DRY GAS SEAL DYNAMIC TEST BENCH

An Undergraduate Research Scholars Thesis

by

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We, Fatin Abbasher[1], Seham Al-Baker[2], Mohamed Fattouh[3], and Ryan Mansour[4], certify that all research compliance requirements related to this Undergraduate Research Scholars thesis have been addressed with our Research Faculty Advisor prior to the collection of any data used in this final thesis submission.

This project did not require approval from the Texas A&M University Research Compliance & Biosafety office.

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ABSTRACT

Dry Gas Seal Dynamic Test Bench

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Centrifugal compressors are among the most common yet critical equipment to maintain acceptable production levels in various industrial processes. Centrifugal compressors take in large volumes of gases, some of which are toxic and extremely dangerous. To contain these gases and avoid any leakages from taking place, dry gas seals are utilized. Dry gas seals need to be maintained and tested regularly, so designing a reliable testing system for the dry gas seal is very important to ensure a safe working environment in a plant. This project focuses on designing a test bench that conforms to API 692 standards such that it conducts dry gas seal testing under specific conditions and measures gas leakage rates. The main requirements of the test bench are to rotate the shaft in the seal up to 6000 RPM, to supply gas with pressure up to 220 bars, to withstand a temperature of 0 upto 20°C and to be able to measure gas leakage rate up to 15 scfm to ensure safe and efficient plant operations.

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NOMENCLATURE

Seal Ring (primary or stationary face-Yellow) – the ring which can move axially. Usually, it is non-rotating and does not have grooves. [5]

Mating Ring (rotating ring- Red) – the ring which does not move axially. usually the rotating face, which generally has grooves. [5]

Primary Seal (first stage) – innermost seal, closest to process side, comprising of both a seal ring and mating ring. [5]

Primary Vent – seal cartridge, compressor porting and vent piping which allows leakage from the primary seal to flow into the vent system and taken to a safe location. [5]

Secondary Seal (second stage) – outermost seal, closest to atmospheric side, comprising of both a seal ring and a mating ring. [5]

Secondary Vent – seal cartridge, compressor porting and vent piping which allows leakage from the secondary seal to flow into the secondary seal vent system. [5]

Process Gas – the main gas stream being compressed in the centrifugal compressor, which is to be sealed from the environment. [5]

Seal Gas – gas supplied to the high-pressure side of a self-acting gas seal, which flows through the gas seal faces and into the compressor. [5]

Secondary Seal Gas – gas supplied to the area between the primary and secondary seals of a tandem self-acting dry gas seal having an interstage labyrinth, used to keep the seal gas from flowing through the secondary seal and to maintain a positive pressure differential across the secondary seal. [5]

Separation Seal (barrier or tertiary seal) – carbon ring seals or split labyrinth seals between the gas seal and bearing. [5]

Separation Gas – a supply of inert gas or air fed into the region between the outermost seal and the shaft bearing. Separates oil from the dry gas seal cartridge and gas from the bearing area. [5]

Leak Rate – the rate of process gas leaked into the atmosphere, measured in ‘scfm’ (Standard Cubic feet per min) or ‘slpm’ (Standard Liter per min).

Dry Gas – gas consisting entirely of methane gas without any liquid hydrocarbons like butane or ethane.

Dynamic Conditions – performing tests while the shaft is rotating.

Dry Clean Gas – dry gas devoid of any micro-particles like dust or liquid droplets.

PMT – Provide a means to.

1. INTRODUCTION

Natural Gas is a significant contributor to the national income in Qatar, one of the world's leading exporters of gas. Qatar's natural gas reserve makes up around 14% of the entire world's deposits [1]. Hence, major Qatari gas companies need to maintain their production levels, which implies that their equipment is always required to be well maintained.

Centrifugal compressors are among the critical equipment that are important to maintain acceptable production levels in process industry. Centrifugal compressors are major turbomachinery used in the oil and gas industry responsible for taking in large volumes of gas and pressurizing it using the kinetic energy generated by a rotating shaft [2]. These compressors take in large volumes of gases, some of which are toxic and extremely dangerous. To contain these gases and avoid any leakages from taking place, dry gas seals are utilized. Dry gas seals are mechanical seals that use static pressure and hydrodynamic forces to prevent leakages. Due to their low leakage rates, centrifugal compressors are generally outfitted with dry gas seals.

Dry gas seals are typically used in high pressures and temperatures. It is imperative to test new and refurbished seals during their life cycle and ensure their leakage rates are within acceptable ranges. Failure to do so could result in a faulty seal leaking large amounts of toxic gas to the surroundings. According to Figure 1.1, spending on dry gas seals is predicted to increase over the years, which consequently causes the demand for dry gas seal testing also to increase.

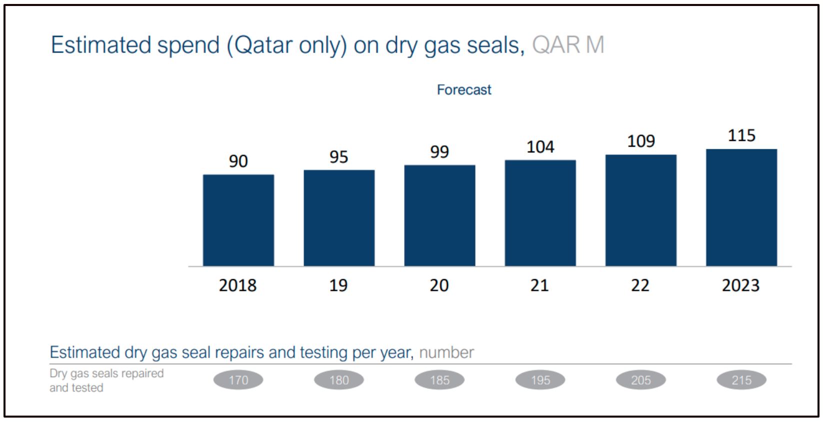

Figure 1.1: Qatar Petroleum Prediction [3]

In this report, a dry gas seal test bench design is presented that is able to examine dry gas seals at any point in their operational life while saving energy. To do this, the project was broken down into its core functions from which requirements were devised. From those functions and requirements, a design of the test bench of how a DGS seal could be tested was developed.

The goal of this research is to provide a means to test the leak rate of a pair of new "tandem with labyrinth" dry gas seals during static and dynamic operations, per API 692.

1.1 Background Theory

To understand the requirements for a test bench, the functionality of a dry gas seal and its respective standards, API 692, must be understood.

1.1.1 Dry Gas Seal

Figure 1.2 displays a centrifugal compressor with a dry gas seal at both ends of the housing. If the dry gas seals were not installed, there would be an opening between the interface of the shaft, housing, and the centrifugal compressor. This opening causes the process gas within the centrifugal compressor to leak out. Thus, a dry gas seal aims to provide effective sealing on turbomachinery to reduce the amount of process gas leaking into the atmosphere.

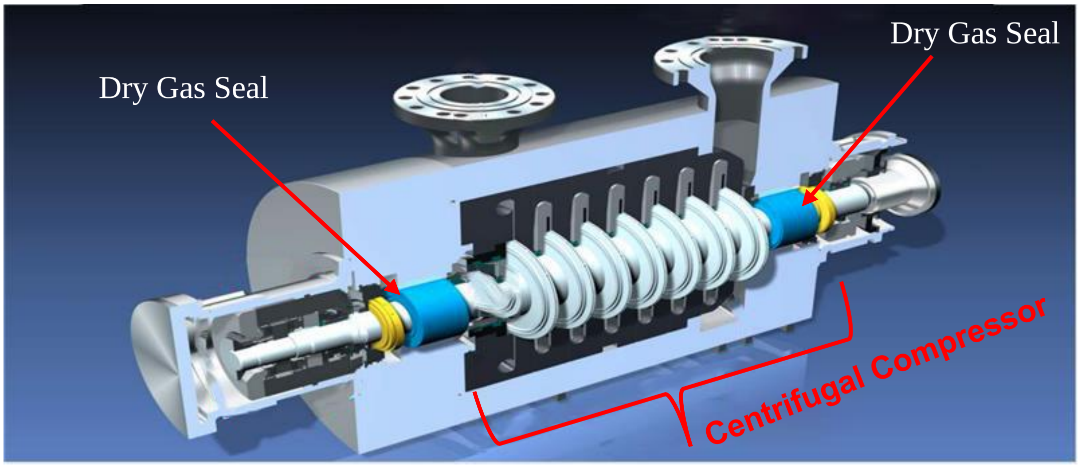

Figure 1.2: Schematic of a Centrifugal Compressor [8]

There are multiple configurations of dry gas seals but tandem with labyrinth dry gas seals are utilized by oil and gas industries since their operations deal with toxic gases. Figure 1.3 shows a cross-section of a tandem with labyrinth dry gas seal. In oil and gas applications, the primary seal gas supply is usually the same as the gas used in the process side, whilst the secondary seal gas supply is nitrogen or air. The primary vent is connected to a flare system whilst the secondary vent is connected to the atmosphere. The primary seal gas supply must be at a pressure higher than the process side so that the gas seals the process side. During dynamic conditions, a gap of 3-5 micron is formed between the two primary seals, therefore some of the primary seal gas supply (about 10%) leaks into the gap and flows out of the primary vent. To prevent the leaked primary seal gas from flowing into the secondary seals and into the atmosphere, the secondary seal gas is applied at a higher pressure than the leaked primary seal gas. Furthermore, the secondary seal acts as a back-up to the primary seal, if the primary seal fails then the secondary seal gas supply manages to push all the leaked process gas and primary gas supply to the primary vent where it is flared.

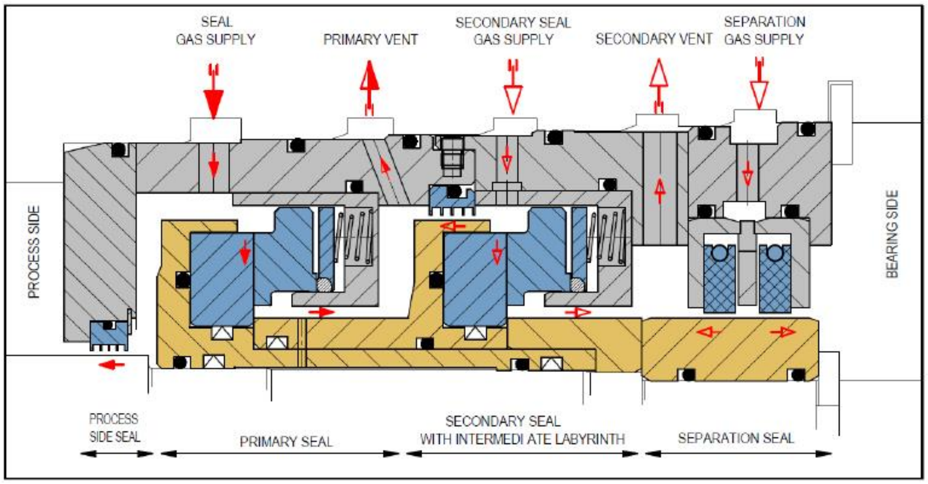

Figure 1.3: Tandem with Labyrinth Dry Gas Seal [5]

1.1.2 API 692

The modern standard used globally to qualify dry gas seals for industrial use is *API Standard 692: Dry Gas Sealing Systems for Axial, Centrifugal and Rotary Screw Compressors* [5]. It covers the minimum dry gas sealing system requirements for use in the petroleum, chemical, and gas industry services, and is increasingly being used to specify dry gas seals on new and upgrade projects. It is a very comprehensive document, and the result of a great effort from expert users, specifiers, and vendors, with the intent of increasing the reliability of dry gas seal systems. The document is divided into four parts:

Part 1: General requirements

Part 2: Dry gas seals

Part 3: Dry gas seal support systems

Part 4: Installation and commissioning

Prior to installation, dry gas seals are tested for validation. API 692 describes the testing procedures required to validate a dry gas seal for usage.

Primary Seal Gas Pressure	Secondary Seal Gas Pressure	Speed	Leakage Acceptance (scfm or slpm)		Test Step
			Primary	Secondary	
Maximum sealing pressure	0	0			5
≥Maximum sealing pressure	Maximum sealing pressure	0			13
Normal sealing pressure	Normal vent pressure	Normal speed			28
>Normal sealing pressure	Normal sealing pressure	Normal speed			55

Figure 1.4: Acceptance Criteria for Dry Gas Seals [5]

Figure 1.4 displays four test procedures required for validation as per the API 692 standards [5]. The first two tests are done under static conditions (non-rotating shaft) whilst the other two are conducted under dynamic operations (rotating shaft).

2. METHODS

2.1 Need Analysis

Dry gas seals are susceptible to leakages regardless of the arrangement and orientation of the seals. The gas leaking is often flammable methane or toxic hydrocarbons. Moreover, leaks result in loss of process gas from the centrifugal compressor leading to a decrease in overall production. It is imperative that the proposed solution can accurately measure the leakage from the seal being tested to validate the seal for usage. The leakage rate depends on multiple factors, such as:

1. The size of the seal
2. The rotational speed of the shaft
3. The operational pressure of the seals
4. The gap between the seals

Since dry gas seals come in different shapes and sizes, the test bench must be specific to the type of seal tested. Hence, the scope of the project is aimed at the Type 28 dry gas seal from John Crane to simplify it and keep it focused [4]. The Type 28 was specifically picked as it is the most used configuration in the market according to the main industry point of contact. The research aims to provide a blueprint on how dry gas seal test benches are operated and built.

Figure 2.1: Type 28AT from John Crane [4]

The test bench must test the typical operating conditions of Type 28 dry gas seals, satisfy the API 692 standards, and perform the API 692 acceptance criteria tests. The test bench needs to provide gas to two channels: the primary seal gas and the secondary seal gas. All the gas supplied to the dry gas seal should be dry clean gas. The primary seal gas should be operating at a pressure higher than the intake pressure of the centrifugal compressor used. While the secondary seal gas supply channel should be at a pressure slightly higher than the leaked primary seal gas supply to prevent toxic gas from leaking into the secondary seals and out to the atmosphere. See Figure 2.1 to gain a better understanding of how a dry gas seal Type 28 looks like.

A control system must also be implemented for the test bench to control the RPM of the shaft that is inserted into the dry gas seal and control the pressure and temperature of the primary and secondary seal gas supply. Furthermore, a monitoring system must also be implemented alongside the control system to monitor the pressure and temperature in all the channels, the vibration of the DGS, and, most importantly, the leakage rate of the primary and secondary vents during dynamic and static operations.

Moreover, prior to testing, the pair of dry gas seals need to be assembled and mounted on the shaft of the test bench. The assembly and mounting process must be easy to maximize the number of dry gas seals being tested.

Finally, the measurements acquired from the sensors require validation; therefore, a simulation of the entire system must be conducted to ensure that the values obtained from the sensors such as the leakage rate are within the simulated acceptable ranges.

2.2 Functions & Requirements of a Dry Gas Seal Test Bench

The function requirements must satisfy the general safety standards API 692 standards. Table 2.1 below shows the requirements of each function as well as the source from which they are taken.

Table 2.1: Functions & Requirements with the source of the requirements

Function	**Requirement**
Measure speed	Sensor is required to be able to measure 0 - 6000 RPM
Rotate to control rotational speed	Shaft must be able to rotate within a range of 2000- 6000 RPM
Transfer power to the shaft	A maximum of 50 kW of power is required to rotate the shaft at the given speeds
Control the primary seal gas	A flow rate of 9.8 m/s (32 ft/s) and a pressure range of 0-220 bar (g)
Control the secondary seal gas	A flow rate of 32 ft/s and a pressure range of 0-220 bar (g)
Control the process side gas channel	The process side channel must be 3 bars lower than the primary gas channel
Supply the seal gases	Source the gas at a flow rate of 32 ft/s and a pressure range of 0-220 bar (g)
Remove moisture from the seal gases	Ensure moisture content in air is less than 0.5 parts per million
Filter the clean gases	Must be filtered at a minimum of 3 microns.
Pressurize the dry, clean, seal gases	Ensure gas is supplied at the required pressure in the range of 0 – 220 bar (g)
Dispose the gas safely	Ensuring gas is disposed without harming operators in the surrounding area.
Measure the pressure & temperature at the supply channel of the primary and secondary seals	Ensure temperatures are below the 220°C and pressure within the 0-220 bar (g) range
Measure the pressure & temperature at the vents of the primary and secondary seals	Measure to ensure temperatures are below the 220°C and pressure within the 0-220 bar (g) range
Measure the gas leakage in primary and secondary channel	Ensure leakage is below the required limit as per the dry gas seal
Measure gas flowrate in the gas supply channels	Measure to ensure flowrate is at the correct rate of 9.8 m/s (32 ft/s)
Secure the DGS on the shaft	Ensure the seals are mounted on the shaft securely and the shaft diameter matches the inner diameter of the dry gas seal
Replace shaft to adjust for different DGS	Adjusting dimension of the shaft based on those of the seals, diameter of shaft ranges from 45 mm to 350 mm
Reduce friction on the shaft	Ensure smooth rotation to avoid wearing of the shaft and failures to heat generated by friction

3. RESULTS

3.1 Test Bench Design

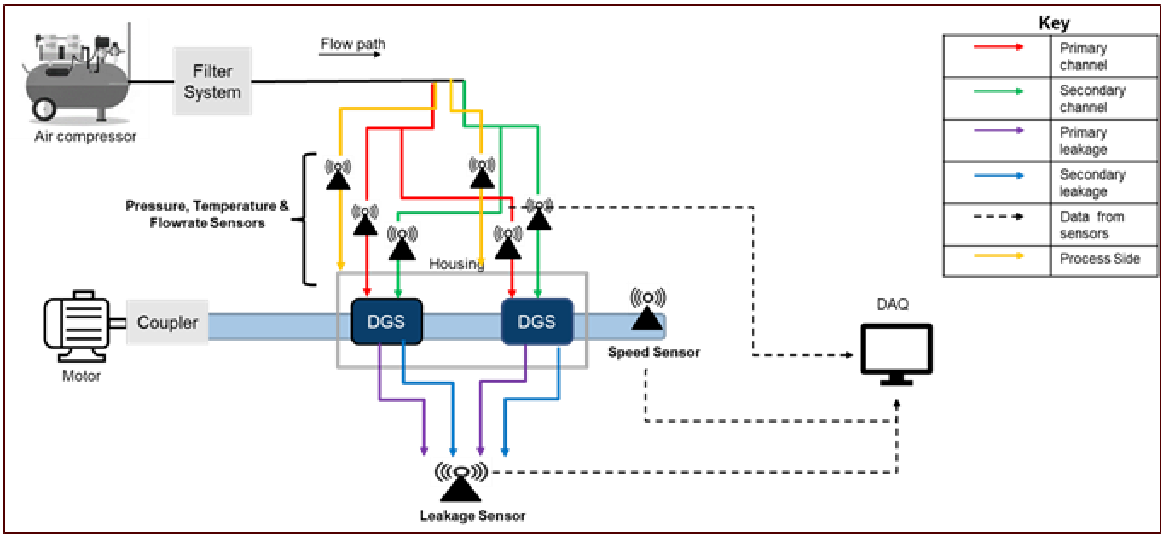

Figure 3.1: Test Bench Concept

For the concept showcased in Figure 3.1, the shaft will be rotated using an electric motor and the system is flushed with air to remove any dust. The setup is powered through electric sockets sourced from the power grid and is connected to an auxiliary power supply in case of power outages. The auxiliary power supply consists of a diesel generator. The shaft speed of the shaft will be monitored through the operation. The pressure, temperature, and leakage in all the gas channels will also be monitored during operation. All the sensors will be connected to a control system. Furthermore, a filter system for gas supplies must filter to 1 µm spherical in size with 99.9% removal efficiency as per API 692 standards. Figure 3.1 also displays a key on the top right where the channels are identified by color.

3.2 Component Selection and Analysis

The design of the test bench consists of the following sections:

1. Supply System Assembly
   a) Compressor
   b) Filter System
2. Rotary System Assembly
   a) Motor
      i) Interface: Motor to the main shaft
         - Coupler
         - Gearbox
         - Bearings
   b) Main Shaft Design

3.2.1 Compressor

The compressor is a vital part of the test bench of the system as it is the primary pressure supplier of the system. As a result, the compressor must adhere to the requirements necessary to test the dry gas seals. To perform the tests, a pressure of 220 bars is required with a flow rate in the range of 10-10^3 acfm.

Based on these criteria, the absolute evaluation was first conducted based on the feasibility, safety, and meeting customer requirements to determine the most suitable compressor type from the list of compressors shown in Figure 3.2.

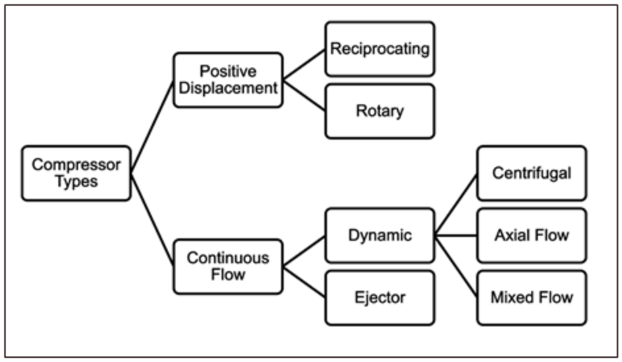

Figure 3.2: Compressor Types

The first compressor type examined was the ejector. It was determined that since the ejector does not contain moving parts and is a dependent source of pressure, it would have to be filtered out. The dynamic and reciprocating compressors were then examined to determine if they pass the evaluation. The following graph in Figure 3.3 illustrates visually the pressure supply capability of different types of compressors along with the flow rates associated with them. Based on the graph, the reciprocating and centrifugal compressors passed this evaluation, filtering out axial and mixed flow compressors.

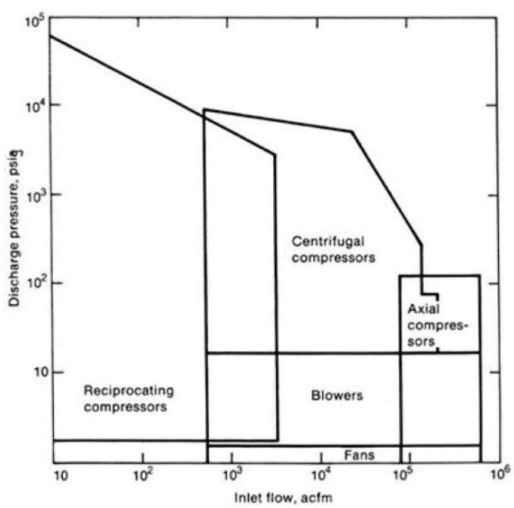

Figure 3.3: Pressure Ranges vs Inlet Flow. Taken from reference [8].

Finally, as for rotary compressors, operating within a small range of pressure varying from 45 – 125 psi, it was filtered out as they did not pass the evaluation [8].

The next step is looking in the market to determine the compressors’ model and type following the design choice for reciprocating compressors as the main pressure supply unit. As reciprocating compressors are notorious for their oil contamination, the first step was identifying whether to use a lubricated compressor or an oil free one.

Even though the lubricated compressor is most costly in the short time, in the long term, it is, in fact, less expensive when considering the maintenance cost and frequency. In addition to that, the contamination has been mitigated by incorporating a filter system in the test bench. Finally, as the test bench is expected to be stationary on-site, the portability of the compressor is not an issue needed to consider. As a result, the lubricated reciprocating compressor was the design choice for this application.

Finally, market research was conducted. Based on the cost, availability, and delivery time, the Adson Piston Air Compressor (Figure 3.4) was the final choice for the pressure supply to the test bench and the Denair High Pressure Piston Booster (Figure 3.5) as a backup [9].

Figure 3.4: Adson Piston Air Compressor

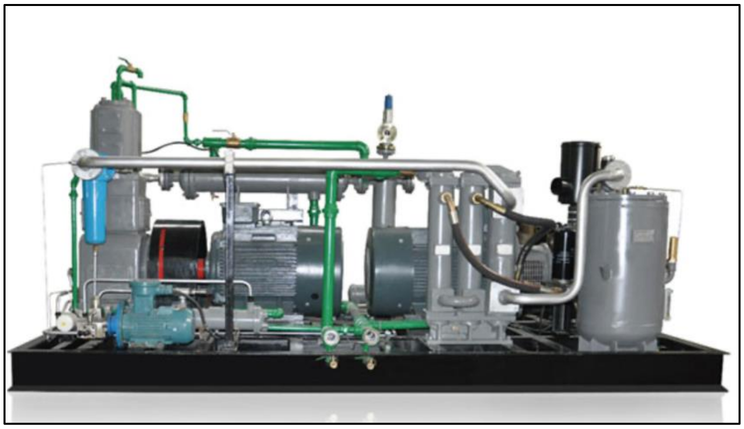

Figure 3.5: Denair High Pressure Piston Booster

3.2.2 Filter System

As mentioned earlier, the running gap between the seals is about 3- 5 µm which indicates that the primary gas supply and the secondary gas supply must be clean and dry. API 692 states that the filter system for gas supplies must filter to 1 µm spherical in size with 99.9% removal efficiency [5]. The filter system should also handle pressures up to 220 bar and temperatures up to 200℃ since our test bench's scope accommodates those pressures & temperatures.

After conducting market research, one filter system was found such that it follows the API 692 standard: John Cranes Filter System [10]. The system (see Figure 3.6) has the following features: capable of filtering contaminants of 1 µm and smaller with an efficiency of 99.9%, capable of removing liquids from the gas down to 0.005 ppm, pressure limits up to 413 bar and temperature limits of -95 C to 482℃. Furthermore, in terms of safety, the interlinked valves provide a safe and zero leak operation.

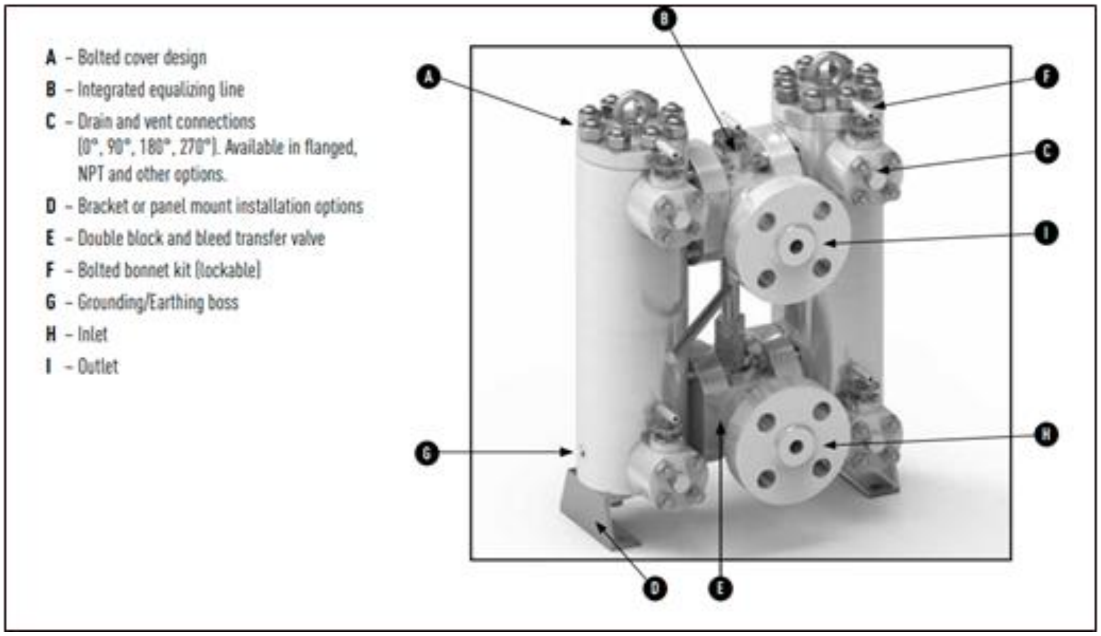

Figure 3.6: John Crane Filter System [11]

3.2.3 Motor

The motor is an essential part of the design, so it is important to investigate multiple motor options in order to get the most suitable one for the design. The motor will be connected to the shaft either directly or through a set of gears depending on the speed and torque output of the motor. In this investigation, three types of motor are studied through exposing them to the functional requirement as well as the relative evaluation. The motor types are: Electrical motors, Hydraulic motors, and Pneumatic motors. Each concept must satisfy the requirement of carrying the load to up 6000 RPM rotation speed. Table 3.1 below shows the values and assumptions used to estimate required power needed from the motor.

Table 3.1: Calculation details of the required power needed for the design

Parameters	Value	Source of estimation
Required Speed	6000 RPM	Requirement of the test
Moment of Inertia of the DGS	2.486 kg.m^2	DGS drawing sheet provided by Qatar Gas
Moment of Inertia of the Shaft	1.857 kg.m^2	Estimation Based on Shaft dimensions
Time set to accelerate the shaft	10 s	Assumption
Performance factor	1.5	Assumption
Required Acceleration	$\alpha = 0.8 \times 628\,\frac{\text{rad}}{\text{s}} \times \frac{1}{10\text{s}} = 50.3\ \text{rad/s}^2$	A factor of 0.8 is multiplied by the angular acceleration since the maximum torque occurs at around 80 % of the target speed. (In electric motors)
Required Torque	$\tau = 1.5 \times 50.3\,\frac{\text{rad}}{\text{s}^2} \times (1.857 + 2.486)\text{kg.m}^2 = 327\ \text{N.m}$	Torque is calculated as the product of the angular acceleration with the moment of inertia of the load multiplied by the performance factor
Required Power	$\text{P} = 628\,\frac{\text{rad}}{\text{s}} \times 327\text{N.m} \div 745.7 = 276\ \text{Hp}$	Calculation

3.2.2.1 Electric Motor

Electric motors are the most known types of motors; they convert electrical power into mechanical power using the electromagnetic forces within their component. Electric motors are also the most used types of motors for various applications as they are easy to implement and power; they can also provide huge power relative to their size.

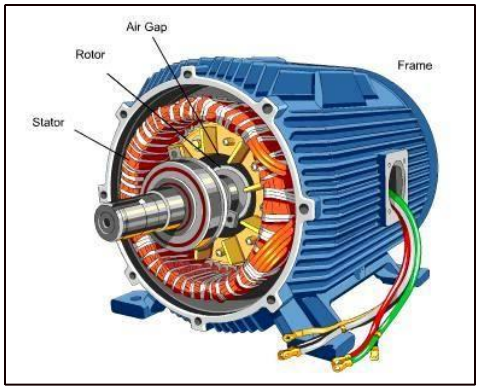

Figure 3.7: Electric motor schematic [12]

Electric motors can be either DC or AC motors in which DC motors can operate at different speeds depending on the supplied voltage while AC motors usually operate at fixed speed unless power inverter or variable frequency drives are used to control the speed. In this design, AC motors are considered to drive more power than DC ones.

3.2.2.2 Hydraulic Motors

Hydraulic motors are the type of motors that convert hydraulic pressure into mechanical work. They work by providing pressurized fluid to the motor. Hydraulic motors are commonly used in applications that require high torque such as winches and crane drives.

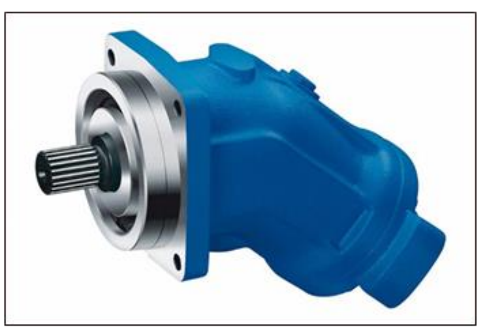

Figure 3.8: Hydraulic Motor [13]

Hydraulic motors come in diverse types and shapes. Figure 3.8 shows the angled piston type hydraulic motor. Each type has its own method of operation as well as its advantages and drawbacks. In general, hydraulic motors are simple in design and operation compared to the electric motors with medium cost depending on the required power. They are used not only for their power but also for their control accuracy. However, they are slow in operation, and they should not be run in a hot environment as that will affect the viscosity of the operating fluid.

3.2.2.3 Pneumatic Motor

Since the main requirement of the design is to supply high rotational speed, the Pneumatic motor might seem to be good candidates as this type of motor is known for its high speed and weight to size ratio with a simple control system. Pneumatic motors, also known as air motors, are motors that work by expanding compressed air [14].

Figure 3.9: Pneumatic Motors [15]

Like Hydraulic motors, pneumatic motors convert pressure energy into mechanical energy (see Figure 3.9). They are used in many industries such as food, chemical and agricultural industries. Their main advantage is they are fitted for heavy duty operations since they are very flexible in terms of their safety and reliability. [15]

Based on the current knowledge and information about the given motor types, electric motors are the best options for the DGS test bench design. However, the evaluation is subject to change based on the progress of the study.

There are several options for the electric motor that can be found online that can match the requirement of 296 Hp or 205 KW. A candidate for the design is the powerful three-phase motor: “W22 IE2 370 kW 4P 355M/L 3Ph 380-415/660//460 V 50 Hz IC411 - TEFC - B3T” (see Figure 3.10). [16]

It is worth mentioning that this product's specifications exceed the needed requirements, but its additional features are worth the investment as they facilitate the use of the motor and ensure safe use of the motor. Also, the excellent performance of this product makes it safe to use as it not going to be used at its full load which will increase its lifetime as well as its maintenance intervals.

Figure 3.10: Image of the electric motor product by WEG that is suitable for the design

Table 3.2 shows the features and specifications of the chosen motor. It is noticed that the RPM of the chosen moto is less than the requirement, but the power output is more than required thus a gearbox is utilized to increase the RPM of the shaft.

Table 3.2: Motor Choice

W22 IE2 370 kW Electric Motor From "Weg"	Features and Specifications	Reason of choice
	Power output: 370 KW (500 Hp) Rated Speed: 1500 RPM Three Phase electric motor with 4 poles Rated voltage: 380V / 660 V Frequency: 50 Hz	Excellent Ventilation system for safe operation Fast control response Excess power output potential

Since the motor chosen has maximum speed of 1500 RPM, a gear box with gear ratio 1:4 is needed to increase the rotational speed to 6000 RPM (see Figure 3.11).

Figure 3.11: Gearbox

In order to provide a means to rotate the shaft, different motor types have been investigated till an electric motor with a power output 500 Hp is chosen. The electric motor is chosen due to its high output power, fast response, and safety system. The motor designed to provide a top speed of 1500 RPM, so gear box of gear ratio 1:4 is implemented to meet the requirement of 6000 RPM on the load that consist of consist of the dry gas seal, the shaft, the gearbox, and the gear couplers.

3.2.3 *Coupler*

There are multiple shafts in our assembly that have different diameters. The motor shaft is 20 mm in diameter, whereas the shaft housing the Dry Gas seal will, at the least, be 45 mm in dimension.

This is because the smallest seal the bench is being designed to test has an inner diameter of 45 mm, which the shaft will go through. We need to provide a means to transmit torque from the small shafts of the gearbox/motor to the main shaft.

One option which was explored is making use of a pair of spur or helical gears that will transmit torque from the output gearbox shaft to the main shaft (see Figure 3.12). Spur gears, while easy to design and manufacture, are only suitable for slow to moderate speeds. They are noisy at high RPMs, and the gear teeth undergo significant wear. Moreover, they are known to vibrate significantly at high speeds, which would accelerate wear and be transmitted to the main shaft housing the Dry gas seal resulting in unwanted axial motion [17]. On the other hand, helical gears are less efficient since more teeth are in contact with each other. This would result in a loss of mechanical energy we intend to transfer to the main shaft. Also, helical gears require consideration of unwanted axial forces, which we are trying to avoid. Axial forces would result in unwanted transverse forces. [17] [18]

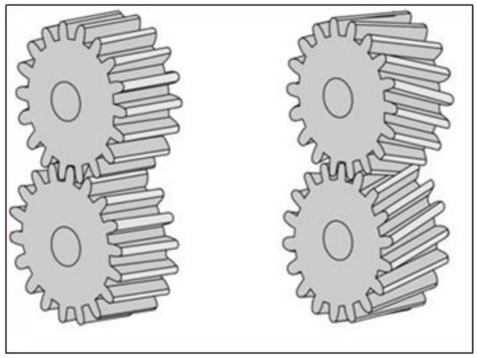

Figure 3.12: Left to right, a pair of spur gears, helical gears [19]

Since gears require additional components to be made safer to use, alternatives were researched by the team. Couplers, per our literature review, fit our requirements. By definition, couplers are devices that are used to join two shafts to transmit torque. Additionally, couplers can limit any axial misalignment, which is advantageous for our use case [20]. Axial misalignment results in a loss of mechanical energy. This results in a lower torque being transmitted to the main shaft. Moreover, it could also damage the shaft, causing it to contact the housing, resulting in wear. Different kinds of couplers exist on the market for varying applications as shown in Figure 3.13.

Figure 3.13: Different types and variations of couplers available in the market [21]

Any selected configuration of couplers chosen must meet the following requirements:

- The coupler must be able to match shafts of large diameters, i.e., up to 350 mm
- The coupler must have high torsional rigidity to prevent deformation when transmitting loads
- The coupler must have a high dynamic load capacity, i.e., greater than 330 kN.m
- The coupler must be able to operate at rpm's greater than 6000 rpm

SKF Co. is a company that specializes in making shaft interfaces. They have an extensive inventory of couplers and are based locally in Qatar. The team studied the data sheets and catalogs for shaft couplers to determine the configurations capable of meeting the design requirements. Table 3.3 shows two possible couplers available from SKF that meet our needs. Gear and Grid couplers both fulfill the condition of fitting large shafts and having a high dynamic load capacity. Gear couplers, however, win since they have a higher maximum rpm limit.

This choice is further confirmed by other Shaft Coupler Manufacturers, like Vulkan Drive Tech and RingFeder Power Transmission tech. Since SKF is currently the only manufacturer in Qatar capable of providing large gear couplers and provide extensive data on their hardware, they will be our choice.

To summarize, Gear couplers were chosen to connect a pair of shafts with differing dimensions. SKF Co. was chosen as the supplier for the couplers since they are based locally in Qatar and have an inventory meeting our design requirements.

Table 3.3: Gear and Grid coupler properties

Coupler	**Shaft capacity**	**Maximum torque capacity**	**Maximum rpm**	**Torsional rigidity**
Gear	13 – 425 mm	550 kN. m	8000 rpm	High
Grid	13 – 350 mm	400 kN.m	8000 rpm	Medium

3.2.4 Bearings

Throughout the design, the transfer of radial load needs to be assured. Moreover, friction between the rotating shaft and its housing needs to be reduced to protect the shaft from wear and loss of mechanical energy. Hence, the interface must provide a means to reduce friction and transfer motion. Bearings, by definition, transfer motion and are designed to reduce friction. Any bearing being used in the system must meet the following requirements:

- Must be able to fit shafts with diameters up to 350 mm, i.e., the Inner diameter of the largest seal our bench must test.
- The bearings must be able to rotate at speeds greater than 6000 rpm

Since only radial loads are being transferred, ball bearings are typically the best choice in such cases as shown in Figure 3.14. However, during market research and literature review, it was noted that ball bearings with large inner diameters have small speed ratings. That is, if a bore with 200 mm is selected, the maximum possible speeds it can be safely operated at is up to 2000 RPMs [22] [23]. This is significantly lower than the requirements for our design. Other mechanical bearings were looked at, such as cylindrical roller bearings and Needle roller bearing [22] [23]. However, these too were limited by lower speed ratings at high inner diameters. Theoretically, bearings can be operated at speeds higher than their rated speeds. This would require careful thermal and wear analysis to ensure the bearings do not fail while working. Moreover, the mechanical bearings would need to be replaced after every test run to ensure a safe operation.

Figure 3.14: Ball bearing units with housing in varying sizes [23]

Another alternative is to use Magnetic bearings as shown in Figure 3.15. Magnetic bearings support moving parts without touching physical contact. They make use of magnetic levitation to transfer load. Due to lack of physical connections, there is very low friction in the system using Magnetic bearings and reduced chances of wear. Initial market research shows that magnetic bearings exist which can support shafts up to 300 mm rotating at speeds greater than 15000 rpm [24] [25]. More market research needs to be done to ensure the magnetic bearings can support transmission of large torques and if the investment in magnetic bearings is economically feasible when compared with the option to replace ball bearings after every use.

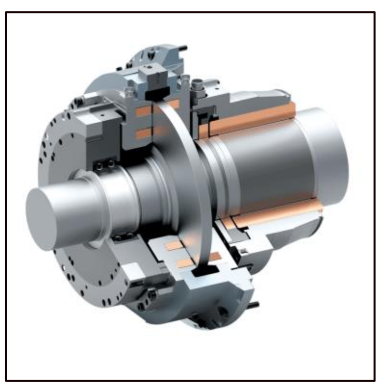

Figure 3.15: A magnetic bearing from SKF with an inner diameter of 220mm [25]

In summary, a cost analysis needs to be done between Magnetic bearings and ball bearings to see which are more economically feasible. Moreover, a detailed thermal analysis needs to be conducted to ensure that the ball bearings being used will not fail during operation.

3.2.5 Main Shaft

The main shaft is designed as the shaft housing the Dry Gas Seal. This is a critical part of our test bench since it transfers torque to the Dry Gas Seal enabling it to build a running gap in between the seals. The main shaft needs to satisfy the following set of requirements per the design:

- Have a Critical speed far greater than 6000 RPM to prevent rotation at natural frequency.
- Stress concentration at critical points i.e., fillets and grooves need to be less than the yield strength of the material to prevent deformations or yielding.
- Must be able to bear loads greater than 300 kg without undergoing significant deformation or deflection.

The main shaft needs to be designed with a safety factor of at least 2 to ensure the shaft can survive twice the maximum load. Based on ISO R-775 Standards on Shaft Ends, a shaft can be designed with a keyway for dimensions up to 600 mm [26]. This meets our requirements, and the Standard ensures that the critical speed is avoided and the stress concentrations at the grooves are within limits. There is no specific main shaft; for every Dry Gas seal being tested, there will be a different main shaft to fit the Dry Gas Seal. This means that there will be a family of main shafts that can fit a specific Dry Gas Seal for the test bench. Further study is required to determine the range of dimensions for these main shafts.

Materials that the main shaft can be made from were studied. The shaft is going to be loaded in torsion and, at the same time, will be bearing the weight of a pair of Dry Gas Seals.

Based on this, material indices were utilized to find materials that maintain their strength when loaded in torsion and axially. The two material indices used are:

1. $\frac{\sigma_f}{\rho}$
2. $\frac{G}{\rho}$

Where G is Shear Modulus, σ_f is yield strength and ρ is density of the material. Figure 3.16 and Figure 3.17 below show the result of maximizing the above-mentioned material indices [27].

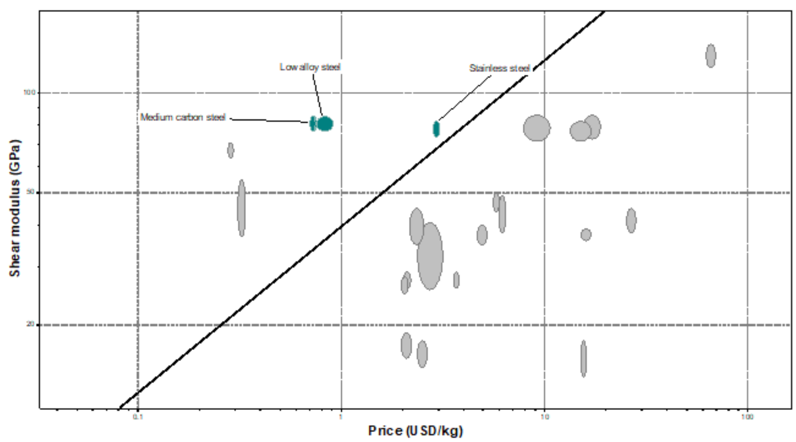

Figure 3.16: Material index showing specific torsional stiffness

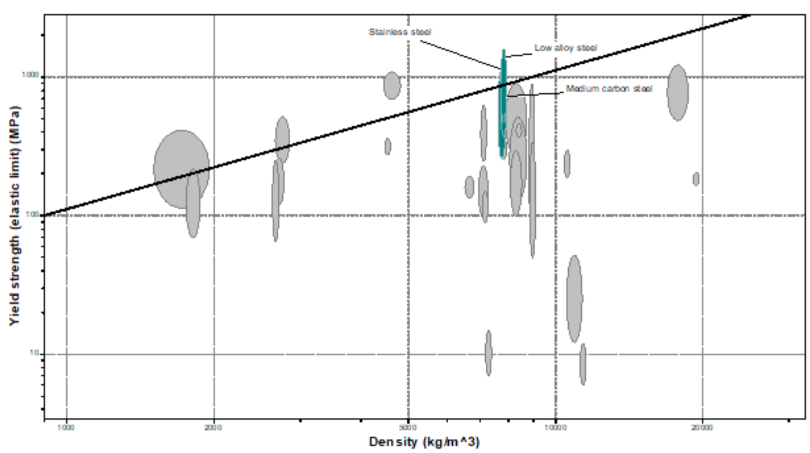

Figure 3.17: Material index showing specific axial stiffness

Based on these indices, we get either Medium Carbon steel, Stainless steel, and Low alloy steels. Steels are more accessible to machines with conventional CNC techniques. Moreover, they are readily available locally from manufacturers like Qatar Steel. A static loading analysis of the main shaft was conducted in SAP2000 with 600 kg loading applied at two different points mimicking the weight of the Dry Gas seals (see Figure 3.18 and Figure 3.19). The deformations throughout the length of the beam are approximately 0.02 mm. This is insignificant compared to the size and diameter of the shaft. Stress analysis shows stresses of 3 MPa at the point of loading, which is well below Steels 220 MPa yield strength. Note the dimensions of the shaft in the model are as follows:

- 1.1 meters in length
- 250 mm in diameter
- Load applied 300 mm from the joints

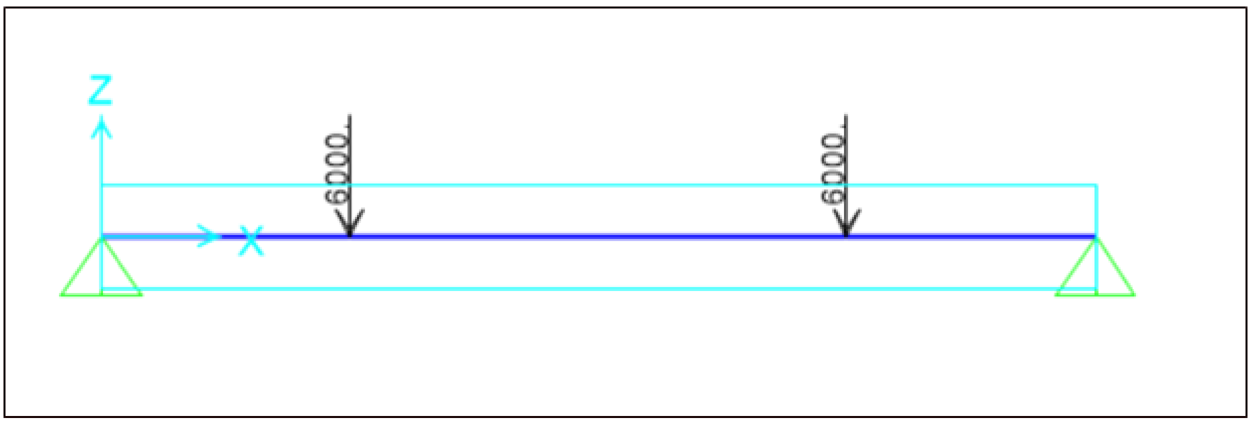

Figure 3.18: Model on SAP 2000, the shaft has a length of 1.1 m per ISO R 775 Standards

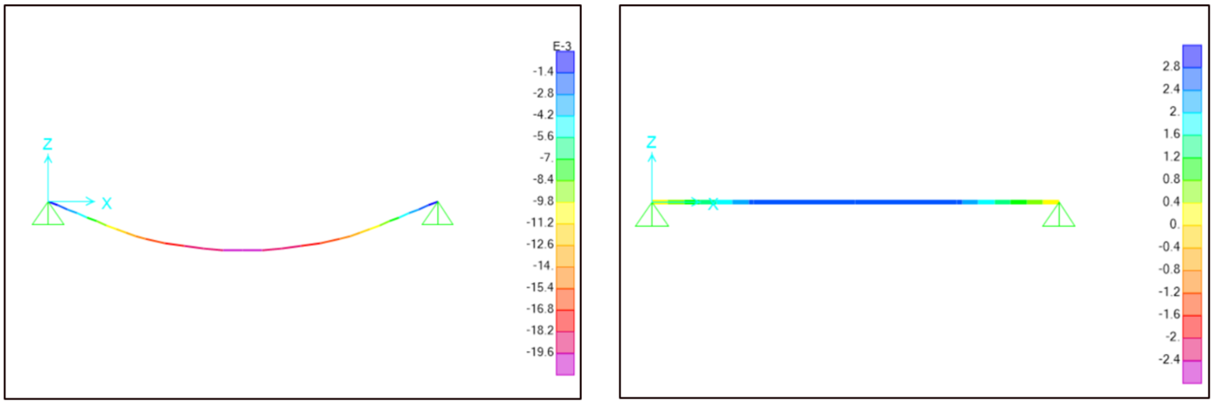

Figure 3.19: a) Contour plot of deflection along the length of the shaft. The maximum deflection is 0.0196 mm at the center of the shaft. b) Contour plot of stress on the shaft, the maximum stress is 2.8 MPa

To summarize, ISO R-775 Standards were chosen as a basis of our main shaft design since they have extensive literature on dimensioning shafts to avoid stress concentrations in keyways and fillets. Steels like Carbon Steel and Stainless steel are chosen as the shaft material since the have the highest specific torsional and axial rigidity. Figure 3.20 shows the final shaft and motor assembly that was created using SolidWorks.

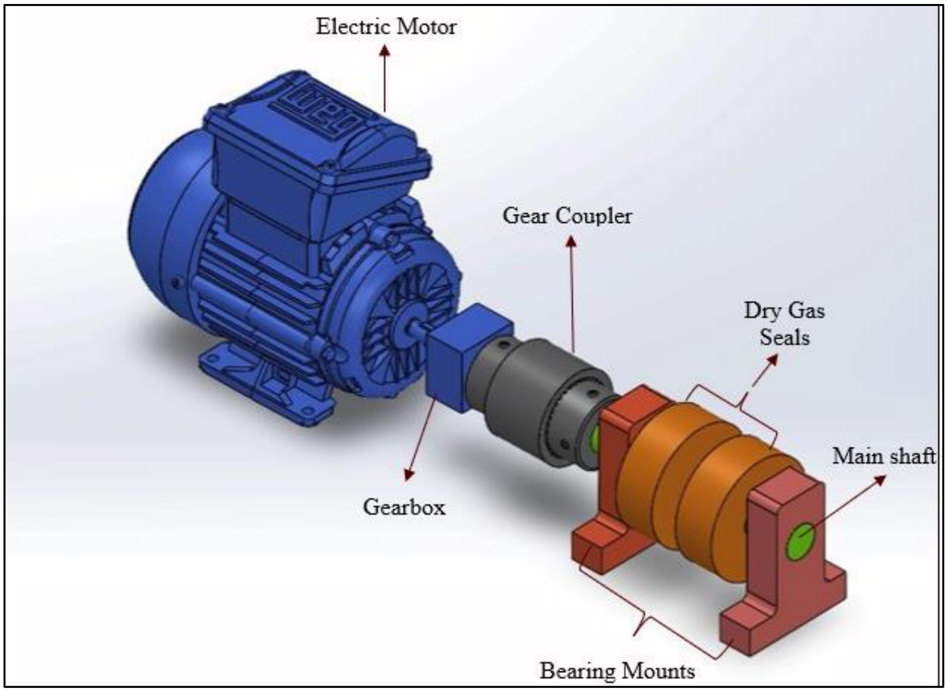

Figure 3.20: An overview of the shaft and motor assembly, not drawn to scale

4. CONCLUSION

Centrifugal compressors are important equipment in natural gas companies. However, due to the toxicity and flammability of natural gases in the compressors, dry gas seals are used to limit their leakage into the environment. They need to be maintained and tested regularly, so designing a reliable testing system for the dry gas seal is very important to ensure a safe working environment in the company.

There are four main systems: supply system, rotary system, control system, and pipe networking system. The main components of the test bench have been chosen, but the design is not yet complete as it is missing the control and pipe networking system. For our future plans, we aim to complete the analysis of the pipe network using ANSYS such as the pipe connection between the filtration system and the housing of the dry gas seal. In addition, design a control system responsible for controlling the flow and pressure in the piping network as well as the rotational speed of the shaft such that it complies with the API 692 standards. Finally, designing the housing for the rotary system (the dry gas seal and shaft assembly).

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