheres, taking into account that the unloading sequence of this test unit was not perfect due to the horizontal alignment, but the conceptual test still proved that the concept is reliable.

More spheres were added to the TUD while the TUD continued to unload spheres reliably. The amount of spheres added on top of the TUD head was increased to four rows. It was found that reliable unloading was achieved with three to four rows of spheres on the TUD head, which was acceptable to maintain the 333 spheres/hour.

The maintenance pipe was inserted at a height of just more than one sphere diameter above the cone (approximately 70 mm). A sphere could thus pass between the maintenance pipe and the cone around the complete maintenance pipe perimeter. The TUD had a steady sequence of sphere intake, causing a disturbance in the sphere-bed around the maintenance pipe. The disturbance helped to maintain a loose sphere bed at the TUD entrance. This led to a constant inflow of spheres onto the TUD head from underneath the maintenance pipe, as can be seen in Figure 66.

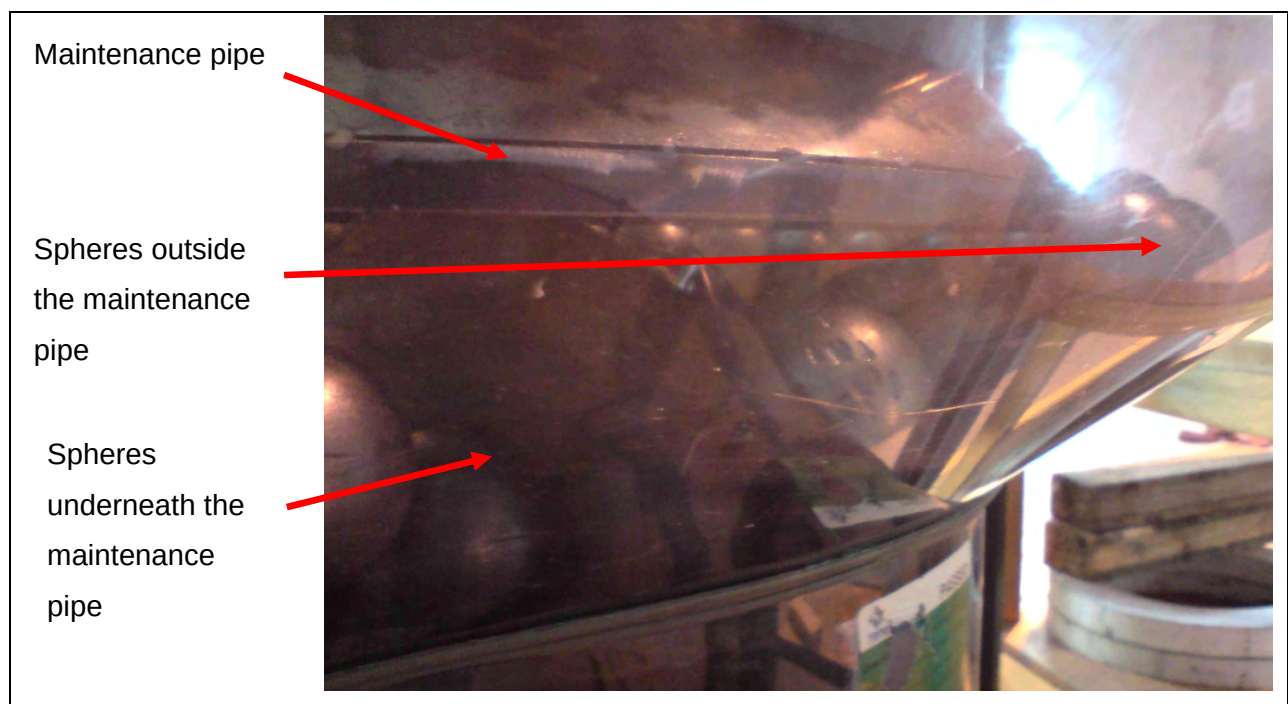


Figure 66: Adjusting the height of the maintenance pipe

The test unit was filled with more spheres until about one third of the capacity of the test unit. This was done to increase the pressure load on the spheres at the entrance opening of the maintenance pipe. After approximately 15 minutes the unloading stopped. A bridge had

formed at the entrance to the TUD head around the servitude pipe, as could be seen in Figure 67.

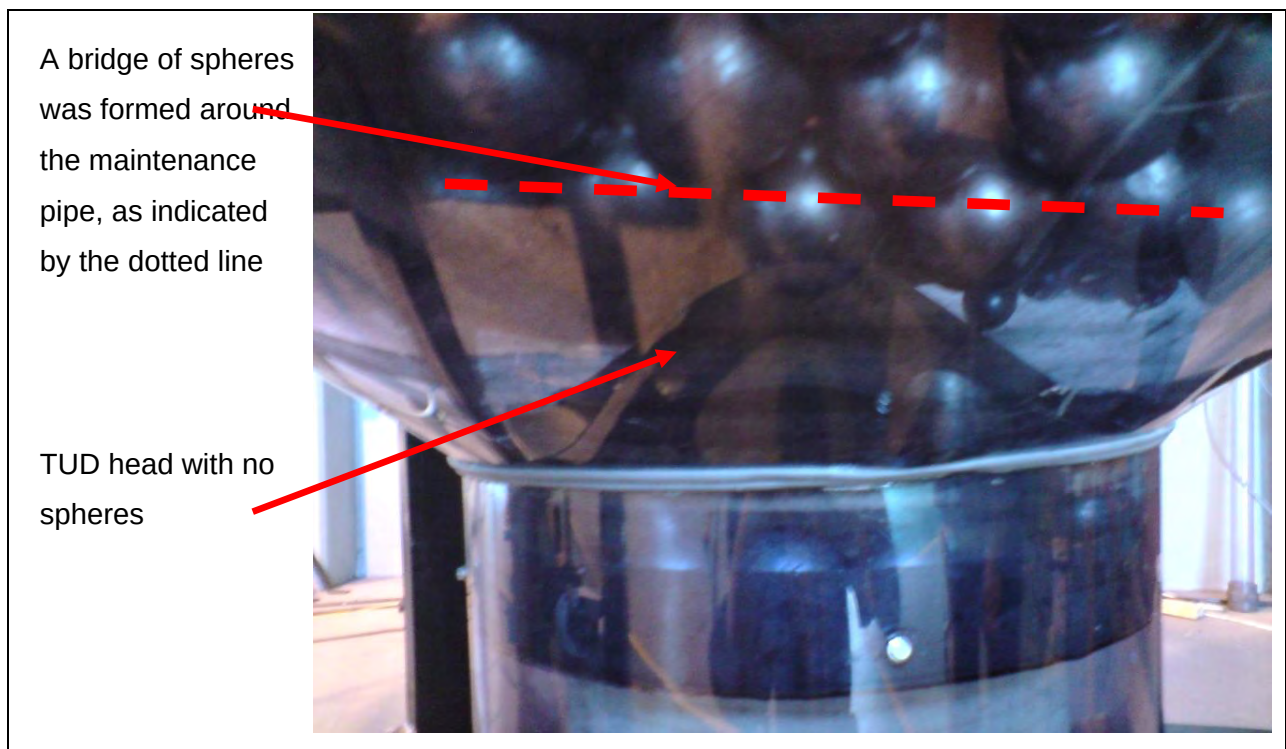


Figure 67: Bridge-forming

The height adjustment of the maintenance pipe caused the bridge. Therefore the pipe was lifted from 70 mm to approximately 85 mm from the cone (distance of 85 mm perpendicular to the cone). After this adjustment the TUD was operated for another 3 hours without bridge-forming. Figure 68 shows a photo taken into the maintenance pipe during operation of the TUD. This pattern of spheres on the TUD head was constantly maintained during operation, which indicated a continuous sphere flow onto the TUD head. It was noted that there had to be three to four layers of spheres on the head for continuous unloading performance:

- Insufficient spheres might allow the TUD to run empty which was under performance;
- Too many spheres caused blockages.

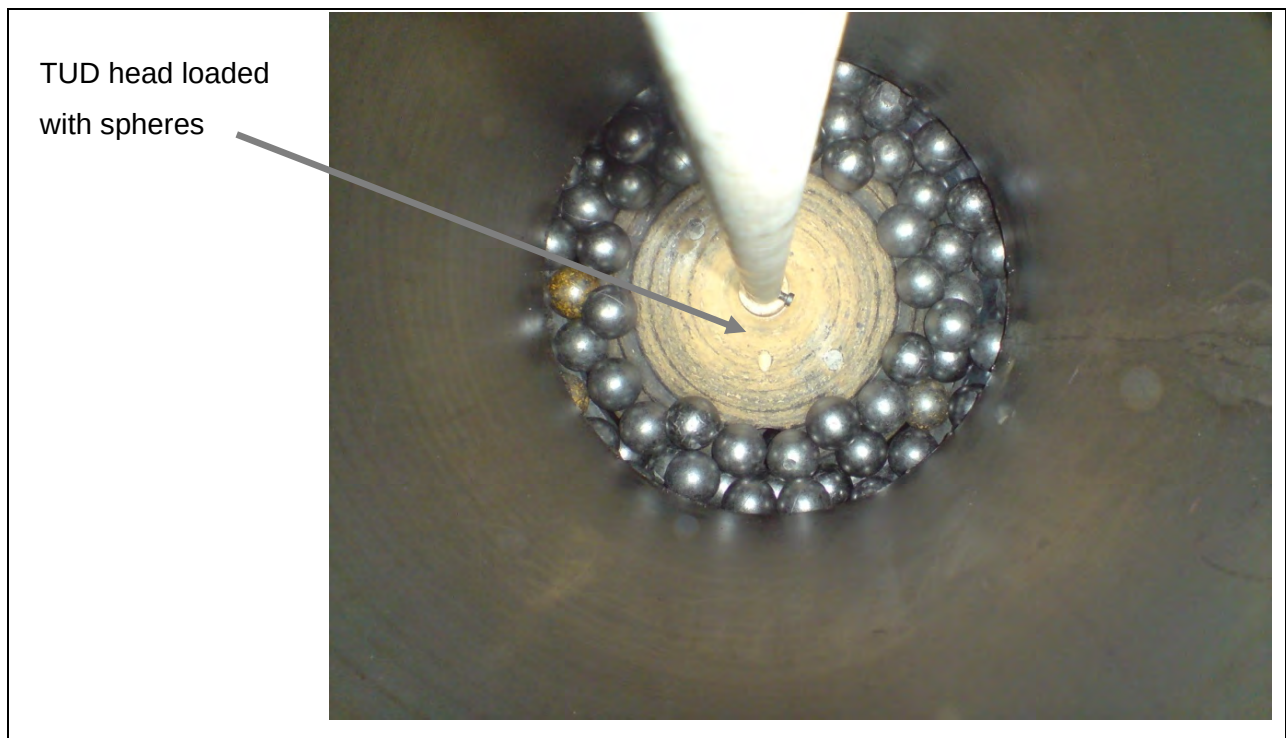


Figure 68: View down the maintenance pipe to TUD head loaded with spheres

Because of the sensitivity of the TUD to its horizontal alignment, the TUD performed better at lower rotational speeds. When the head was rotating slowly, the string of spheres tended not to roll horizontally down towards the exit hole on the second level when the exit was at its lowest horizontal point. However, spheres could still roll down towards the exit and exit at a 4 second interval. Refer to Table 20 for results.

The rotational speed of the TUD head was increased to the maximum allowed by the VSD, approximately 10 min^{-1} . Taking the horizontal misalignment into account, unloading did achieve the 333 spheres per hour, although not at sequentially spaced intervals. The TUD head could thus be rotated at 10 min^{-1} without risking damage the TUD head mechanism or the spheres. Refer to Table 20 for results.

With the TUD being affected by horizontal alignment it was difficult to set the unloading speed exactly. However, by counting the spheres per minute the speed could be adjusted until approximately 334 spheres were unloaded per minute. Table 20 lists the experimental amount of spheres unloaded against the TUD rotational speed. These values were a bit lower (amount of spheres against revolutions per minute) than predicted. This was due to the spheres bumping into each other as they moved around the TUD head on the second level, causing slower sphere feed on the second level.

Table 20 Sphere unloading performance

VSD Setting	Seconds per revolution	Rotational speed (min ⁻¹)	Spheres per minute (average)	Spheres per hour (average)	Seconds between spheres
50	50	1.20	22	1320	2.73
40	60	1.00	16	960	3.75
30	83	0.72	13	780	4.62
20	123	0.49	9	540	6.67
10	244	0.25	5	300	12.00

The counted unloaded spheres per minute as a function of the rotational speed of the TUD can be visualised on a graph as in Figure 69, while the calculated spheres per minute as a function of TUD rotational speed is given in Figure 70. The 333 spheres/minute can thus be achieved.

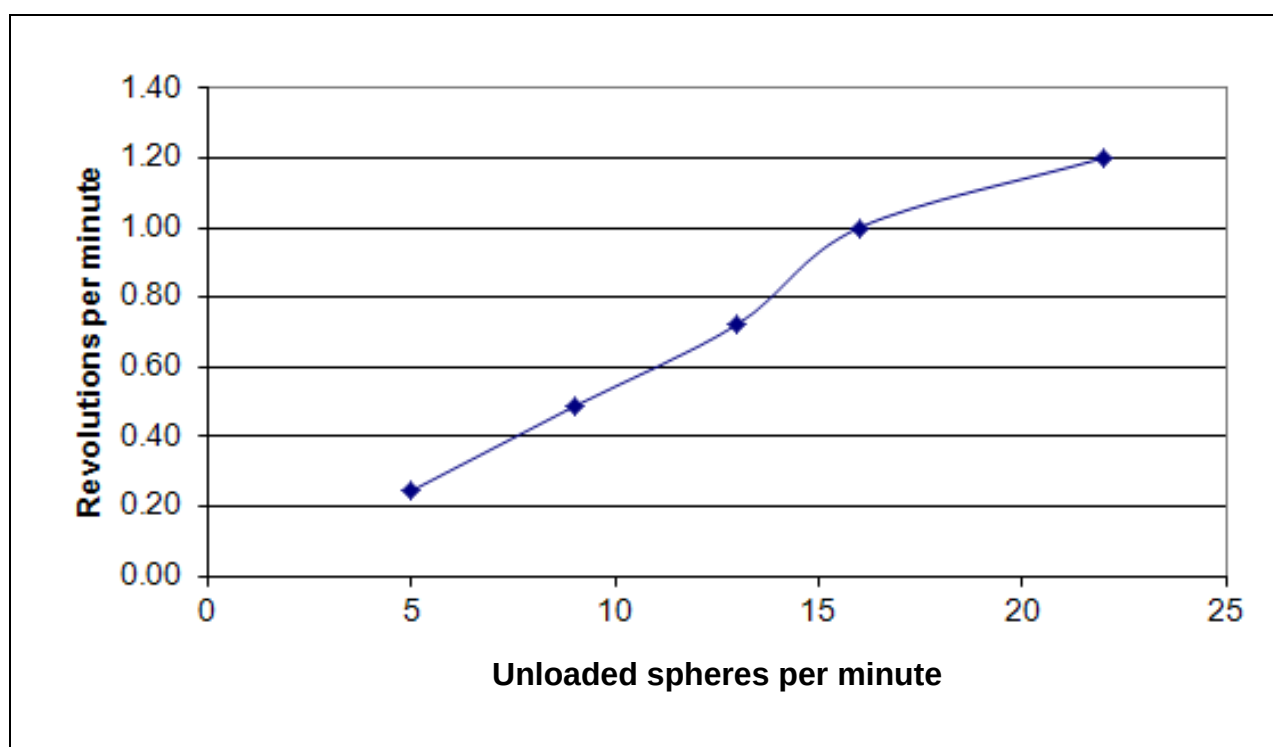


Figure 69: Experimental spheres per minute versus TUD rotational speed

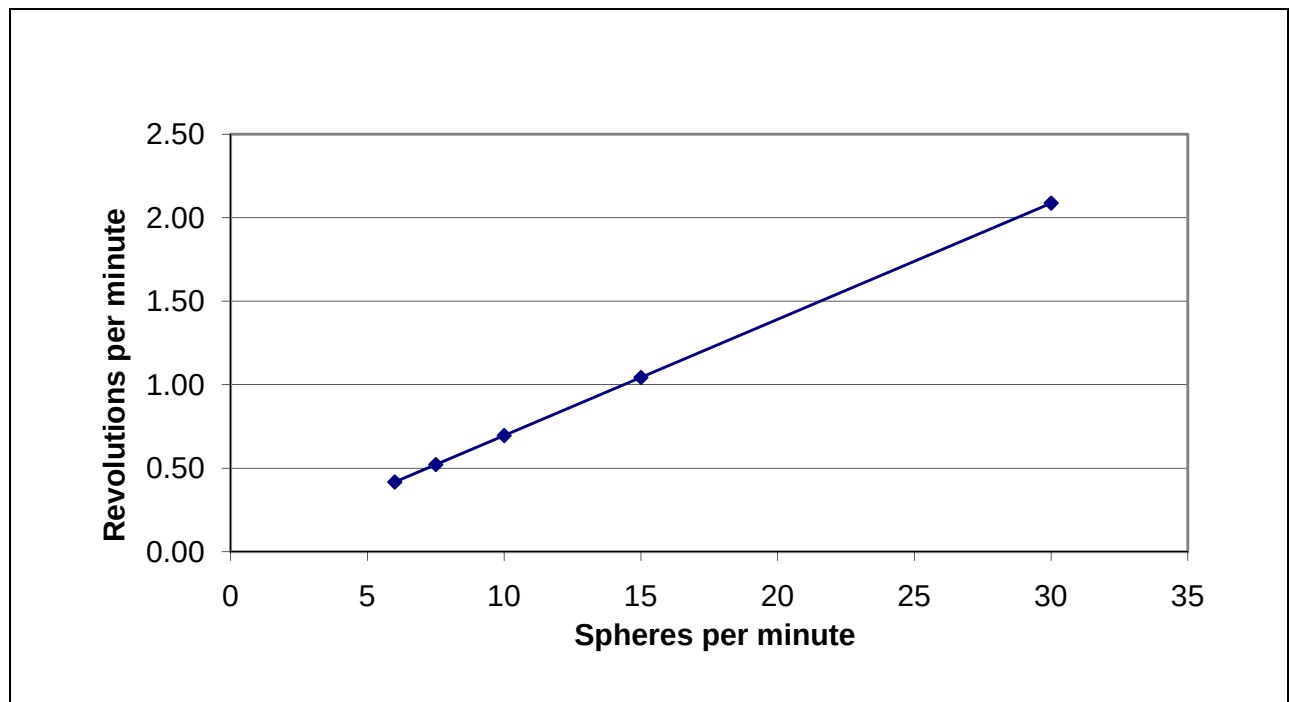


Figure 70: Theoretical spheres per minute versus TUD rotational speed

Broken spheres were added to the test unit. The number of broken spheres made up approximately 5 % of the contents in the test unit, however when they were removed by the separator the pieces were returned back into the test unit.. The sampling size was as indicted in Figure 71. Sizes ranged from 10 mm pieces to approximately 50 mm. The result was that smaller broken sphere pieces were discharged through the separation opening, while larger pieces were transported through the system. The unusable spheres and pieces of spheres are shown in Figure 71.

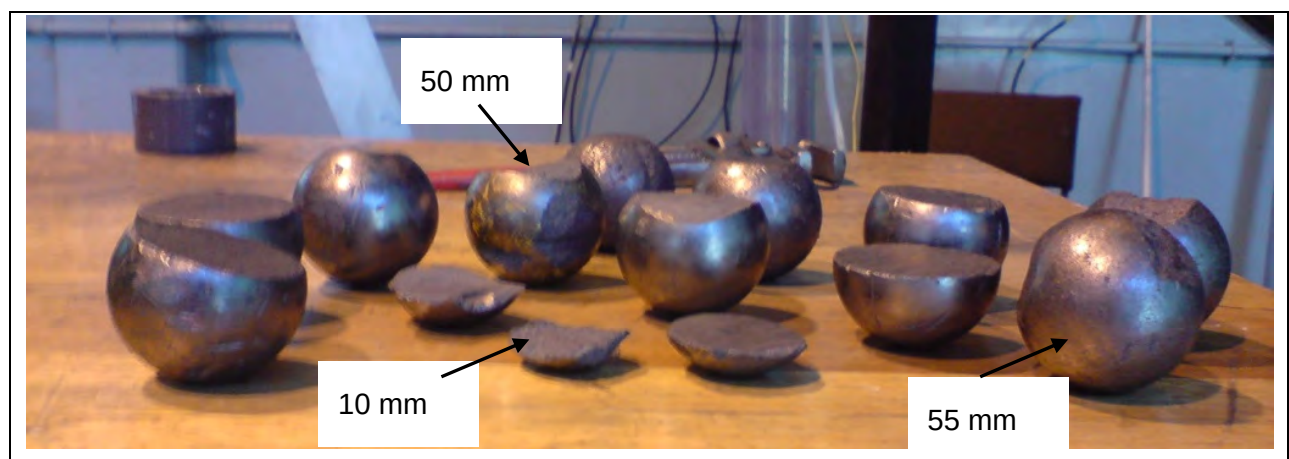


Figure 71: Size range of typical unusable spheres and pieces of broken spheres

In this range of damaged spheres indicated in Figure 72 the smaller pieces were separated, while the larger pieces went through the unloading system together with undamaged spheres.



Figure 72: Unusable sphere size range that could cause blockage in lifting line

The size of the broken spheres that can be removed from the usable spheres is a function of the gap between the TUD head and the ring cone sleeve. Presently the gap is 17 mm, which correlates with the size of broken spheres removed, Figure 73. The separating gap for the sphere pieces might be adjusted in future, if deemed necessary, to separate these broken spheres sizes. However, when the gap is too large, the spheres will be clamped between the TUD head and the ring cone sleeve.

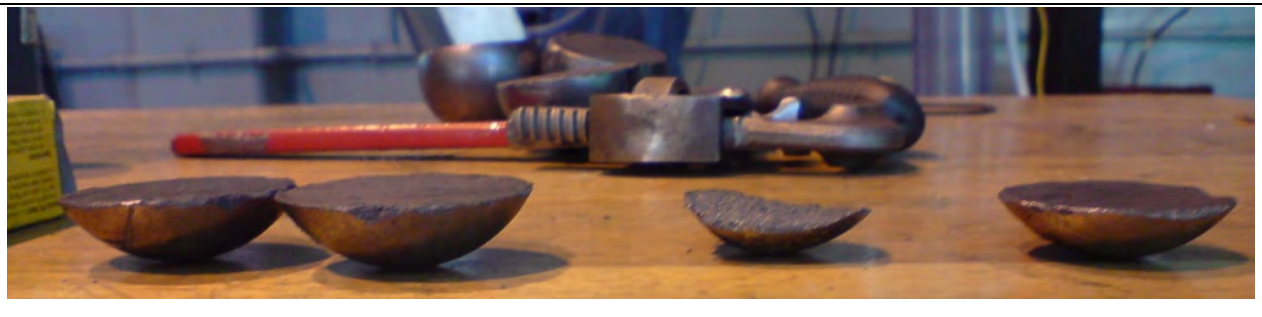


Figure 73: Separated and removed pieces of broken spheres

The servitude pipe height adjustment setting was still at 85 mm. The pieces of broken spheres did not cause blockages underneath the servitude pipe.

The TUD was able to remove all spheres and pieces from the test unit. Spheres and broken sphere pieces slid down to the TUD head, where they were then removed from the test unit. The broken sphere pieces were separated through the discharge openings, while spheres moved through the sphere pipe segments.

The maintenance pipe height adjustment setting was still correct and could be maintained at 85 mm perpendicular to the cone.

For this test the item called “Ring, Cone Sleeve” was installed in both positions, once with the 41°-side facing up and then with and 55°-side facing up. It was found that the 41° angle was too steep which caused the sphere to wedge between the TUD head and the ring. The ring was turned over again with the 55° angle facing up. At 55° (i.e. 35° to the horizontal) no clamping occurred.

5.4 CONCLUSION

The sphere geometry and sphere integrity were maintained during the unloading test, thus no spheres were damaged by the test unit. However, the horizontal misalignment influenced the overall performance of the TUD with regard to sequential timing between spheres. These findings are as follows:

- TUD installation requires accurate horizontal installation, (Test 1A);
- The ring cone must be installed with the 55° angle on the top (resulting in 35° angle), (Test 6).

Further improvement to the concept can be done. Figure 63 indicates humps or obstacles that can be added to the ring cone to prevent spheres from rolling to the lowest point, thus reducing the effect of horizontal misalignment. These humps should space the spheres on the second level to prevent them from touching. Due to these humps, the top hole will position the sphere in the space between the bumps and the sphere will fall into the bottom hole of the TUD head when the hole passes by. The height of the hump should be less than the space between the TUD head and the sphere to prevent sphere clamping between TUD head and the cone sleeve.

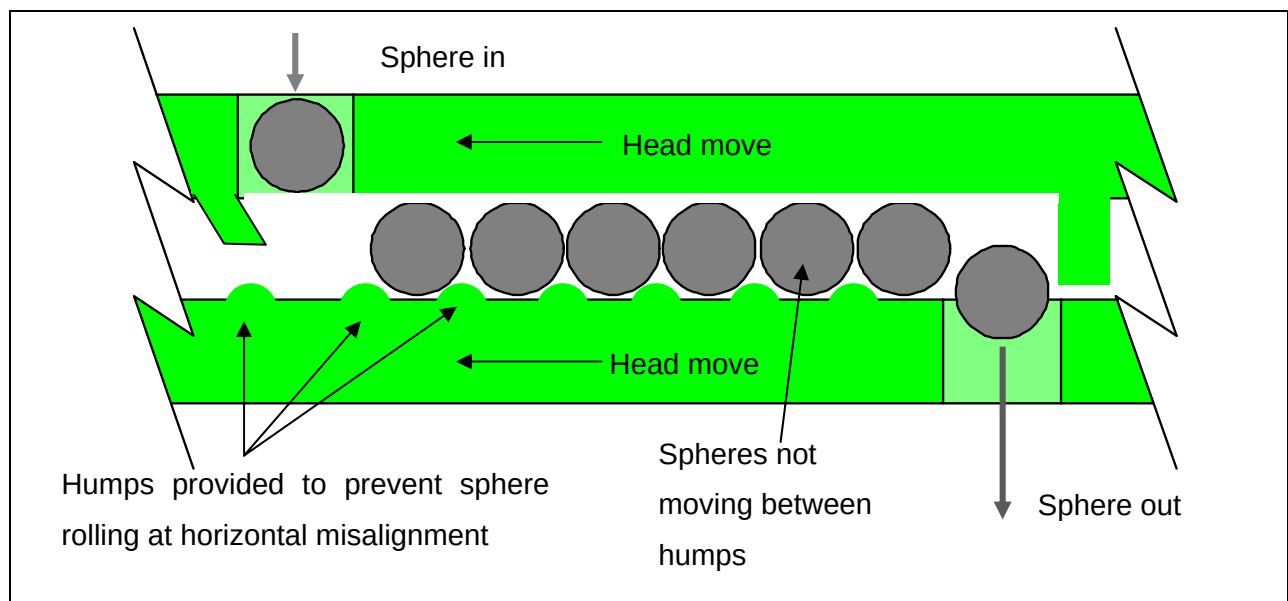


Figure 74: Humps proposal to limit sensitivity for horizontal misalignment

6. CONCLUSIONS

The study started with a description of the environment and geometry within which the tank unloading device must be developed. The tank unloading device functions were identified and listed.

Existing concept designs make use of moving components that will fail in the helium environment. Some of these sphere handling concepts make use of pockets to locate the spheres. But with broken sphere pieces these pockets can be filled with a sphere piece and a sphere, causing a blockage due to an over filled pocket.

Based on the design requirements and interfacing layout two different concepts were identified and tested. Scale models of both the suction and gravity concepts were built to test their operability and control. The gravity concept proved to control the sphere unloading better than the suction concept and was thus selected to develop further.

The incremental developments required for the improvement of the gravity feed concept were a greater challenge than was expected. It was discovered that spheres could be damaged by the rotating head of the improved gravity concept. Attempts were made to protect the sphere against damage by the head by inserting an impact-limiting device between the head and the sphere. A metal and a silicone spring were developed as impact-limiting devices. Both these designs required maintenance:

- Metal due to metal fatigue that would cause the spring to fail;
- Silicone that would not be able to withstand the radioactive environment for as long as metal would.

Maintenance actions must be limited in the nuclear environment. Therefore it was decided to improve the concept further by taking the spheres again through the centre of rotation. The head functions were separated and another row was added to the head for the singulizing function. This improvement was successful in the unloading function, without damage to the spheres. The rotational speed of the head was calculated and the concept was shown to be theoretically viable.

A full-scale test unit was built to verify the design. Some tests were required for the final geometrical layout while other tests were required for verification of the design. Dimensions that were not known before the tests were defined during the tests.

The unloading function was successful regarding the unloading of 333 spheres per hour, but a steady unloading rhythm could not be achieved. The tests indicated that the concept was sensitive to horizontal alignment and therefore a steady unloading sequence could not be achieved. To limit the sensitivity to horizontal alignment the design was improved again by adding humps to the ring, the stationary part around the rotating head. The function of these humps was however not tested in the test unit, but it is believed that they will make the design

fully compliant with all the requirements, because the rhythmic unloading was the only non-compliance.

The 85 mm distance between the cone and the maintenance pipe, measured perpendicular to the cone, will prevent bridge forming, while preventing overfilling of spheres onto the TUD head. This distance of 85 mm maintains three to four rows of spheres on the head which is adequate to maintain a constant sphere unloading.

A robust design was developed and tested with a full-scale test unit for the unloading of nuclear fuel spheres. These spheres can now be unloaded by this tank unloading device from the storage tank without damage to the fuel spheres.

The tank unloading device has been developed with materials that can withstand the nuclear environment. Furthermore the unloading device can operate in a graphite dust filled helium environment. Maintenance requirements have been reduced by placing the maintenance intensive items outside the pressure boundary. Therefore the actuator is situated outside the helium pressure boundary. The actuator is a normal off-the-shelf item used at various places in the plant. For the second concept no special requirements such as angle sensing or torque limitation have been placed on the actuator.

The pressure boundary is penetrated with an item called shaft penetration, which is used in other machinery where the helium pressure boundary is penetrated. Since this proven technology has no effect on the unloading principle, no attention has been given in this study to the shaft penetration.

7. REFERENCES

- [1] Dudley T. et al., The Fuel Handling and Storage System (FHSS) Model for the Pebble Bed Modular Reactor (PBMR) Plant Training Simulator, Proceedings of 3rd International Topical Meeting on High Temperature Reactor Technology, Johannesburg, South Africa, October 2006
- [2] Venter P.J., Mitchell M.N., Fortier F., PBMR reactor design and development, 18th International Conference on Structural Mechanics in Reactor Technology, Beijing, China, August 7-12, 2005
- [3] Matzner D., PBMR Existing and Future R&D Test Facilities, Proceedings of 2nd International Topical Meeting on High Temperature Reactor Technology, Beijing, China, September 2004
- [4] Cachon L. et al., Tribology in High Temperature Helium: First Phase of the CEA Helium Technology Program, Proceedings of 2nd International Topical Meeting on High Temperature Reactor Technology, Beijing, China, September 2004
- [5] Cogliati J.J., Ougouag A. M., Pebble bed reactor dust production model, Proceedings of 4th International Topical Meeting on High Temperature Reactor Technology, Washington, DC USA, September 2008
- [6] Li C.C., Sheehan J.E., Friction and wear studies of graphite and a carbon-carbon composite in air and in helium, Department of Energy, October 1980
- [7] Luo X., Yu S., Sheng X., He S., Graphite friction coefficient for various conditions, Institute of Nuclear Energy Technology, Tsinghua University, Beijing, Science in China, Vol. 44 Supp., 248-252, August, 2001
- [8] Liu J. G., Design and full scale test of the fuel handling system, Nuclear Engineering and Design 218 (2002) 169-178, 2002
- [9] Hong R. et al., Reactor safety and mechanical design for the Annular Pebble Bed Advanced High Temperature Reactor, University of California, Department of Nuclear Engineering, May 19, 2009
- [10] Dong J., Yu S., Concept of pebble bed based HTGR with fast pebble discharge system, Proceedings of 2nd International Topical Meeting on High Temperature Reactor Technology, Beijing, China, September 2004
- [11] Hrovat Dr. M., Grosse K., Manufacture of high corrosion resistant fuel spheres for high temperature pebble bed modular reactors (PBMR), Proceedings of 3rd International

Topical Meeting on High Temperature Reactor Technology, Johannesburg, South Africa, October 2006

- [12] IAEA, INSAG 12, Basic Safety Principles for Nuclear Power Plants, Vienna, Austria, 1999
- [13] U.S.NRC, Next Generation Nuclear Plant Phenomena Identification and Ranking Tables (PIRTs) Volume 1: Main Report, March 2008
- [14] Theymann W., Engel R., Demus H., Ceramic coatings for protection against frictional wear and diffusion welding in HTR helium, Nuclear Engineering and Design 119 (1990) 447-457, 1990
- [15] Petti D. A., et al., Key Differences in the Fabrication, Irradiation of U.S. and German Triso-coated Particle Fuel and their Implications on Fuel Performance, Idaho Falls, Idaho, INEEL/EXT-02-00300, June 2002
- [16] IAEA, INSAG 10, Defence in Depth in Nuclear Safety, Vienna, Austria, June 1996
- [17] Pipeline Engineering, Sphere unloading concept, www.pipelineengineering.com [Website accessed on 16 June 2010]
- [18] Tennis robot, Pick-up mechanism for tennis balls, <http://tennisrobot.org/pick-up-mechanism/> [Website accessed on 16 June 2010]
- [19] Tippmann Pneumatics Inc, A-5 Owner's manual CO₂ powered Paintball Marker, Ford Wayne, USA
- [20] V&P Scientific, Inc., Scientific Levitation Ball Loader VP725C, Operating Instructions, San Diego, USA
- [21] Oak Ridge National Laboratory, Conceptual design of the pebble bed reactor experiment, Oakridge, USA, May 17, 1962
- [22] Tian J., Loading and unloading scheme of the ordered bed modular reactor, 3rd International Topical Meeting on High Temperature Reactor Technology, Johannesburg, South Africa, October 1, 2006
- [23] Wacker Silicones, Elastosil brochure, www.wacker.com [Website accessed on 15 August 2010]
- [24] James Walker, brochure for Shieldseal Elastomers for use with ionising radiation, www.jameswalker.biz [Website accessed on 15 August 2010]
- [25] Silicone Engineering, Cellular Silicone Elastomer, www.silicone.co.uk [Website accessed on 15 August 2010]
- [26] Rogers Corporation, www.rogerscorp.com [Website accessed on 15 August 2010]

- [27] Dupont, Design Handbook Vespel S Line, www2.dupont.com [Website accessed on 15 August 2010]
- [28] Shigley J. E., Mechanical Engineering Design, 1st Metric Edition, McGraw-Hill, New York, 1986
- [29] Wolfram Mathworld, <http://mathworld.wolfram.com/SpherePacking.html> [Website accessed on 25 October 2010]
- [30] Harmonic Drive, Catalogue for the FHA series, www.harmonicdrive.net [Website accessed on 25 August 2010]
- [31] Leine & Linde, Catalogue for angle encoders, www.mclennan.co.uk [Website accessed on 25 August 2010]
- [32] Heid Antriebstechnik, Catalogue for Electromagnetic Stationary Field Multi-Disc Clutch, www.heid-antriebstechnik.at [Website accessed on 25 August 2010]
- [33] Shelley Steels, EN24 Material properties, <http://www.shelleysteels.co.uk/en24.htm> [Website accessed on 12 August 2010]
- [34] PBMR Fuel Spheres, Product Specification, PB-FTP-0001, Centurion, South Africa

8. APPENDIX A

A 1. PURPOSE

A1.1 Purpose of this calculation record

The purpose of this calculation record is to capture all the calculations that have been done to date on the TUD Drive Assembly and the head kinematics during operation of the TUD Drive Assembly.

A1.2 Scope of this calculation record

The torque needed to operate the TUD at the required sphere discharge rate will be calculated as well as the capabilities of the drive shafts. It will also be determined if a spring is required to protect the sphere against damage should the sphere be jammed between the head and housing. If so, the spring types and their sizes will be determined.

A1.3 Applicability

This document is applicable to the TUD drive assembly and the TUD head.

A 2. APPLICABLE AND REFERENCE DOCUMENTATION / DATA

- [1] Petti D. A., et al, Key Differences in the Fabrication, Irradiation of U.S. and German Triso-coated Particle Fuel and their Implications on Fuel Performance, INEEL/EXT-02-00300, June 2002
- [2] Bonuskor Steel, Catalogue
- [3] Mechanics of Materials, ISBN 0 534-37133-7, 5th edition
- [4] Mechanical Engineering Design, ISBN 0-07-056898-7, 1st Metric

A 3. ASSUMPTIONS / EXCLUSIONS

The following assumptions were made:

- 1) A height of more than 1m of spheres will not have a weight effect on the head ;
- 2) No friction in system, except where the friction of the spheres on the head is stated;
- 3) Head has only one exit hole.

A4. NUMENCLATURE

A4.1 SYMBOLS

α	Angular acceleration	rad/s ²
ρ	Density	kg/m ³
σ	Stress	MPa
τ	Shear stress	MPa
ω	Angular velocity	rad/s
μ	Friction coefficient	
ν	Poisson's ratio	
A	Area	m ²
D	Diameter	m
E	Modulus of elasticity	GPa
F	Force	N
I	Moment of inertia	
J	Moment of inertia around shaft center	m ² kg
Pf	Packaging factor	
S	Section modulus	m ³
Sf	Safety factor	
T	Torque	Nm
V	Volume	m ³
W	Weight	N
b	Breadth	m
k	Spring stiffness	kN/m
l	Length	m
m	Mass	kg
n	number of ...	
r	Radius	m
t	Time in seconds	sec
v	Linear velocity	m/s
Δx	Distance change	m

A4.2 SUBSCRIPTS

μ	Friction Coefficient
L	Life
a	Amplitude
act	Actuator
c	Coil (spring)
cut	Value where sphere will be cut
e	Effective
f	Fatigue
gr	Graphite
i	Increment
im	Impact
m	Mean
max	Maximum
min	Minimum
mw	music Wire
o_	Outer of _ (something)
pl	Plate (spring)
r	Rubber
se	Endurance limit in shear
s	Sphere
sh	Shaft
sp	head
sq	Square

sr	Silicone rubber
st	Steel
sy	Torsional yield
step	Used in algorithm for time interval
t	Total
th	Thickness
tl	Torque limiter
tn	Turns for coil spring
ut	Ultimate in tension
xx	Material not yet defined
y	Yield
0	At time is 0 seconds (beginning / first of)
1	Second of -- (something)
24	EN 24 type steel

A5. CALCULATION TECHNIQUE

- 1) Calculate moment of inertia of the head.
- 2) Calculate the torque required to accelerate the head, and include:
 - 2.1) The minimum angle needed to accelerate the head to optimum speed at the first sphere exit.
 - 2.2) Calculate the friction of the spheres on the head.
- 3) Determine the impact properties of the sphere.
- 4) Determine the k-value of the spring for the stuck sphere between head and housing.
- 5) What spring should be used to prevent sphere damage if a sphere gets stuck.
- 6) The possibility to use silicone as a spring.
- 7) Calculate the weakest part in the shaft and the torque that could be transmitted.
- 8) Determine the maximum acceleration time for acceleration before the shaft breaks.

A6 CALCULATING HEAD IMPACT RESULTS

A6.1 CALCULATION METHODOLOGY

The head rotation speed will be calculated with regard to the required TUD sphere discharge. The maximum torque to cut a sphere will be used to determine the maximum head deceleration as the result of the head's moment of inertia.

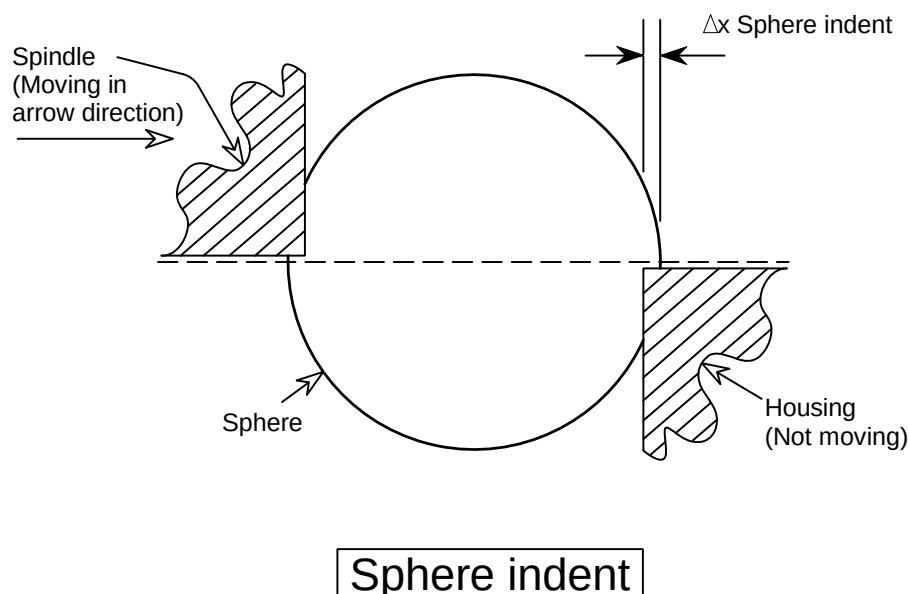


Fig. A1 Sphere indentation

A6.2 INPUT DATA

Minimum required sphere discharge time:

$$\Delta t_s := 6 \text{ sec}$$

Head start (or stop) speed:

$$\omega_0 := 0 \cdot \frac{\text{rad}}{\text{s}}$$

Max allowable sphere force [1]:

$$F_{\text{cut}} := 18 \text{ kN}$$

Head shaft mass:

$$m_{\text{sh}} := 37 \text{ kg}$$

Head mass:

$$m_{\text{sp}} := 224 \text{ kg}$$

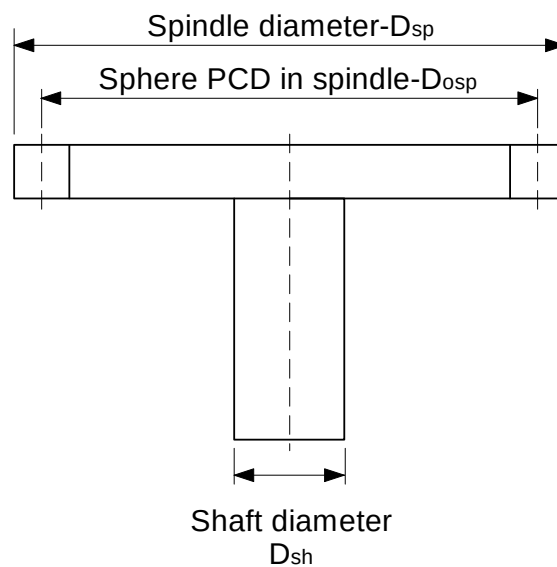


Fig. A2 Head dimensions

Head sphere PCD:

$$D_{\text{sp}} := 640 \text{ mm}$$

Head sphere radius:

$$r_s := \frac{D_{\text{sp}}}{2}$$

$$r_s = 0.32 \text{ m} \quad (1)$$

Head outer diameter:

$$D_{\text{osp}} := 720 \text{ mm}$$

Shaft diameter:

$$D_{\text{sh}} := 80 \text{ mm}$$

There are two different head diameters:

- the larger (720mm) is to calculate the head momentum change
- and the smaller (640mm) to calculate the force on the sphere with the sphere PCD.

A7.1 TORQUE REQUIRED FOR HEAD ACCELERATION

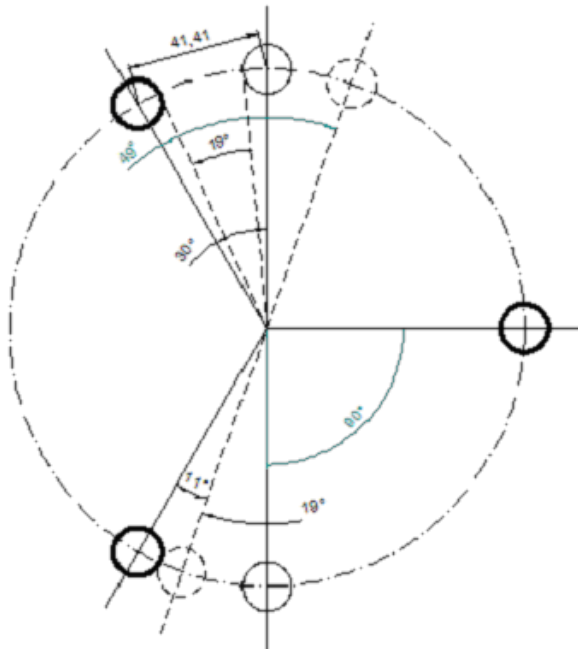
$$\omega_{\text{sp}} := \frac{2 \cdot \pi}{\Delta t_s} \quad \omega_{\text{sp}} = 1.047 \frac{1}{s} \quad (2)$$
$$\text{Shaft moment of inertia: } J_{sh} := 0.5 \cdot m_{sh} \cdot \left(\frac{D_{sh}}{2} \right)^2 \quad (3)$$
$$\text{Head moment of inertia: } J_{sp} := 0.5 \cdot m_{sp} \cdot \left(\frac{D_{osp}}{2} \right)^2 \quad (4)$$
$$\text{Total moment of inertia: } J_t := J_{sp} + J_{sh} \quad J_t = 14.545 \text{ m}^2 \cdot \text{kg} \quad (5)$$


Fig. A3 Head start position

Acceleration angle: $\theta_a := 19^\circ$

$$\text{Acceleration for } 19^\circ \text{ angle: } \alpha_a := \frac{\omega_{sp}^2 - \omega_0^2}{2 \cdot \theta_a} \quad \alpha_a = 1.653 \frac{1}{s^2} \quad (6)$$
$$\text{Acceleration for } 19^\circ \text{ angle:} \quad t_{19} := \frac{\omega_{sp} - \omega_0}{\alpha_a} \quad t_{19} = 0.633 \text{ s} \quad (7)$$
$$\text{Torque for deceleration:} \quad T_a := J_t \cdot \alpha_a \quad T_a = 24 \cdot \text{N} \cdot \text{m} \quad (8)$$

A7.2 TORQUE REQUIRED FOR FRICTION OF SPHERES ON THE HEAD

To determine the amount of torque required to start rotating the head a full load of spheres will cause friction and thus higher torque is needed to accelerate the head. In this section it is assumed that 1meter of spheres on top of the head would be effective. A value of $\mu := 0.6$ is assumed as effective.

A7.2.1 INPUT DATA

Height of spheres on head:	$L_s := 1\text{m}$	[Assume high value]
Friction coefficient:	$\mu := 0.6$	[Assume high value]
Packing factor:	$P_f := 0.6$	[2]
Graphite density:	$\rho_{gr} := 1740 \frac{\text{kg}}{\text{m}^3}$	

A7.2.2 DETAIL CALCULATIONS FOR THE SPHERE FRICTION ON THE HEAD

Head area:	$A_{sp} := \frac{\pi \cdot D_{sp}^2}{4}$	(10)
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Sphere volume (with packaging factor):	$V_s := \frac{\pi \cdot D_{sp}^2}{4} \cdot L_s \cdot P_f$	(11)
--	---	------

Mass of sphere volume:	$M := V_s \cdot \rho_{gr}$	(12)
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Weight of sphere volume:	$W := M \cdot g$	$W = 3294 \text{ N}$	(13)
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Force on head:	$F_\mu := W \cdot \mu$	(14)
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Shear pressure on head:	$P_\mu := \frac{F_\mu}{A_{sp}}$	(15)
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The pressure (P_μ) is acting on the whole top area of the head. An amount of torque will be required to turn the head under this pressure. Since the pressure is not on an actual radius the torque can be found by integrating the radius over the hole top area of the head. See Fig. 4.

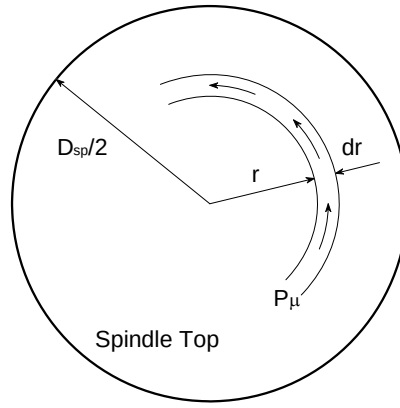


Fig. A4 Diagram of a head top indicating the friction radius

Torque required:

$$T_{\mu} := \int_0^{D_{sp} \cdot 0.5} r \cdot (\pi \cdot r \cdot P_{\mu}) dr \quad T_{\mu} = 210.8 \cdot \text{N} \cdot \text{m} \quad (16)$$

Total torque required:

$$T_{ta} := T_a + T_{\mu} \quad T_{ta} = 234.84 \cdot \text{N} \cdot \text{m} \quad (17)$$

A8 CALCULATING THE SPHERE PROPERTIES AT IMPACT

Torque to cut:

$$T_{cut} := \frac{F_{cut} \cdot D_{sp}}{2} \quad T_{cut} = 5760 \cdot \text{N} \cdot \text{m} \quad (18)$$

Acceleration to cut:

$$\alpha_{cut} := \frac{T_{cut}}{J_t} \quad \alpha_{cut} = 396.018 \frac{1}{s^2} \quad (19)$$

Impact time to cut the sphere:

$$\Delta t_{im} := \frac{\omega_{sp}}{\alpha_{cut}} \quad \Delta t_{im} = 0.003 \text{ s} \quad (20)$$

Head speed at sphere impact:

$$v_{sp} := \frac{\omega_{sp} \cdot D_{sp}}{2} \quad v_{sp} = 0.335 \frac{\text{m}}{\text{s}} \quad (21)$$

Sphere indentation after impact:

$$\Delta x_{im} := v_{sp} \cdot \Delta t_{im} \quad \Delta x_{im} = 0.89 \cdot \text{mm} \quad (22)$$

Sphere spring constant at cutting point:

$$k_s := \frac{F_{cut}}{\Delta x_{im}} \quad k_s = 20313.278 \cdot \frac{\text{kN}}{\text{m}} \quad (23)$$

Conclusion

The impact time of the head impact on the sphere must be longer than 0.003 seconds to be able to maintain sphere integrity.

A9 EFFECT OF MOMENTUM CHANGE AND ACTUATOR TORQUE INPUT IN THE CASE WHERE A SPHERE IS JAMMED

To lengthen the impact time on the sphere the effect of a spring has been investigated. The the first part of the following section is the design of the actuator to provide the torque to drive the head. The rotating head will then be used to determine the possibility to lengthen the impact time by introducing a spring in the design.

A9.1 Actuator and Torque Limiter Properties

Actuator Properties:

FHA - 50A - XX50 (or substitute)

Rated output torque: 313Nm

Continuous stall torque: 438Nm

Maximum output torque: 1577Nm

For the concept design this actuator was selected because it can continuously provide the required torque to rotate the head. However, the smaller one in the series would have been sufficient, but this one was selected because it is also used in other designs, so only one spare is to be held in the plant store.

Maximum torque: $T_{\max \text{Act}} := 1577 \text{ N}\cdot\text{m}$

Maximum force on the sphere: $F_{\text{MAX}} := \frac{T_{\max \text{Act}}}{r_s} \quad F_{\text{MAX}} = 4928.125 \text{ N} \quad (24)$

Continuous torque: $T_{\text{CONST}} := 438 \text{ N}\cdot\text{m}$

Continuous force on the sphere: $F_{\text{CONST}} := \frac{T_{\text{CONST}}}{r_s} \quad F_{\text{CONST}} = 1368.75 \text{ N} \quad (25)$

Torque limiter:

FM 80 (Heid Antriebstechnik)

Below max shaft torque: 1250Nm

Maximum torque: $T_{\max \text{tl}} := 1250 \text{ N}\cdot\text{m}$

Maximum force on the sphere: $F_{\text{MAXTL}} := \frac{T_{\max \text{tl}}}{r_s} \quad F_{\text{MAXTL}} = 3906.25 \text{ N} \quad (26)$

Continuous torque: $T_{\text{RATED}} := 313 \text{ N}\cdot\text{m}$

Continuous force on the sphere: $F_{\text{RATED}} := \frac{T_{\text{RATED}}}{r_s} \quad F_{\text{RATED}} = 978.125 \text{ N} \quad (27)$

Damping force: acting at sphere radius due to sphere friction. $F_e := \frac{T_{\mu} \cdot 2}{D_{\text{sp}}} \quad F_e = 658.7 \cdot \text{N} \quad (28)$

The units must be removed from the values because MathCAD cannot solve the iteration if units are added to the values.

$$\omega_{\text{init}} := 1.047$$

$$F_{\max} := \frac{F_{\text{MAX}}}{N} \tag{29}$$

$$F_{\text{const}} := \frac{F_{\text{CONST}}}{N} \tag{30}$$

$$F_{\text{maxtl}} := \frac{F_{\text{MAXTL}}}{N} \tag{31}$$

$$F_{\text{rated}} := \frac{F_{\text{RATED}}}{N} \tag{32}$$

$$F_{\text{eff}} := \frac{F_{\text{e}}}{N} \tag{33}$$

$$\Delta t_{\text{step}} := 0.00001$$

$$k_{\text{t}} := 8000 \frac{\text{kN}}{\text{m}}$$

A9.2 DETERMINE THE RATIO FOR THE SPRING STIFFNESS AND TORQUE LIMITER CUT-OFF VALUE

A9.2.1 ALGORITHM METHODOLOGY

To protect the sphere against damage a spring has been built into the system between the sphere and the head. In the figure below the ball denotes the head and the earth bar the housing. The spring has been placed between them.

In the following algorithm the previous α (acceleration) value is used to calculate the new velocity (ω) and then the new distance (x). The acceleration is calculated from the formula $\Sigma T = J \cdot \alpha$. Three forces are used in this calculation namely the spring force, the actuator force and the friction force. All the forces are related to the PCD of the sphere hole in the head (head diameter of 640mm). The actuator force has a cut-off limit. The cut-off limit is the torque limiter between the drive and the head.

The force on the spring will only be active if there is displacement on the spring. The friction force must always be against the direction of rotation and therefore the force is positive with ω larger than zero.

After impact the actuator force equals the spring and friction force for head movement, hereafter the force on the head from the actuator will be equal to the torque limiter setting.

By changing the k-value of the spring the Δx travel and the stop time can be adjusted. The k-value must be chosen to keep the force acting on the sphere below the sphere cutting force, as indicated in the graphs below.

Assume that the sphere has to be protected fully by the spring and no sphere indentation is acceptable.

By assuming that the head and penetration shafts do not need any protection, it would only be necessary to protect the sphere. The question is how much the torque limiter cut-off needs to be. For continuous drive of the head the actuator must drive against the friction of the spheres on the head and an actuator was selected with a rated torque of more than the friction torque.

Note that the algorithm works without units, therefore right through the algorithm the units have been removed, but afterwards added again.

A9.2.2 INPUT DATA

Head sphere radius:

$$r_{sp} := \frac{D_{sp}}{2} \quad (34)$$

Assume a spring constant for the spring in the housing:

Spring constant for spring:

$$k_t := 2500 \frac{\text{kN}}{\text{m}}$$

Assume torque limiter setting of 300Nm. This is just above the required continuous torque for driving against friction on the head.

Torque limiter max torque:

$$T_{tl} := 300 \text{N} \cdot \text{m}$$

Torque limiter force on sphere:

$$F_{tl} := \frac{T_{tl}}{D_{sp} \cdot 0.5} \quad F_{tl} = 937.5 \text{ N} \quad (36)$$

Time step for loop:

$$\Delta t_{\text{step}} := 0.00001$$

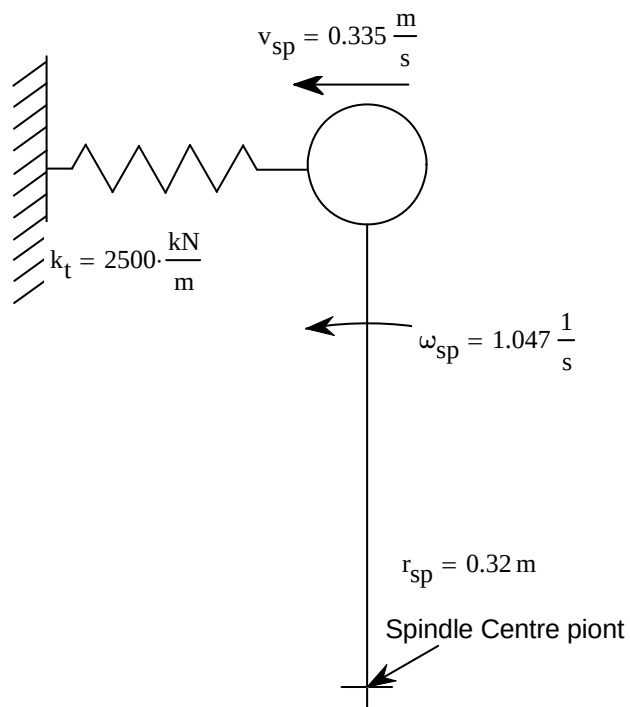


Fig. A5 Spring force diagram

The figure below shows a schematic diagram of the spring with the forces acting on the spring and the resulting distance of spring travel, or deflection (Δx). The length used in the calculations is the indicated length L_v . The direction of the forces are indicated.

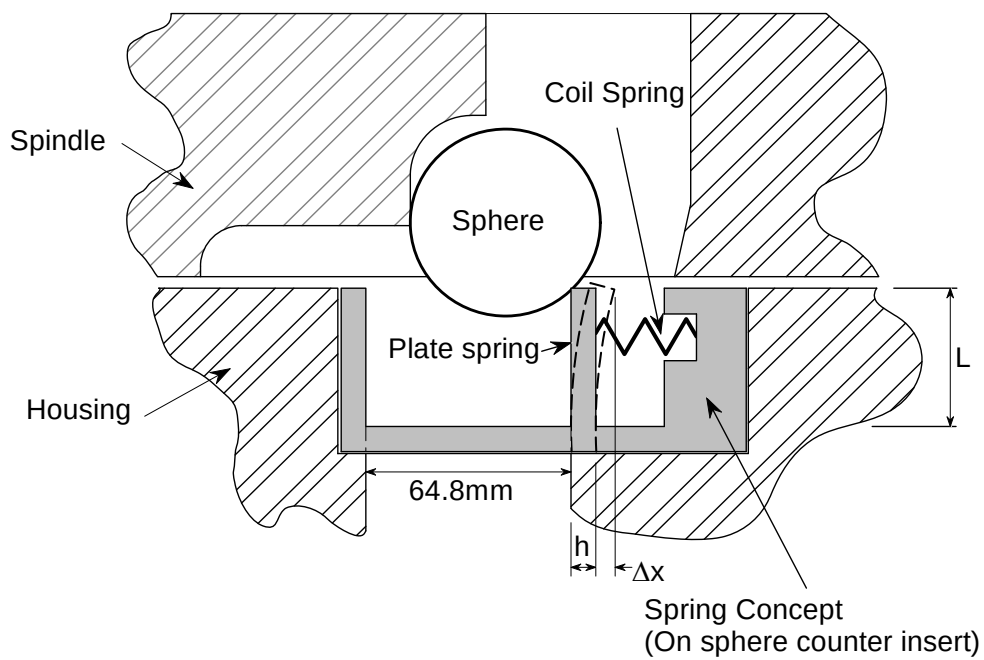


Fig. A6 Spring Concept

A9.2.3 DETAIL CALCULATIONS

$$\begin{pmatrix} x \\ \omega \\ \alpha \\ F_{st} \\ Fact \end{pmatrix} := \begin{array}{l} \omega_0 \leftarrow \omega_{sp} \cdot s \\ x_0 \leftarrow 0 \\ F_{st_0} \leftarrow 0 \\ F_{tl_0} \leftarrow 0 \\ \alpha_0 \leftarrow 0 \\ i \leftarrow 0 \\ \text{while } i \cdot \Delta t_{step} < 0.3 \\ \quad i \leftarrow i + 1 \\ \quad \omega_i \leftarrow \omega_{i-1} + \alpha_{i-1} \cdot \Delta t_{step} \\ \quad \omega_i \leftarrow 0 \text{ if } |\omega_i| < 10^{-4} \\ \quad x_i \leftarrow x_{i-1} + \omega_i \cdot \frac{D_{sp} \cdot 0.5}{m} \cdot \Delta t_{step} \\ \quad F_{st_i} \leftarrow x_i \cdot k_t \cdot \left(\frac{m}{N} \right) \cdot (x_i \geq 0) \\ \quad F_{fric} \leftarrow \begin{cases} \frac{F_e}{N} & \text{if } \omega_i > 0 \\ -\frac{F_e}{N} & \text{otherwise} \end{cases} \\ \quad Fact_i \leftarrow \begin{cases} F_{st_i} + F_{fric} & \text{if } \omega_i = \omega_{sp} \cdot s \\ \frac{F_{tl}}{N} & \text{otherwise} \end{cases} \\ \quad Fact_i \leftarrow \frac{F_{tl}}{N} \text{ if } Fact_i > \frac{F_{tl}}{N} \\ \quad \alpha_i \leftarrow (Fact_i - F_{st_i} - F_{fric}) \cdot \frac{D_{sp} \cdot 0.5}{J_t} \cdot (m \cdot kg) \\ \quad \begin{pmatrix} x \\ \omega \\ \alpha \\ F_{st} \\ Fact \end{pmatrix} \end{array}$$

i-steps for time:

$$i := 0 .. \text{last}(x)$$

time as per i-steps:

$$t_i := i \cdot \Delta t_{step} \cdot s$$

Unit implementation:

$$\begin{pmatrix} x \\ \omega \\ \alpha \\ F_{st} \\ \text{Fact} \end{pmatrix} := \begin{pmatrix} x \cdot m \\ \omega \cdot \frac{1}{s} \\ \alpha \cdot \frac{1}{s^2} \\ F_{st} \cdot N \\ \text{Fact} \cdot N \end{pmatrix}$$

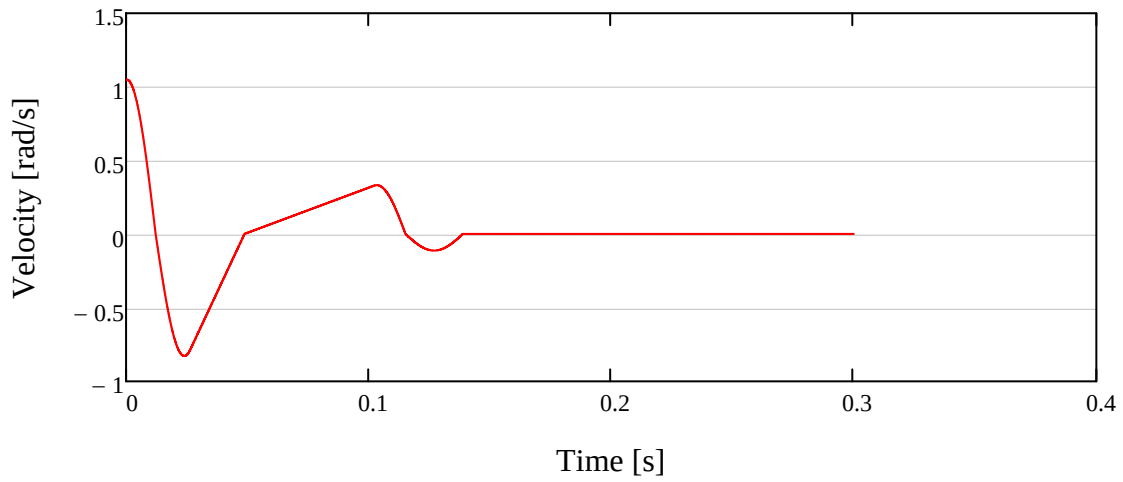


Fig. A7 Spindle Angular Velocity

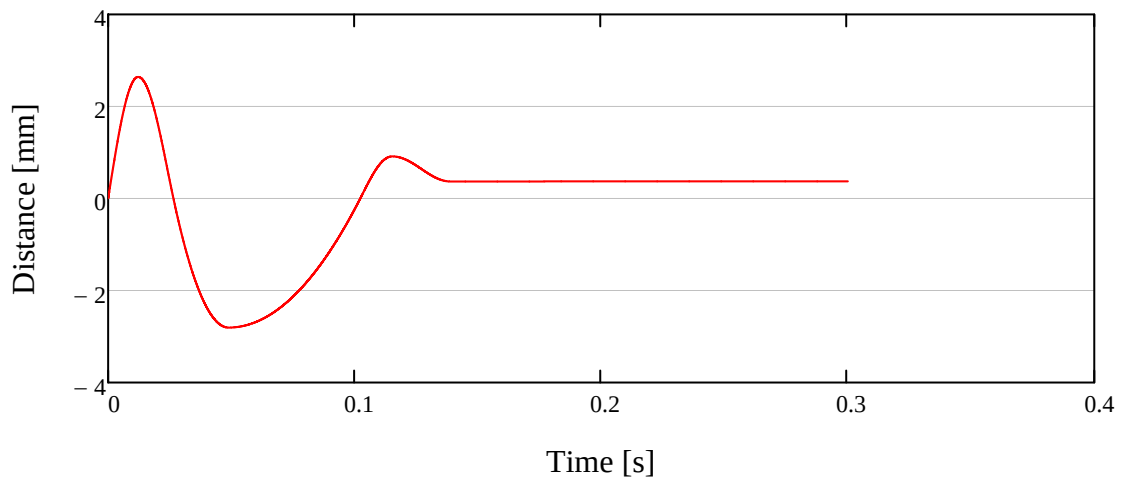


Fig. A8 Spring Deflection

The correct torque limiter setting will allow the head to be turned in the other direction of rotation after impact as can be seen on the graph. With the head moving through the 0mm into the negative implies that the head releases the sphere again. Should the sphere be stuck it will thus be removed.

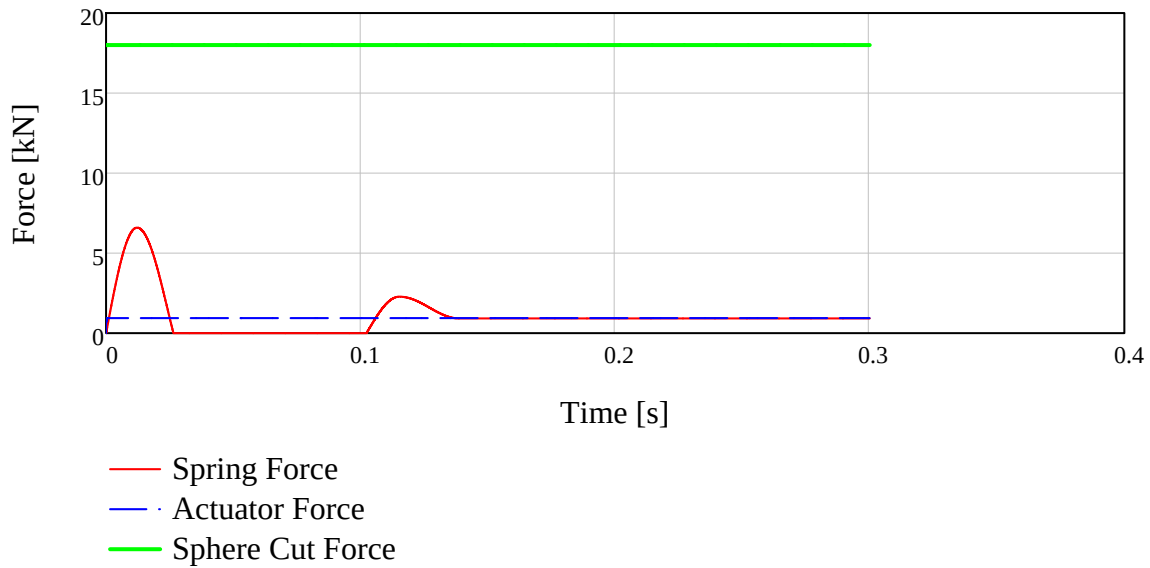


Fig. A9 Combined forces

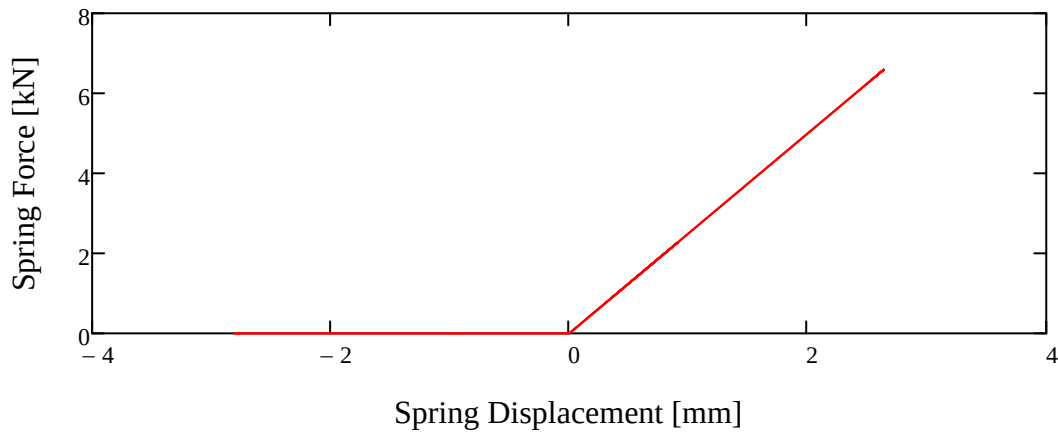


Fig. A10 Spring Force / Displacement Curve

A9.2.4 CONCLUSION

It is stated that the torque limiter setting will be above the constant drive torque. A value of 300 Nm for the torque limiter was set. The question is what must the spring constant be? The following is required:

- "re-bounce" of the head;
- Force on sphere below 18 kN.

For a torque limiter value of 300 Nm and a spring constant of more than 600 kN/m there will be re-bounce. A spring constant of 19000 kN/m will give a sphere force of 18 kN.

A9.3 SPRING CALCULATION

From the geometrical three dimensional model it is also possible to build the spring into the housing. The following calculations are according a concept to build the spring into the housing above the sphere counter insert. The sphere counter insert is a removable item which means that the spring is now replaceable should the spring fail.

For these calculations the output values from section 9.1.3 for the spring force (F_v) and spring travel distance (Δx) will be used to determine the spring.

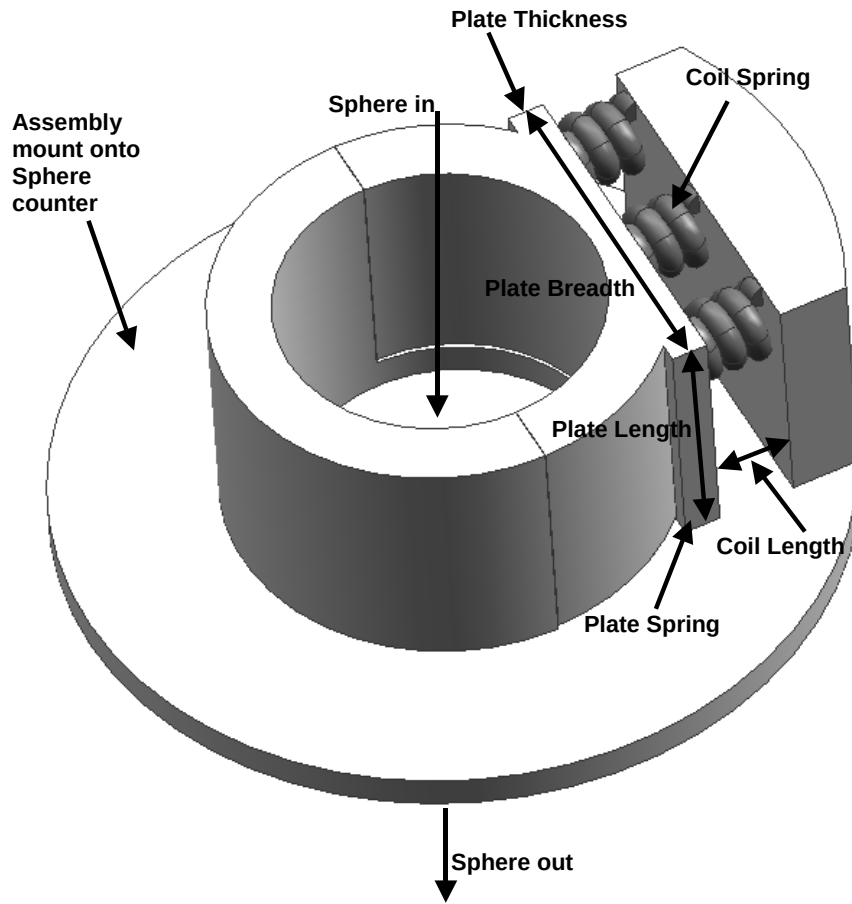


Fig. A11 Mechanical Concept Lay-out

A9.3.1 SPRING DIMENSIONS FOR PLATE AND COIL SPRING

In figure 13 is a schematic lay-out of the spring assembly. The total spring assembly deflection is D_x . The total space available for the length of the plate spring is L_t and the active plate spring length is on the center line of the coil spring. The coils are positioned at:

$$L_{pl} = L_t - y \quad (37)$$

$$\text{where } y = \frac{D_c}{2}. \quad (38)$$

The coil diameter (D_c) is a function of the plate spring length (b_v) and the number of coils (n_r) to be fitted.

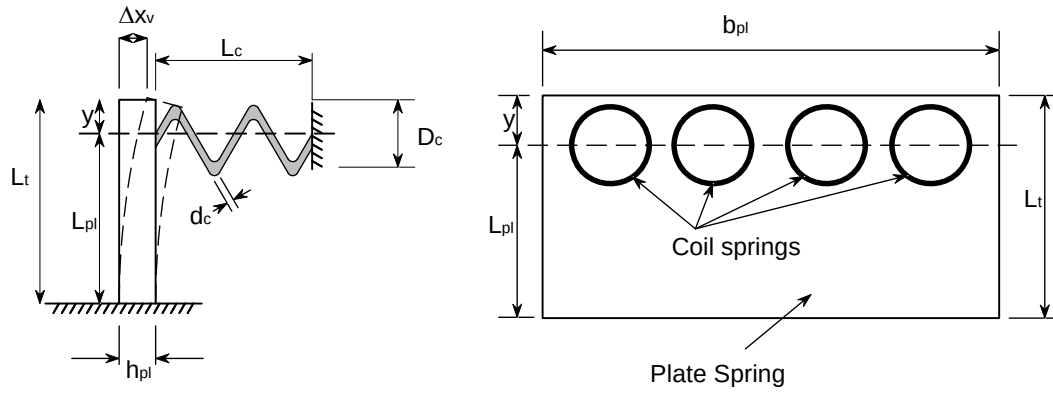


Fig.A12 Schematic Spring Configuration

A9.3.2 INPUT DATA

Force to cut sphere [1]:	$F_{\text{cut}} := 18\text{kN}$	
Maximum force on sphere: (From algorithm)	$F_{\text{st}} := \max(F_{\text{st}})$	$F_{\text{st}} = 6.593 \cdot \text{kN}$
Maximum spring deflection: (From algorithm)	$\Delta x_{\text{stv}} := \max(x)$	$\Delta x_{\text{stv}} = 2.64 \cdot \text{mm}$
Modulus of elasticity:	$E_{\text{xx}} := 210\text{GPa}$	
Spring length (maximum available):	$L_t := 50\text{mm}$	
Spring width (maximum available):	$b_{\text{pl}} := 90\text{mm}$	
Modulus of elasticity for spring wire:	$E_{\text{mw}} := 209\text{GPa}$	
Poisson's ratio:	$\nu := 0.3125$	
Number of cycles:	$n_L := 10^6$	
Preloads of parallel springs:	$F_{\text{min}} := 10\text{N}$	

A9.3.3 DETAIL CALCULATIONS

A9.3.3.1 THE SPRING PLATE ALONE

Required plate spring moment of inertia:	$I_{\text{rq}} := \frac{F_{\text{st}} \cdot L_t^3}{3 \cdot E_{\text{xx}} \cdot \Delta x_{\text{stv}}}$	(39)
--	--	------

Required plate spring thickness:	$h_{\text{rq}} := \sqrt[3]{\frac{I_{\text{rq}} \cdot 12}{b_{\text{pl}}}}$	$h_{\text{rq}} = 4.044 \cdot \text{mm}$	(40)
----------------------------------	---	---	------

Section modulus:

$$S_{rq} := \frac{b_{pl} \cdot h_{rq}^2}{6} \quad (41)$$

Bending moment:

$$M_{mom} := F_{st} \cdot L_t \quad (42)$$

Maximum bending stress:

$$\sigma_{rq} := \frac{M_{mom}}{S_{rq}} \quad \sigma_{rq} = 1343.9 \cdot \text{MPa} \quad (43)$$

A9.3.3.2 CONCLUSION FOR SPRING PLATE ALONE

The required spring with dimensions given above will result in the k-value as set above the algorithm for the steel spring. The resulting bending stress will be too high for normal available steels, thus making the spring unreliable.

A9.3.3.3 THE SPRING PLATE ALONE, MODIFIED FOR LOWER BENDING STRESS

Modify spring thickness:

$$h_{mod} := 6 \text{ mm}$$

Plate spring moment of inertia:

$$I_{mod} := \frac{b_{pl} \cdot h_{mod}^3}{12} \quad (44)$$

k-value for plate spring [13:892]:

$$k_{mod} := \frac{3 \cdot E_{xx} \cdot I_{mod}}{L_t^3} \quad k_{mod} = 8164.8 \cdot \frac{\text{kN}}{\text{m}} \quad (45)$$

A9.3.3.4 CONCLUSION FOR MODIFIED SPRING PLATE ALONE

Making the spring thicker reduces the deflection of the spring. But less deflection results in a higher force due to the shorter impact time. The result will again be a spring with too high a bending stress. A plate spring alone will not be reliable.

A9.3.4 THE SPRING PLATE AND COIL SPRING COMBINATION

The k-value of the spring assembly must match the k-value given for the algorithm. The spring is a parallel assembly. For a parallel assembly the total k is:

$$k_{as} = k_{pl} + k_{ca} \quad (46)$$

The plate spring will be a weaker spring than required, backed by coil springs.

The question is what the k-values for each must be?

The following procedure will be used to characterize the spring geometry by available space and spring design limitations. This will be used to obtain the k-value for the coil springs.

Choose a approximate spring diameter of $D_1 := 12\text{mm}$. The spring index C is the coil diameter divided by the wire diameter and must not be less than $C_c := 6$ [4:360], for which

$$d_1 := \frac{D_1}{6} \text{ gives } d_1 = 2\cdot\text{mm} \quad (47)$$

With a thresh := 0.1, which means rounded to the nearest 0.1mm diameter, gives the next available wire diameter:

$$d_c := \left(\text{Round} \left(\frac{d_1}{\text{mm}}, \text{thresh} \right) \cdot \text{mm} \right) \quad (48)$$

$$\text{With the spring index the mean spring diameter } D_c := (d_c) \cdot C_c \quad (49)$$

$$\text{Coil wire diameter:} \quad d_c = 2\cdot\text{mm}$$

$$\text{Spring mean diameter:} \quad D_c = 12\cdot\text{mm}$$

$$\text{Spring outer diameter:} \quad D_{co} := D_c + d_c \quad (50)$$

$$\begin{array}{ll} \text{Number of springs:} & nr := \frac{b_{pl}}{D_c} \\ \text{Space available on plate spring} & \end{array} \quad (51)$$

$$\text{Number of coil springs:} \quad nr_c := \text{floor}(nr) \quad nr_c = 7$$

$$G_c := \frac{E_{mw}}{2 \cdot (1 + \nu)} \quad (52)$$

$$\text{Number of turns (assume):} \quad n_{ct} := 3$$

$$\begin{array}{lll} \text{k-value for one coil spring} & k_{cs} := \frac{d_c^4 \cdot G_c}{8 \cdot D_c^3 \cdot n_{ct}} & k_{cs} = 30.717 \cdot \frac{\text{kN}}{\text{m}} \\ \text{[14:363]:} & & \end{array} \quad (53)$$

$$\begin{array}{lll} \text{k-value for parallel coil spring} & k_{ca} := k_{cs} \cdot nr_c & k_{ca} = 215.021 \cdot \frac{\text{kN}}{\text{m}} \\ \text{assembly:} & & \end{array} \quad (54)$$

The coil spring is positioned $\frac{D_c}{2}$ below the plate spring's top length. But by assuming that the Δx for the coil spring is the same as for the plate spring the force of the spring can be calculated.

$$\text{Force on one coil spring:} \quad F_c := k_{cs} \cdot \Delta x_{stv} \quad F_c = 81.013 \text{ N} \quad (55)$$

A9.3.4.1 Fatigue calculations for a coil spring

$$\text{Force amplitude on 1 Spring:} \quad F_a := \frac{F_c - F_{\min}}{2} \quad F_a = 35.5 \cdot \text{N} \quad (56)$$

$$\text{Mean force on 1 Spring:} \quad F_m := \frac{F_c + F_{\min}}{2} \quad F_m = 45.5 \cdot \text{N} \quad (57)$$

$$\text{Shear-stress multiplication factor [4:360]:} \quad K_s := 1 + \frac{0.5}{C_c} \quad (58)$$

$$\text{Stress amplitude on 1 Spring:} \quad \tau_a := K_s \cdot \frac{8 \cdot F_a \cdot D_c}{\pi \cdot d_c^3} \quad \tau_a = 146.928 \cdot \text{MPa} \quad (59)$$

$$\text{Mean stress on 1 Spring:} \quad \tau_m := K_s \cdot \frac{8 \cdot F_m \cdot D_c}{\pi \cdot d_c^3} \quad \tau_m = 188.308 \cdot \text{MPa} \quad (60)$$

$$\tau_{\max} := \tau_a + \tau_m \quad \tau_{\max} = 335.24 \cdot \text{MPa} \quad (61)$$

For the material the choice will be Chrome silicon (UNS G92540, AISI 9254). This material can be used up to 250°C. [4:367]. From [4:368] Table 10-2 the properties for the material are available.

$$\text{Constant A from table 10.2 [4:368]:} \quad A_{\text{mw}} := 2000 \text{MPa}$$

$$\text{Exponent m from table 10.2 [4:368]:} \quad m_{\text{mw}} := 0.112$$

$$\text{Ultimate strength in tension [4:368]:} \quad S_{\text{ut}} := \frac{A_{\text{mw}}}{\left(\frac{d_c}{\text{mm}}\right)^{m_{\text{mw}}}} \quad S_{\text{ut}} = 1850.6 \cdot \text{MPa} \quad (62)$$

$$\text{Yield strength [14:368]:} \quad S_y := 0.75 \cdot S_{\text{ut}} \quad (63)$$

$$\text{Torsional yield strength [14:368]:} \quad S_{\text{sy}} := 0.577 \cdot S_y \quad S_{\text{sy}} = 800.851 \cdot \text{MPa} \quad (64)$$

$$\text{Maximum endurance limit in shear [4:270, 375]:} \quad S'_{\text{se}} := 465 \text{MPa}$$

$$\text{Reliability 0.90 [4:251]:} \quad k_c := 0.897$$

$$\text{Wahl correction factor [4:360]:} \quad k_{c1} := \frac{4 \cdot C_c - 1}{4 \cdot C_c - 4} + \frac{0.615}{C_c} \quad (65)$$

$$\text{Effect of curvature [4:360]:} \quad K_c := \frac{k_{c1}}{K_s} \quad (66)$$

Modifying factor for stress concentration [4:243]:

$$k_e := \frac{1}{K_C} \quad (67)$$

Bending endurance limit [4:270]:

$$S_{se} := k_C \cdot k_e \cdot S'_{se} \quad S_{se} = 360.769 \cdot \text{MPa} \quad (68)$$

The following calculations are to calculate the live-cycle of the spring

[4:241]:

$$b_h := \frac{-1}{3} \log \left(\frac{0.8 \cdot S_{ut}}{S'_{se}} \right) \quad b_h = -0.168 \quad (69)$$

Unit implementation:

$$S_{ut1} := S_{ut} \cdot \frac{1}{\text{Pa}} \quad S_{ut1} = 1.851 \times 10^9 \quad (70)$$

Unit implementation:

$$S'_{se1} := S'_{se} \cdot \frac{1}{\text{Pa}} \quad S'_{se1} = 4.65 \times 10^8 \quad (71)$$

[4:241]:

$$C_L := \log \left[\frac{(0.8 \cdot S_{ut1})^2}{S'_{se1}} \right] \quad C_L = 9.673 \quad (72)$$

Mean fatigue strength [4:241]:

$$S'_f := 10^{C_L \cdot b_h} \cdot \text{Pa} \quad S'_f = 465 \cdot \text{MPa} \quad (73)$$

Safety factor:

$$n := \begin{cases} \frac{S'_f}{\tau_a} & \text{if } \tau_a > 0 \\ 0 & \text{otherwise} \end{cases} \quad n = 3.165 \quad (74)$$

Spring failure when $\tau_a = S_{se}$: [4:271, 374]

$$\text{Failure}_{se} := \begin{cases} \text{"yes"} & \text{if } \tau_a \geq S_{se} \\ \text{"No"} & \text{otherwise} \end{cases} \quad \text{Failure}_{se} = \text{"No"} \quad (75)$$

or whenever $\tau_{\max} = \tau_a + \tau_m = S_{sy}$: [4:271, 374]

$$\text{Failure}_{sy} := \begin{cases} \text{"yes"} & \text{if } \tau_{\max} \geq S_{sy} \\ \text{"No"} & \text{otherwise} \end{cases} \quad \text{Failure}_{sy} = \text{"No"} \quad (76)$$

With a coil spring design that complies with the failure criteria, the design of the plate spring can be done.

Required k-value for parallel plate spring:

$$k_{pl} := \begin{cases} (k_t - k_{ca}) & \text{if } k_{ca} < k_t \\ 0 & \text{otherwise} \end{cases} \quad k_{pl} = 2284.979 \cdot \frac{\text{kN}}{\text{m}} \quad (77)$$

Force on plate springs:

$$F_{pl} := \begin{cases} k_{pl} \cdot \Delta x_{stv} & \text{if } k_{pl} > 0 \\ 0 & \text{otherwise} \end{cases} \quad F_{pl} = 6.026 \cdot \text{kN} \quad (78)$$

Plate spring moment of inertia:

$$I_{pl} := \frac{F_{pl} \cdot L_t^3}{3 \cdot E_{xx} \cdot \Delta x_{stv}} \quad (79)$$

$$\text{Required plate spring thickness} \quad h_{pl} := \sqrt[3]{\frac{I_{pl} \cdot 12}{b_{pl}}} \quad h_{pl} = 3.925 \cdot \text{mm} \quad (80)$$

$$\text{Section modulus:} \quad S_{pl} := \frac{b_{pl} \cdot h_{pl}^2}{6} \quad (81)$$

$$\text{Bending moment:} \quad M_{mom} := F_{pl} \cdot L_t \quad (82)$$

$$\text{Maximum bending stress:} \quad \sigma_{pl} := \frac{M_{mom}}{S_{pl}} \quad \sigma_{pl} = 1304.2 \cdot \text{MPa} \quad (83)$$

A9.3.5 CONCLUSION

The coil spring cycle failure was predicted with conditions: $\tau_a = S_{se}$ and $\tau_{max} = \tau_a + \tau_m = S_{sy}$. With these conditions valid the plate spring was designed. The sphere has some properties to absorb some of the momentum change, but these were not taken into account. The force acting on the springs are still very high resulting in possible failure. Another concept could be to replace the coil springs with a hard rubber.

A10 THE USE OF SILICONE AS ALTERNATIVE FOR SPRING MATERIAL

This is a concept to replace the metal spring assembly with a silicone insert acting as a spring. Advantages include manufacturing price and less sphere damage.

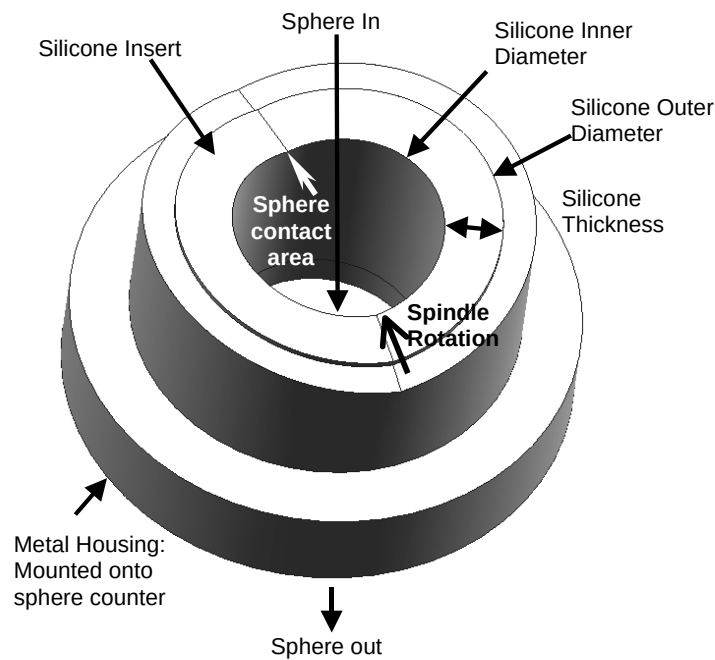


Fig. A13 Silicone Insert and Housing concept

A10.1 METHODOLOGY

The silicone properties given in the table below will be used in the calculations to see if it is feasible to use silicone and to use silicone and test the hardness of silicone.

The silicone HT-1500 has a Durometer Shore "A" of 75, and then 70 for HT-1270, 60 for HT-1260 and 50 for HT-1451.

The properties of silicone are given as % indentation at a certain pressure applied by the object, in this case the sphere. The figure 14 and 16 gives a graphical view of the indentation process.

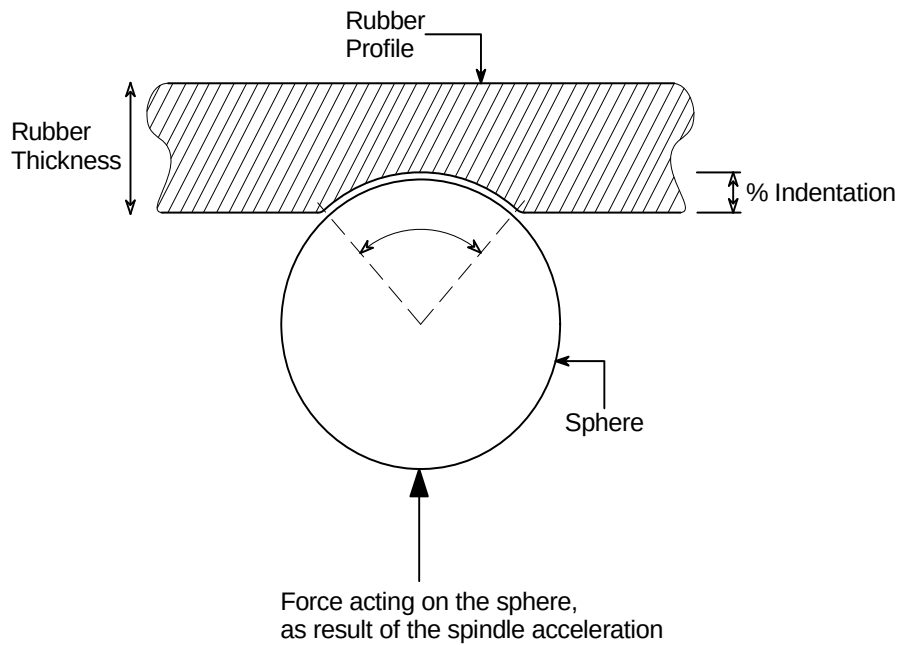


Fig. A14 Sphere Circular Indentation Pattern

A10.2 INPUT DATA

The table below contains random silicone properties from reference 12. The purpose is to get an idea of what is required for silicone properties in terms of hardness. The HT-1500 has the highest hardness.

SiliconeData :=

Required-unknown		Required-unknown		HT-1500		HT-1270	
Indent	Delta P	Indent	Delta P	Indent	Delta P	Indent	Delta P
%	[Pa]	%	[Pa]	%	[Pa]	%	[Pa]
0	0	0	0	0	0	0	0
0.084	344737.9	0.175	344739.9	2.1	344737.9	3.9	344737.9
0.196	689475.8	0.408333	689479.8	4.9	689475.8	7.5	689475.8
0.292	1034214	0.608333	1034220	7.3	1034214	10.7	1034214
0.372	1378952	0.775	1378960	9.3	1378952	14.1	1378952
0.432	1723690	0.9	1723700	10.8	1723690	17.05	1723690
0.488	2068427	1.016667	2068439	12.2	2068427	20.05	2068427
0.55	2413165	1.145833	2413179	13.75	2413165	22.8	2413165
0.6	2757903	1.25	2757919	15	2757903	25	2757903
0.644	3102641	1.341667	3102659	16.1	3102641	27.3	3102641
0.692	3447379	1.441667	3447399	17.3	3447379	29.3	3447379

Silicone selection between the four types, as given in the above table, can now be done. After selecting a type of silicone, the calculations will follow according to the selected silicone.

SiliconeType :=

Silicone type selection:

- Required 1
- Required 2
- HT-1500
- HT-1270
- HT-1260

Convert table into smooth function for further analysis.

Compression % collection:

$$\%compress := \left| \begin{array}{l} col \leftarrow \text{round}(\text{SiliconeType} \cdot 2 - 2) \\ \text{SiliconeData}^{(col)} \end{array} \right. \quad (84)$$

Pressure collection:

$$\text{Pressure} := \begin{cases} \text{col} \leftarrow \text{round}(\text{SiliconeType} \cdot 2 - 1) \\ \text{SiliconeData}^{\langle \text{col} \rangle} \end{cases} \quad (85)$$

Spline form:

$$\text{vs} := \text{lspline}(\% \text{compress}, \text{Pressure}) \quad (86)$$

Pressure interpolation:

$$P_{\text{rubber}}(\%_c) := \text{linterp}(\% \text{compress}, \text{Pressure}, \%_c) \cdot \text{Pa} \quad (87)$$

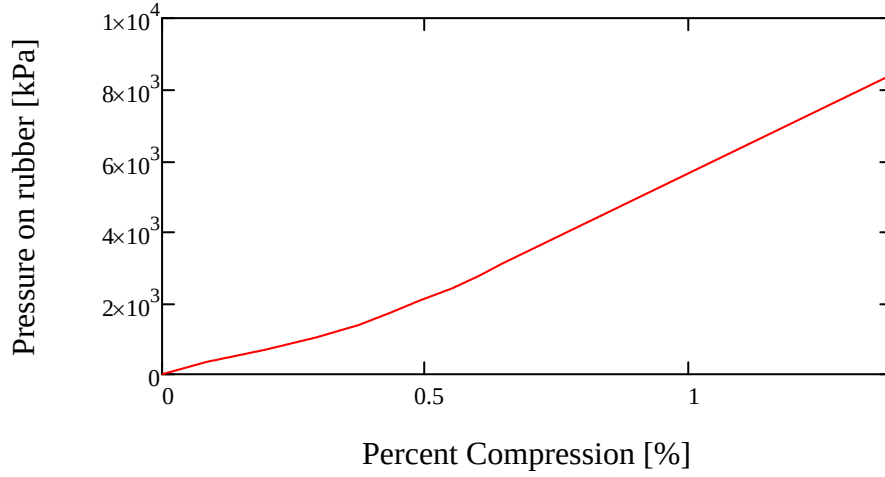


Fig. A15 Rubber Hardness

Torque limiter max torque

$$T_{\text{tls}} := 300 \text{ N} \cdot \text{m}$$

Torque limiter force on sphere:

$$F_{\text{tls}} := \frac{T_{\text{tls}}}{D_{\text{sp}} \cdot 0.5} \quad F_{\text{tls}} = 937.5 \text{ N} \quad (88)$$

Sphere diameter:

$$d_{\text{sf}} := 60 \text{ mm}$$

Sphere radius:

$$r_{\text{sf}} := \frac{d_{\text{sf}}}{2} \quad (89)$$

Rubber thickness:

$$t_r := 20 \text{ mm}$$

Rubber inner radius:

$$r_{\text{ri}} := \frac{64.8}{2} \text{ mm} \quad (90)$$

Rubber outer diameter:

$$d_{\text{ro}} := (r_{\text{ri}} + t_r) \cdot 2 \quad d_{\text{ro}} = 104.8 \cdot \text{mm} \quad (91)$$

A10.3 DETAIL CALCULATIONS

The area of contact increases as the sphere indents the silicone.

The Figure below was used to define the area of contact of the sphere acting as the pressure area on the rubber.

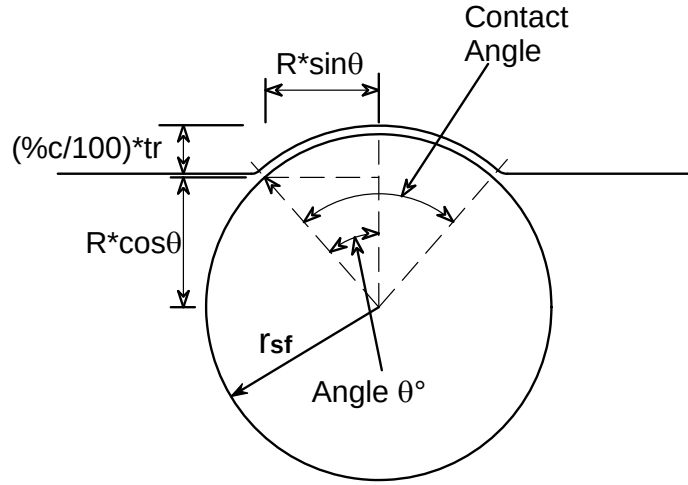


Fig. A16 Diagram Showing Sphere Contact Area

The area of contact between the sphere and the silicone is found by calculating

$$A = \frac{\pi \cdot d^2}{4} \text{ where } d = 2 \cdot R \cdot \sin \theta \text{ as in figure 19.} \quad (92)$$

The contact area is a function of the indentation into the silicone. The distance indent is calculated with

$$\Delta x_{id} = r_s - \left(\frac{C}{100} \cdot t_r \right). \quad (93)$$

$$q \text{ is calculated with } \cos(\theta) = \frac{r_s - \Delta x_{id}}{r_s}. \quad (94)$$

$$\text{Now the new diameter of contact with } d = 2 \cdot r_s \cdot \sin(\theta). \quad (95)$$

Pressure on rubber:

$$P_r := \text{Pressure} \cdot \text{Pa}$$

Area of contact:

$$A_{cs1}(C) := \frac{\pi \cdot \left[r_s \cdot \sin \left[\arccos \left(\frac{r_s - \left(\frac{C}{100} \cdot t_r \right)}{r_s} \right) \right] \right]^2}{4} \quad (96)$$

Force (acting):

$$F_r(C) := \overrightarrow{(P_{\text{rubber}}(C) \cdot A_{cs1}(C))} \quad (97)$$

Indentation:

$$\%C(\Delta x) := \frac{\Delta x}{t_r} \cdot 100 \quad (98)$$

k-value for rubber:

$$k_r(\Delta x) := \frac{F_r(\%C(\Delta x))}{\Delta x} \quad (99)$$

$$\Delta x_r := \frac{\%compress}{100} \cdot t_r$$

$$\Delta x_r = \text{m} \tag{100}$$

	0
0	0
1	1.68·10 ⁻⁵
2	3.92·10 ⁻⁵
3	5.84·10 ⁻⁵
4	7.44·10 ⁻⁵
5	8.64·10 ⁻⁵
6	9.76·10 ⁻⁵
7	1.1·10 ⁻⁴
8	1.2·10 ⁻⁴
9	1.288·10 ⁻⁴
10	1.384·10 ⁻⁴

Take %.c equal to %compress:

$$\%_c := \%compress$$

$$\tag{101}$$

	0
0	0
1	2.911
2	13.585
3	30.357
4	51.563
5	74.849
6	101.46
7	133.406
8	166.322
9	200.831
10	239.773

Then the force relative to the % compression of the silicone will be:

$$F_r(\%_c) = \text{N}$$

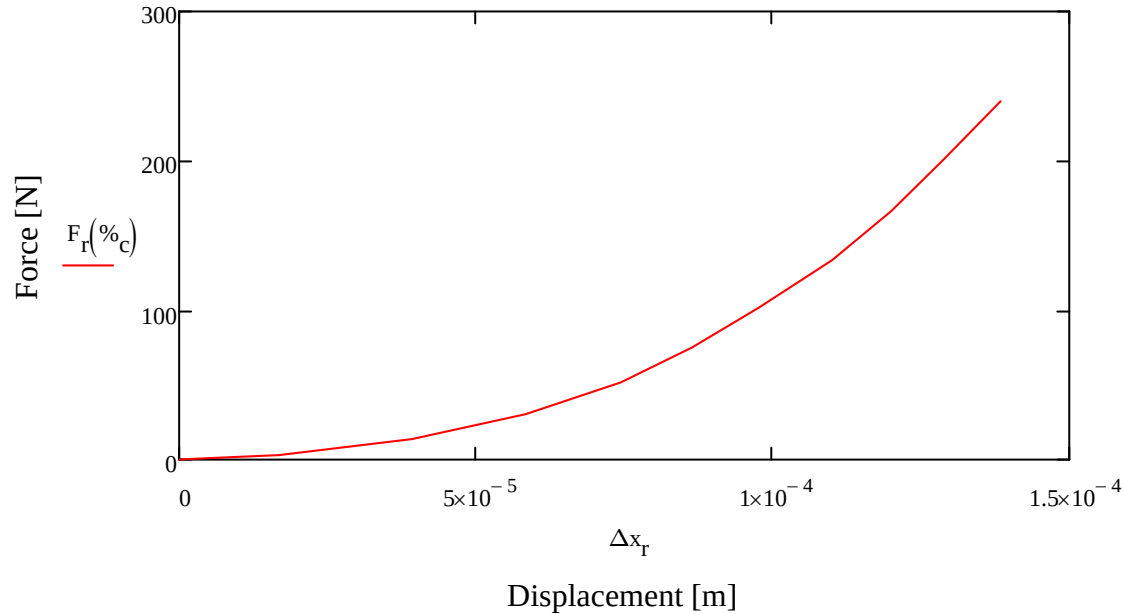


Fig. A17 Force/displacement graph

A10.3.1 ALGORITHM FOR SILICON INSERT

The algorithm below is the same as that of the metal spring, except that the k-value is now replaced by the silicone k-value which is a function of Dx.

$$\begin{array}{l}
 \left(\begin{array}{c} x_{sr} \\ \omega \\ \alpha \\ F_{srv} \\ Fact \end{array} \right) := \left| \begin{array}{l}
 \omega_0 \leftarrow \omega_{sp} \cdot s \\
 x_{sr0} \leftarrow 0 \\
 F_{srv0} \leftarrow 0 \\
 F_{tl0} \leftarrow 0 \\
 \alpha_0 \leftarrow 0 \\
 i \leftarrow 0 \\
 \text{while } i \cdot \Delta t_{\text{step}} < 0.17 \\
 \quad i \leftarrow i + 1 \\
 \quad \omega_i \leftarrow \omega_{i-1} + \alpha_{i-1} \cdot \Delta t_{\text{step}} \\
 \quad \omega_i \leftarrow 0 \text{ if } |\omega_i| < 10^{-4} \\
 \quad x_{sr_i} \leftarrow x_{sr_{i-1}} + \omega_i \cdot \frac{D_{sp} \cdot 0.5}{m} \cdot \Delta t_{\text{step}} \\
 \quad F_{srv_i} \leftarrow x_{sr_i} \cdot k_r(x_{sr_i} \cdot m) \cdot \left(\frac{m}{N}\right) \cdot (x_{sr_i} \geq 0) \\
 \quad F_{fric} \leftarrow \left| \begin{array}{l} \frac{F_e}{N} \text{ if } \omega_i > 0 \\ -\frac{F_e}{N} \text{ otherwise} \end{array} \right. \\
 \quad Fact_i \leftarrow \left| \begin{array}{l} F_{srv_i} + F_{fric} \text{ if } \omega_i = \omega_{sp} \cdot s \\ \frac{F_{tls}}{N} \text{ otherwise} \end{array} \right. \\
 \quad Fact_i \leftarrow \frac{F_{tls}}{N} \text{ if } Fact_i > \frac{F_{tls}}{N} \\
 \quad \alpha_i \leftarrow (Fact_i - F_{srv_i} - F_{fric}) \cdot \frac{D_{sp} \cdot 0.5}{J_t} \cdot (m \cdot kg) \\
 \left(\begin{array}{c} x_{sr} \\ \omega \\ \alpha \\ F_{srv} \\ Fact \end{array} \right)
 \end{array} \right.
 \end{array}$$

i-steps for time:

$i := 0 \dots \text{last}(x)$

time as per i-steps:

$$t_i := i \cdot \Delta t_{\text{step}} \cdot s \quad (102)$$

Unit implementation:

$$\begin{pmatrix} x_{\text{sr}} \\ \omega \\ \alpha \\ \text{Fsrv} \\ \text{Fact} \end{pmatrix} := \begin{pmatrix} x_{\text{sr}} \cdot \text{m} \\ \omega \cdot \frac{1}{\text{s}} \\ \alpha \cdot \frac{1}{\text{s}^2} \\ \text{Fsrv} \cdot \text{N} \\ \text{Fact} \cdot \text{N} \end{pmatrix} \quad (103)$$

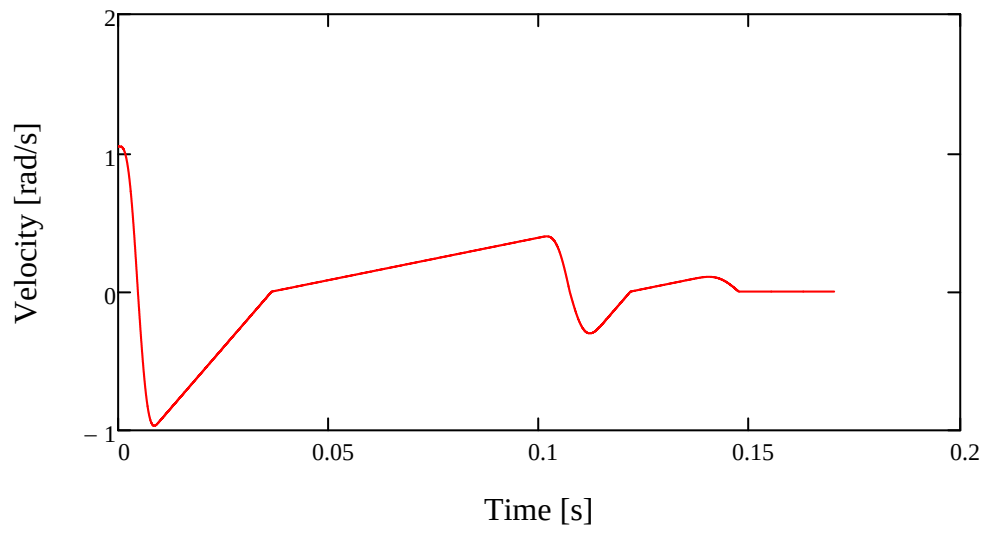


Fig. A18 Spindle Velocity

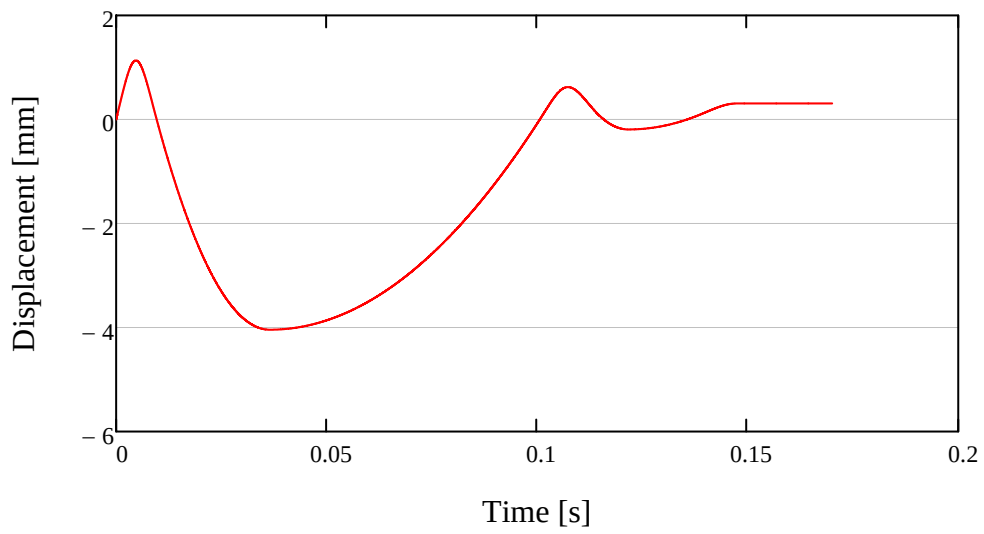


Fig. A19 Deflection of Silicon at impact

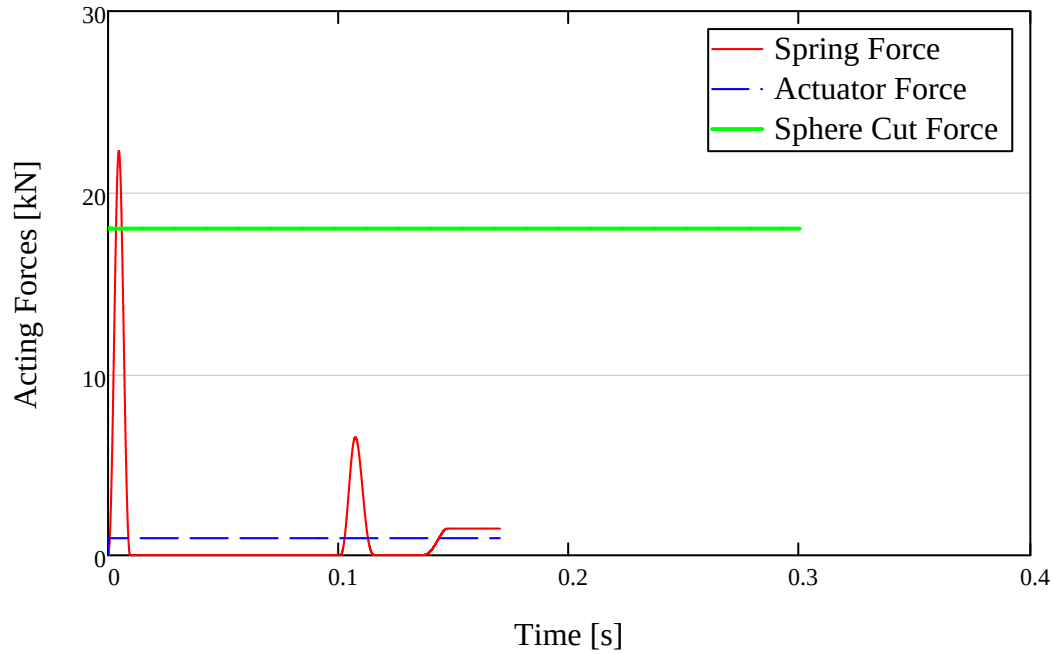


Fig. A20 Acting Forces

Spring constant at maximum force:

$$k_s := \frac{\max(F_{sr})}{\max(x_{sr})} \quad k_s = 19660.85 \cdot \frac{\text{kN}}{\text{m}} \quad (104)$$

A10.4 CONCLUSION:

The calculations for the silicone spring options started with the table containing four random silicone properties. During the calculations it was found that all four selections was too soft. A fifth column was added after running the algorithm and by changing the data in the column iteratively properties of what should be required was derived. This iteration was also done while changing the torque limiter setting. The torque limiter and spring constant in combination give the amount of head return. The idea is currently to let the head travel back to before the impact point to release the sphere.

Silicone does not have a straight line for a spring constant, but the spring constant given above at maximum impact force relates to the spring constant required from the metal spring.

In this algorithm the sphere impacted on a straight silicone strip with a straight forward impact, but the right insert will be found that will result in a greater impact area and therefore less indentation into the silicone.

The result is that silicone can be used for a spring. It can fit in the same position in the housing as the metal spring.

A11 CALCULATE SHAFT PROPERTIES FOR TORQUE TRANSMITTING

A11.1 METHODOLOGY OF SHAFT CALCULATION

The Penetration shaft and the head shaft are the two shafts to be used. The material properties for the shafts to be able to cut spheres will be calculated by selecting the weakest part of each shaft. The weakest part of the penetration shaft is the 30mm diameter part and in the penetration shaft the 27mm square at the end. Figure 4 shows a picture of both shafts indicating the critical points.

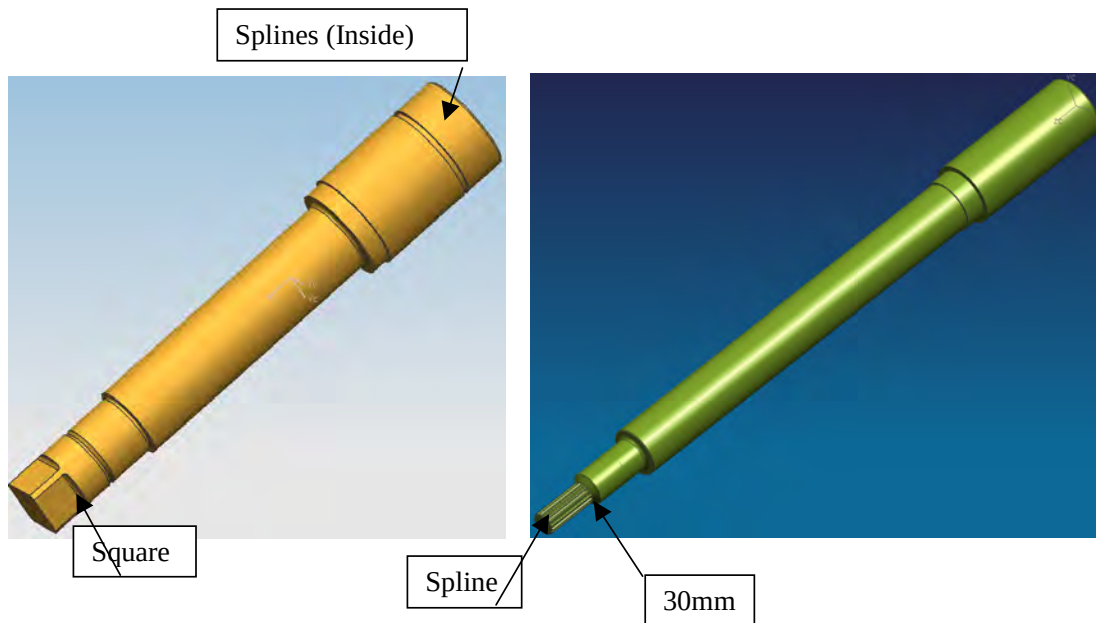


Fig.A21 Penetration shaft (Left) and head shaft (Right)

A11.2 INPUT DATA

EN 24 (Condition V)
steel yield stress:

$$\sigma_{24} := 835 \text{ MPa} \quad [5]$$

If the shafts can be replaced we do not need a safety factor as high as 3 for fatigue, thus in this case a safety factor of 1.2 was used.

Safety factor:

$$S_f := 1.2$$

Effective radius of square:

$$r_{sq} := \frac{35 \text{ mm}}{2} \quad (105)$$

The actual effective square radius will be a bit more because the square is a bit larger than the inscribed square.

Square section of penetration shaft:

$$b_{sq} := 27 \text{ mm}$$

Shaft diameter (smallest dia):

$$D_{sp1} := 30 \text{ mm}$$

A11.3 DETAIL CALCULATIONS

A11.3.1 MATERIAL REQUIRED FOR PENETRATION SHAFT TO BE ABLE TO CUT A SPHERE

The polar moment of inertia for a square : $I_p = \frac{b \cdot h}{12} \cdot (h^2 + b^2)$ (106)

[3:877]

Square moment of inertia: $I_{sq} := \frac{b_{sq} \cdot b_{sq}}{12} \cdot (b_{sq}^2 + b_{sq}^2)$ (107)

Required shear stress: $\tau_{sq} := \frac{T_{cut} \cdot r_{sq}}{I_{sq}}$ $\tau_{sq} = 1138 \cdot \text{MPa}$ (108)

A11.3.2 MATERIAL REQUIRED FOR HEAD SHAFT TO BE ABLE TO CUT A SPHERE

Required shear stress: $\tau_{sp1} := \frac{T_{cut} \cdot 16}{D_{sp1}^3 \cdot \pi}$ $\tau_{sp1} = 1086 \cdot \text{MPa}$ (109)

A11.4 CONCLUSION:

The shear stress required to be able to cut spheres for both shafts is quite high. Thus it would not be possible to cut spheres with these shafts. A torque limiter would be required to protect these shafts.

A12 SUMMARY OF RESULTS

A12.1 HEAD IMPACT

Minimum impact time: $\Delta t_{im} = 0.003 \text{ s}$

Minimum impact distance: $\Delta x_{im} = 0.886 \cdot \text{mm}$

These two values show the need for a time delay and a spring will therefore be used.

A12.2 REQUIRED SHAFT MATERIAL PROPERTIES FOR SPHERE CUTTING

Penetration shaft shear stress: $\tau_{sq} = 1138 \cdot \text{MPa}$

Head shaft shear stress: $\tau_{sp1} = 1086.5 \cdot \text{MPa}$

Protect shafts against damage.

A12.3 METAL SPRING

Spring constant for metal spring: $k_t = 2500 \cdot \frac{\text{kN}}{\text{m}}$

Maximum impact force on spring: $F_{st} = 6.593 \cdot \text{kN}$

Maximum spring travel after impact: $\Delta x_{stv} = 2.637 \cdot \text{mm}$

Torque limiter setting: $T_{tl} = 300 \cdot \text{N} \cdot \text{m}$

The metal spring assembly fails because not enough space is available to place a reliable spring.

A12.4 SILICONE SPRING

Maximum impact force on spring: $\max(F_{srv}) = 22.287 \cdot \text{kN}$

Maximum spring travel after impact: $\max(x_{sr}) = 1.134 \cdot \text{mm}$

Spring constant at maximum force: $k_s = 19660.85 \cdot \frac{\text{kN}}{\text{m}}$

Torque limiter setting: $T_{tls} = 300 \cdot \text{N} \cdot \text{m}$

A13 CONCLUSION

Protection for the shafts and the sphere will be required. The metal spring did not pass the endurance test, because there is not enough space to install the required amount of springs to improve the spring life expectancy. The calculations showed that a silicone spring could also be used.

A table with required silicone properties was drawn up that gives a negative distance return of the head, which means that the head releases the sphere. Therefore the silicone spring could be feasible.

A14 RECOMMENDATIONS

Source silicone with properties higher than the "HT-1500". The two required columns is an indication of required properties.

A15 BOLT CACLCULATIONS

Number of M8 bolts:	$nr_{m8} := 8$	
Diameter of M8 bout:	$D_{m8} := 8.2\text{mm}$	
Maximum motor torque:	$T_{\max} := 1250\text{N}\cdot\text{m}$	
Diameter of motor bolts:	$D_m := 208\text{mm}$	
Outer diameter of housing:	$D_{ho} := 300\text{mm}$	
Thickness housing wall:	$t_w := 10\text{mm}$	
Diameter of housing bolts:	$D_h := 290\text{mm}$	
Diameter of M6 bout:	$D_{m6} := 5\text{mm}$	
Area of one M8 bolt:	$A_{m8} := \frac{\pi D_{m8}^2}{4}$	(110)
Diameter of torque limiter:	$D_{tl} := 260\text{mm}$	
Diameter of torque limiter rod:	$D_{tlr} := 12\text{mm}$	

A15.1 BOLT CACLCULATIONS FOR SHEAR (TORSION)

Force on all 8 motor bolts:	$F_{mb} := \frac{T_{\max}}{\left(\frac{D_m}{2}\right)}$	$F_{mb} = 12.019\cdot\text{kN}$	(111)
Shear stress per motor bolt:	$\tau_b := \frac{F_{mb}}{nr_{m8} \cdot A_{m8}}$	$\tau_b = 28.449\cdot\text{MPa}$	(112)
Force on all housing bolts:	$F_{hb} := \frac{F_{mb} \cdot D_m}{D_h}$	$F_{hb} = 8.621\cdot\text{kN}$	(113)
Area of all housing bolts:	$A_{hbt} := \frac{F_{hb}}{\tau_b}$	$A_{hbt} = 303.021\cdot\text{mm}^2$	(114)
Area of one M6 bolt:	$A_{m6} := \frac{\pi D_{m6}^2}{4}$	$A_{m6} = 19.635\cdot\text{mm}^2$	(115)
Number of M6 bolts:	$nr_{m6} := \frac{A_{hbt}}{A_{m6}}$	$nr_{m6} = 15.433$	(116)

15.2 BOLT CALCULATIONS FOR TENSION (MOTOR WEIGHT)

Motor mass: $M_m := 14.5\text{kg}$

Motor mass: $M_w := M_m \cdot g$ $M_w = 142.196\text{ N}$ (117)

A15.3 TORQUE LIMITER STOP

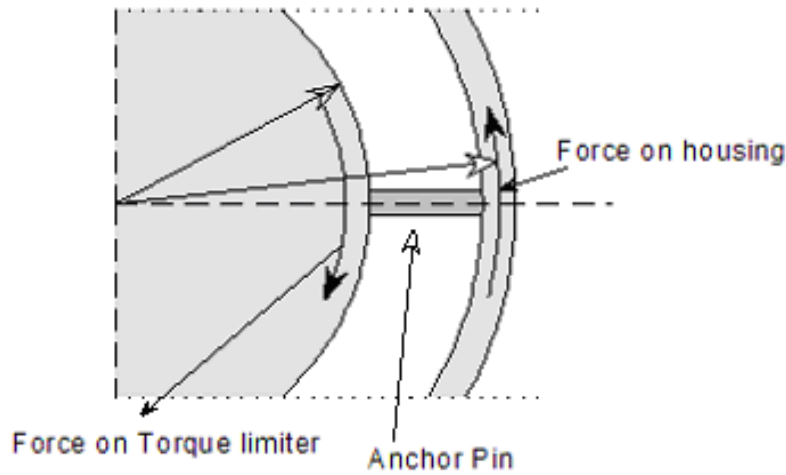


Fig.A22 Fixing to prevent Torque Limiter rotation

Force on anchor pin (Torque limiter): $F_{atl} := \frac{T_{\max}}{\frac{D_{tl}}{2}}$ $F_{atl} = 9615.4\text{ N}$ (118)

Force on anchor pin (Housing): $F_{ah} := \frac{T_{\max}}{\frac{D_h}{2}}$ $F_{ah} = 8620.7\text{ N}$ (119)

Resulting force on pin: $F_{ra} := F_{atl} - F_{ah}$ $F_{ra} = 994.695\text{ N}$ (120)

A15.4 TORQUE ON COVER SHELL

Inner diameter of housing: $D_i := \sqrt{D_{ho}^2 - \frac{A_{hbt} \cdot 4}{\pi}}$ $D_i = 299.356\text{ mm}$ (121)

Housing thickness: $t_h := \frac{(D_{ho} - D_i)}{2}$ $t_h = 0.322\text{ mm}$ (122)

Safety margin for max torque: $n_{sm} := \left(\frac{t_w - t_h}{t_w} \right) \cdot 100$ $n_{sm} = 96.8$ (123)

