DESIGN OF A ROTARY VALVE FOR PRESSURISED STEAM

TALENT NYAWO

KTH Industrial Engineering
and Management

Master of Science Thesis
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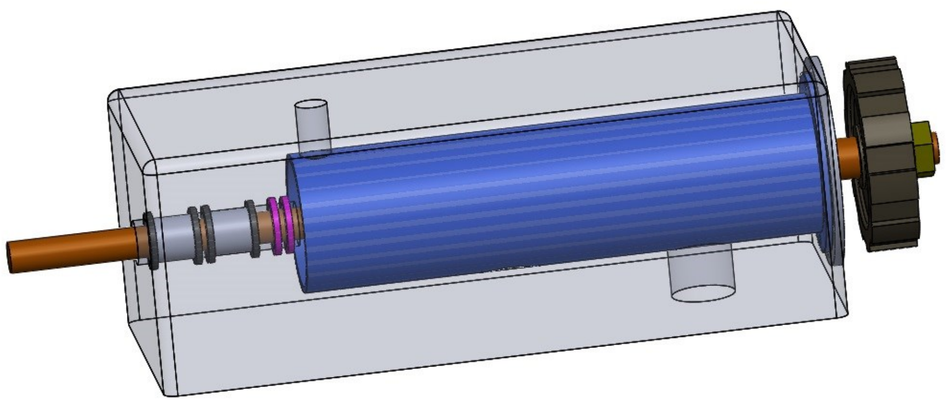

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Utveckling av rotationsventil för trycksatt ånga

Talent Nyawo

Godkänt	Examinator	Handledare
2016-09-15	Ulf Sellgren	Kjell Andersson
	Uppdragsgivare	Kontaktperson
	Ranotor AB	Peter Platell

Sammanfattning

Denna rapport är gjord på ett examensarbete som var utfört på uppdrag av det svenska företaget Ranotor AB. Syftet var att utveckla en konceptuell lösning för en rotationsventil som skall fungera i en miljö med hög temperatur och högt tryck. Ventilen skall arbeta under höga rotationshastigheter, vilket kräver korta öppettider.
Tekniska hjälpmedel såsom SolidWorks, ANSYS och MATLAB användes för att modellera och analysera de konceptuella lösningarna.
Slutlösningen valdes från ett flertal olika koncept, varpå detta vidareutvecklades och optimerades. Betydande material och gastätningslösningar identifierades och utvärderades för att hitta den bästa lösningen. Optimering av individuella komponenter och hela anordningen gjordes med avseende på spänning, termisk- och dynamisk analys. De givna specifikationerna uppfylldes och resultaten var tillfredsställande.
Resultaten ger en teoretisk bas för vidareutveckling och applicering av en rotationsventil in en miljö med hög temperatur och högt tryck.

Nyckelord: roterande ventil , uniflow motor, gastät försegling , rörlig bryttidpunkt , hög-tryck-hög-temperatur

KTH Industrial Engineering and Management	**Master of Science Thesis MMK 2016:107 MKN 176** **Design of a rotary valve for pressurized steam** Talent Nyawo	
Approved 2016-09-15	Examiner Ulf Sellgren	Supervisor Kjell Andersson
	Commissioner Ranotor AB	Contact person Peter Platell

Abstract

This Master thesis is a project commissioned by the Swedish company Ranotor AB. The objective of this thesis is to develop a conceptual solution for a rotary valve mechanism that has to work efficiently in a high-temperature and high-pressure environment. The valve is to operate at high rotational speeds which calls for very short opening time.

Modern engineering tools namely Solidworks, Ansys and Matlab, were employed for modelling and analysis of the conceptual solution.

The best design solution was selected from three developed concepts, and the selected concept was further developed and optimized. Major material candidates and gas-tight sealing solution were identified and evaluated and the optimal material and seal design was chosen. Optimization of the individual components as well as the whole assembly was performed based on stress, thermal and dynamic analysis. The given design specifications and functions were fulfilled and the results were satisfactory.

The obtained results provide a theoretical foundation for the development and application of a rotary valve in high-temperature and high-pressure environment.

Keywords: *rotary valve, uniflow engine, gas-tight sealing, variable cut-off time, high-pressure-high-temperature*

FOREWORD

The author would like to express his gratitude to Ranotor AB for the opportunity to work on this interesting and challenging project.
Particular thanks goes to Ovel and Peter Platell for their invaluable support and research at Ranotor AB.
Special appreciation goes to my supervisor Kjell Andersson and my examiner Ulf Sellgren at KTH Royal institute of Technology, for their great knowledge, advise and interesting discussions.

Talent Nyawo

Stockholm, June 2016

NOMENCLATURE

This section lists the Notations and Abbreviations that are used in this Master thesis.

Notations

Symbol	Description
E	Young´s modulus (Pa)
F	Filling ratio (%)
L_s	Stroke length (mm)
M_b	Bending moment (Nm)
p	Pressure (Pa)
P	Power (W)
P_d	Design power (W)
P_{corr}	Corrected power (W)
Q	Volumetric flow rate (m^3/s)
r	Radius (m)
t	Time (s)
T	Temperature (°C)
v	Velocity (m/s)
W_b	Section modulus (m^3)
Z_f	Opening length (mm)
ρ	Density (m^3/kg)
Δh	Change in enthalpy (kJ/kg)
φ_f	Filling angle (°)
σ_e	Equivalent stress (Pa)
σ_y	Yield stress (Pa)
τ	Shear stress (Pa)

Abbreviations

BDC	Bottom Dead Center
BRV	Bishop Rotary Valve
CAD	Computer Aided Design
CHP	Combined Heat and Power

CAE	Computer Aided Engineering
HTHP	High-temperature-high-pressure
IC engine	Internal Combustion engine
ms	Milliseconds
rpm	round per minute
SF	Safety Factor
TDC	Top Dead Center

Table of Contents

1. Introduction

This chapter outlines the project introduction, the targeted goals, limitations and the method used in this project.

1.1. Project Introduction

This Master thesis is based on the subject of designing and modelling a valve concept which is part of a modern steam expander. The project is in collaboration with KTH Royal Institute of Technology, and the Swedish company Ranotor AB. The rotary valve is a critical component of a modern steam expander. The other part of the overall project involves the designing of the cylinder package, which has to be assembled together with the rotary valve mechanism and prototype the whole engine.

1.2. Thesis Project Aim

The objective of this project is to generate and analyse a conceptual solution of a valve system that can work efficiently in the conditions of high-temperature and high-pressure as well as meeting the given specifications.

This thesis project is a subproject of a ‘big project’ which has the overall aim of having a full CAD model of the whole engine which is going to be prototyped in order to test the feasibility and compactness of the engine. Several projects have been conducted before, working on different components of the engine. The research and skills from different engineers together with the vast experience gathered by Ranotor AB were used as input in these projects. A PhD thesis has also been carried out with the aim of developing an analytical rigid body kinematics and inverse dynamics model of the steam expander. The results from these projects are to be used as the foundation on which this Master thesis is based on.

Ranotor AB is aiming at putting all the accumulated experience in visualising the technology through prototyping of a new multi-cylinder axial piston steam engine. The engine involves several new components such as a single rotating valve, novel sealing in the crankshaft and on the piston.
Ranotor has investigated several concepts for the modern steam engine. During the years a concept that involves a multi-cylinder axial reciprocating piston engine has been developed that seems to offer the best overall qualities.
Ranotor has carried out substantial analysis work on thermodynamics and fluid mechanics for this modern steam engine. The company has also carried out kinematic analysis of the axial piston engine that is laying the foundation to this thesis work.

1.3. Deliverables

The following are the expected outcome of this project

- A complete conceptual solution of the rotary valve including the housing and sealing
- CAD model of the proposed solution (3D and 2D in Solidworks)
- Design concept description (working principle)
- Analysis of the concept (dynamics, mechanical stresses and thermal effects)
- Selected material

1.4. Delimitations

In order to achieve the aimed goals within the given timeframe, the project was limited to the above expected outcomes and the following aspects are beyond the scope of this thesis project:

- Tribological and wear analysis
- Computational fluid dynamics

1.5. Method

The project starts with background research where prior art in the engine technology and valve systems is reviewed. Information is gathered from different sources like scientific articles and patents. After information gathering, at least two concepts are to be generated where the best solution will be selected using the Pugh Matrix. Each component of the selected mechanism is to be allocated a material before modelling for optimal functionality and strength. Concept modelling include stress, thermal and dynamic analysis of individual components and the assembly as a whole is to be performed. A number of modern engineering tools are to be employed for analysis. This include modern CAE tools like Matlab, Ansys, and SolidWorks.

2. State of the art

This chapter gives the details of the overall project and the technology behind it. This is the base on which this thesis is established.

2.1. Background information

During the oil crisis in the 70s the Swedish car manufacturer SAAB launched an ambitious program to develop a modern and compact steam engine. This seemingly promising technology was later abandoned due to inefficient, bulky and insufficient performing solutions. (J. P Norbye, December 1974). The emergency of new materials and technology, increasing demand for low fuel consumption and lower harmful emissions has justified the 'rebirth' of a steam engine.
A modern steam engine is full of potential and is promising in developing alternative fuels and making use of natural resources which are more friendly to the environment and can be helpful in future as oil resources are running out.

Recent research has proved the possibility of a modern high efficiency, powerful and compact steam engine. (P. Platell, 1993). A modern steam engine has several key advantages over the conventional internal combustion engine. These advantages include the following;

- Fuel flexibility: this include different forms of biomass, waste products from the petrol industry, by-products from industries, as well as waste products from agricultural and forestry industries.
- Full torque at low rpm and maximum efficiency at part load. This is contrary to the internal combustion engine which gradually offers a maximum torque and reaches its best efficiency close to full load.
- Free NO_x emissions
- High power density: contrary to popular belief, a modern high performance steam engine has inherent possibility to offer high power density.
- With a thermal battery, a modern steam engine can offer regenerative engine braking and storing cheap and green electricity in the same way as electric propulsion.

With the support of the US military, the Florida based company Cyclone Power has developed and built a modern steam engine. On the other side, the Swedish based company Ranotor AB has continued SAAB's program in developing a high performance modern steam engine. These modern research and findings have proved beyond doubt that a compact modern steam engine is practically possible.

Ranotor AB has also patented a number of key technologies that would allow its technology to be employed in the automotive industry. Another possible application of the steam engine is in CHP (combined heat and power), and the recovery of waste heat from industrial process and to save as a range extender in an electric vehicle.

Steam engine*:* A steam engine is a heat engine that performs mechanical work using steam as its working fluid. (Wikipedia) Heat is received from a high temperature source and steam is formed in the steam generator. The generated heat is then converted to mechanical work, through the utilisation of a piston expander. There are different piston expanders as well as thermal cycles which can be used depending on the required power, fuel supply, and other restrictions. The Rankine cycle and the Sterling cycle are some of the examples of the cycles that are employed in a modern steam engine.

2.2. Overview of the piston expander

Basing on the conventional concept of a steam engine, Ranotor has investigated and developed several concepts of a modern steam engine. A multi-cylinder axial reciprocating piston engine has been developed and it seem to offer the best overall qualities in terms of cost/performance. The engine involves several new components such as a single rotating valve, novel sealing in the crankshaft (wobble plate) and on the pistons.
The configuration of the engine is shown in Figure 1:

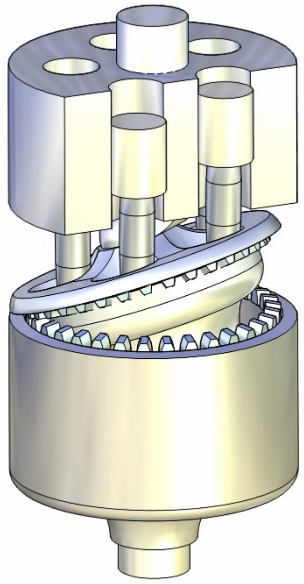

Figure 1: Representation of an axial piston expander with the wobble plate. (R S. Löfstrand, 2009)

The main components of the axial piston expander are the cylinder head, the piston, the piston rod, wobble plate, crank shaft, and the engine block.

Figure 2 gives the schematic representation of the engine.

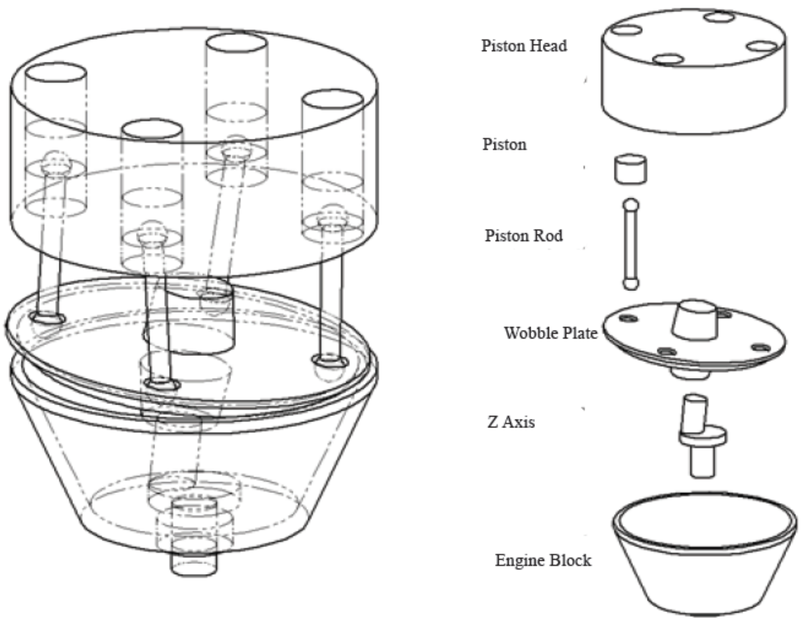

Figure 2: Schematic representation of the piston expander

The main concept of the steam engine consists of a five-cylinder axial piston expander with a wobble plate. The wobble mechanism connects the pistons and the crankshaft. In-between the wobble plate and the pistons are the piston rods.

There are different configurations of the axial piston expander depending on how the tilted plate is connected. Depending on the engine configuration, the tilted plate has different names like the swash plate, the bent axis or the wobble plate. (T. C. Dickenson, 1999)

In the wobble plate configuration, the reciprocating motion of the pistons is transformed to the rotational motion of the engine shaft by the use of the piston rods and the wobble plate. A full cycle of each piston gives a complete revolution of the engine shaft. The engine shaft (also called the Z-shaft) delivers out the mechanical energy performed by the pistons. Pure nutation is achieved through the synchronisation of the wobble plate which is connected to the engine block via a gear.

The pistons are disposed around the crankshaft in a concentric circle, the wobble plate is responsible for controlling the axial position of the pistons in the piston head. The steam will be filled and emptied as the crankshaft rotates.

2.3. Rotating valve as part of the piston expander

The valve system on the expander is responsible for controlling the flow of the steam in and out of the cylinders. The rotating single valve is part of the new components introduced in the modern steam expander.
The valve has been proposed many times before in engine history due to its many inherent advantageous features over poppet valves. It was claimed that improved breathing and burning characteristic with the rotating valve gave higher specific power and higher efficiency than conventional poppet valves.
These features were satisfactorily demonstrated but there were problems that prevented the concept to be commercially viable. The problem was to realize a design that offer both reliability and low oil consumption. To get long life-time and reliability it took a lot of oil that flooded the valve at low operation speed where sufficient hydrodynamic lubrication is not present.
With a rotating valve an axial piston engine will offer attractive synergies when it comes to realizing a very compact engine.
The rotating valve is probably the most critical component in the high speed (5000-6000 rpm) modern steam engine. It has to operate without oil as lubricant and this impose tribological challenges. However modern materials seem to offer the desired properties.
The tribological problems that engineers faced during the 80s will likely be easier to solve with modern material coatings. It seems that presence of water and certain coatings might offer attractive tribological conditions.
Ranotor has developed prototypes of a rotating valve for an axial piston engine, which is illustrated in Figure 3.
In Figure 3, an AC compressor is shown with a rotating valve on top of it. The figure is just to illustrate how a multi-cylinder axial piston engine with one rotating valve in the centre would look like. To the right of Figure 3 the rotating valve is inserted in a rotating valve housing.

Figure 3: Illustration of the piston expander with a single rotating valve

3. Prior art in valve technology

This chapter is a preliminary study of valve technology specifically used in engines. A review of different valve and actuation systems is presented.

3.1. Valve overview

Valves are found in a wide range of applications. This chapter focuses on different mechanisms of engine valves, different valve actuation systems, and valve patents related to this project.

In general engine valves are classified into three main groups namely

- Poppet or mushroom valves
- Sleeve valves
- Rotary valves

3.1.1. Poppet valves

Poppet valves are the commonly used type of valves in internal combustion engines. This valve type is actuated by the use of cams and/or rocker arms whereas a spring closes the valve by forcing it against the valve seat. Their main advantages of poppet valves over other valve types are:

- It's a simple system
- Consistent response times
- Easy maintenance of the sealing
- Provides tight sealing, meaning virtually no leakages

Their main drawbacks are;

- The need for stiffer springs for quick response of the valve. This results in increased frictional loses and high power demand
- Noisy operation
- Valve floating at high rpm

However, poppet valves have proved to be beneficial for the most part and they have dominated the automotive industry in internal combustion engines. Figure 4 illustrates the general layout of a poppet valve.

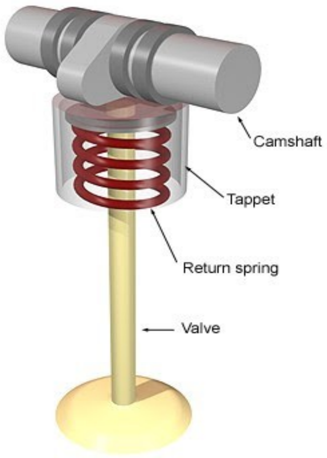

Figure 4: An example of a poppet valve. (Google/Images)

3.1.2. Sleeve valve

This valve mechanism is made up of machined sleeves which slides between the piston and the cylinder wall in an internal combustion engine (Figure 5) The sleeve ports align with cylinder's inlet and exhaust ports during the engine cycle:

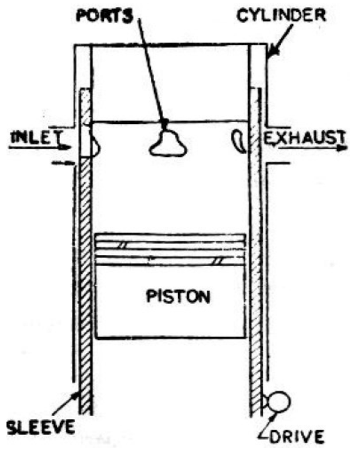

Figure 5: Sleeve Valve (Google/Images)

Its main advantages over other valves are:

- Simplicity of construction
- Long lifespan and silent operation
- Constant power demand due to the absence of springs
- Increased volumetric efficiency

Disadvantages.

- High oil consumption
- Leakages due to difficulty-to-achieve sealing solutions
- Limited speed of up to 3000 rpm

3.1.3. Rotary valve

This type of valve regulates fluid flow by the rotation of a spool which communicates alternately with inlet and exhaust ports. The rotating spool aligns with holes in the valve casing to give the required operation which could be either intake or exhaust. Figure 6 is an example of a rotary valve.

Figure 6: Bishop Rotary valve assembly. (T. Wallis 2007)

Rotary valve has unhindered flow thus enhance the power output of the engine, they offer a full port instead of a port partially blocked by a poppet valve which obstruct flow, induces pressure drop and reduces intake.

Advantages:

- Uniform motion,
- Quiet operation
- High compression ratio
- Reliable at high rpm
- Compact and lightweight cylinder head designs
- Minimal actuation energy

Disadvantages;

- Difficulties in pressure sealing,
- Uneconomical valve lubrication
- Mechanical and thermal distortion

However, regardless of the numerous advantages offered by rotary valves, their use in engines has been to a limited extend.

3.2. Valve actuation systems

Valves require an additional mechanism that actually opens and closes them. Special devices are required to automatically or remotely open and close the valve. These devices are called actuators. "Actuators which control the movement of a valve can be manual or automatic and are a major ancillary item for valves." (T. C. Dickenson). There are several valve actuation systems that can be employed for this operation. These different actuation mechanisms include mechanical, pneumatic, hydraulic, and electrical systems.

3.2.1 Pneumatic valve actuation

This actuation system uses compressed air to provide the force required to open the valve. The closing force can either come from the pneumatic system itself, a spring or both. A spring can be used to close or open the valve and the actuator performs the reverse operation. A double acting actuator can be used for both closing and opening the valve (Figure 7). The compressibility of air makes it difficult for pneumatic actuators to have a smooth motion and high accuracy. Pneumatic actuators are also limited in their application because of their high losses due to leakages and limited forces. Despite all this, pneumatic actuators are known for their quick responses and do not require a large amount of forces.

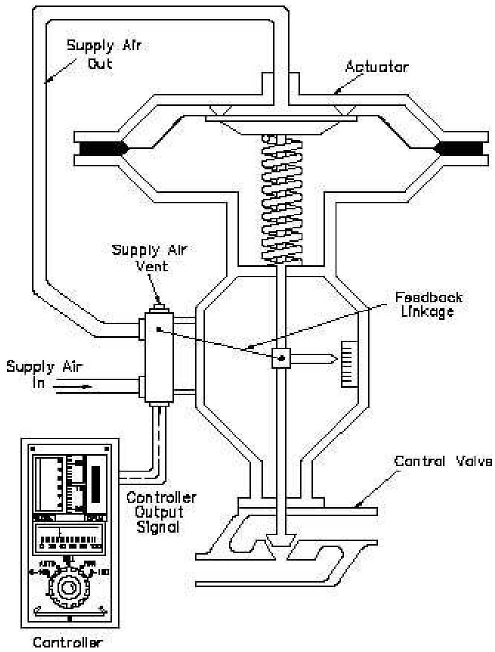

Figure 7: An illustration of a pneumatic actuator. (Google/Images)

3.2.2 Hydraulic actuation

This uses the same principle as pneumatic actuators except that it uses pressurized fluid instead of compressed air. Rotary valve hydraulic actuators are used for rotary motion and linear valve hydraulic actuators provide linear motion to linear valves. Hydraulic actuators are more sophisticated and expensive and are limited in their operating conditions as hydraulic oil cannot withstand high temperatures. Hydraulic actuators can be used for fail safe operations and provides high forces. Figure 8 is an illustration of a hydraulic actuation.

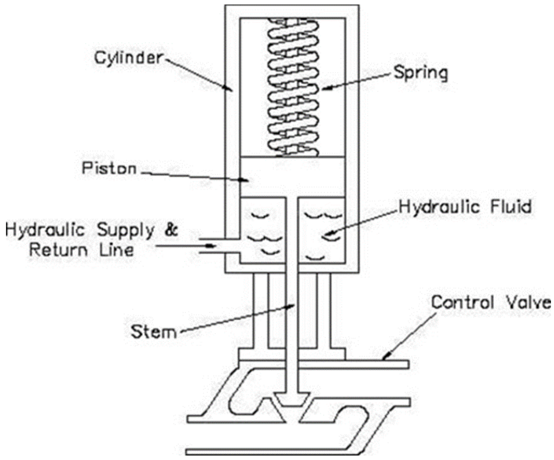

Figure 8: Hydraulic actuation system. (Google/images)

3.2.3 Electrical actuation

These include solenoids, motors (stepping, DC, and AC). They offer quick response and are easy to install. Their main drawback is that they only offer two operating conditions, either fully open or fully closed. Solenoid actuators are limited in their applications because they offer small actuation forces. Figure 9 illustrates an electric solenoid actuator.

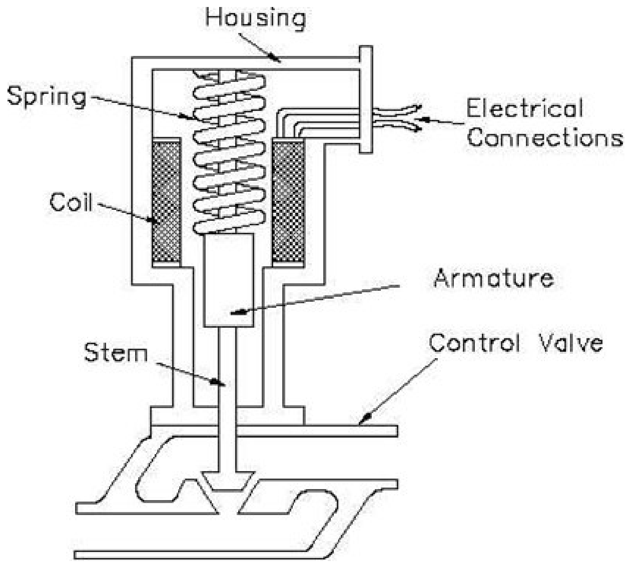

Figure 9: Electric solenoid actuator. (Google/Images)

3.2.4 Mechanical actuation

These include camshaft, rocker arms, lifters, and push rods (Figure 10). In poppet valves the actuation is achieved by using a rocker arm and/or cam to depress the valve stem. A spring closes the valve. Mechanical actuation is reliable and easy to maintain,
The main challenge with this system is its additional energy losses due to numerous components that are necessary for operation. Its manufacturing cost are quiet high, and is limited in speed.

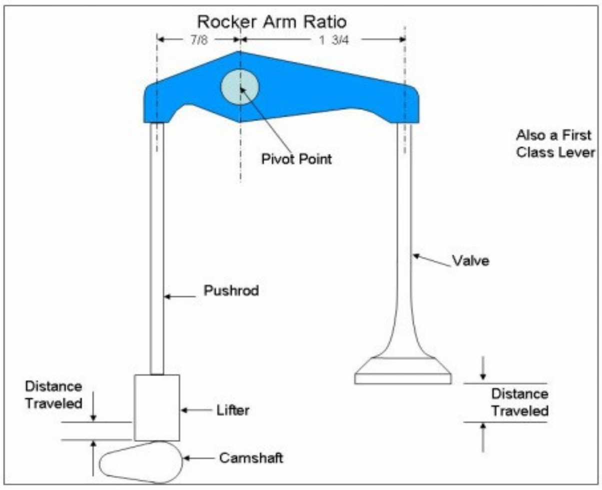

Figure 10: Cam and rocker arm actuation system. (Google/Images)

3.3. Rotary valve development

Since the invention of the IC engine, the rotary valve has proved to be promising in solving the challenges faced in using poppet valves. The main benefits being the absence of reciprocating components which turned out to reduce the level of vibrations and noise. As a result, a smooth running engine can be conceived with decreased pumping losses and a high overall engine output. 'This has inspired numerous inventors over the past century to chance their hand at developing Rotary Valve engines.' (T. Wallis 2007). However, the old concepts in Rotary valve technology have generally failed because of difficulties in gas and oil sealing, excessive friction and mechanical and thermal distortion of the valve ports. These challenges that engineers encountered by then could be solved using modern materials and tribological coatings, hence the 'old concepts can be revised utilizing today's technology in order to resolve the problems that occurred in the past.' (P. Ramy, D. Hoi, 2011). This has prompted modern inventors to revisit the rotary valve designs and came out with hybrid solutions to meet the challenges posed by rotary valves in the past century.
To mention but a few, the following inventions were developed recently and are of great interest to this project.

3.3.1 The Bishop Rotary valve

In 1997 Bishop started working with Ilmor engineering (later Mercedes-Ilmor) in developing their Rotary Valve technology for F1 engines. The Bishop Rotary Valve (BRV) is an axial flow rotating valve incorporating both the inlet and the outlet ports in the same valve. (T. Wallis 2007).
The valve is rotatable about an axis within the cylinder bore of a cylinder head. The peripheral opening of the inlet and exhaust is achieved by the communication between the bore and the cylinder. The axis of rotation of the valve is perpendicular to the cylinder axis and the flow into and out of the valve is approximately parallel to the valve axis. In the other version of the valve where one valve supplies many cylinders, the valve axis is parallel to the crankshaft axis. Depicted below is a cross-sectional view of the IC engine employing the rotary valve technology.

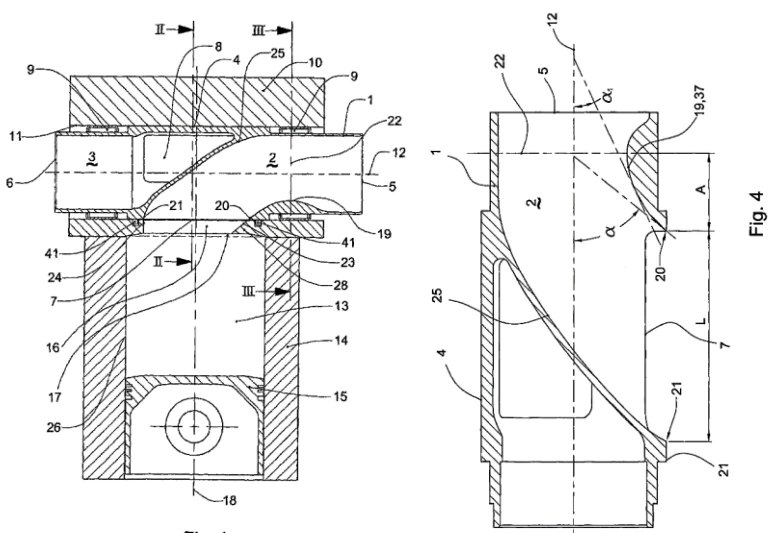

Fig. 1

Figure 11: An embodiment of the Bishop Rotary Valve Invention. (US patent #7,584,741, September 2009)

3.3.2 Lamberts' Rotary Valve

This Rotary valve assembly can be used in IC engines as well as in steam engines. The rotary valve assembly is responsible for controlling the flow of the working fluid (which could be an air/fuel mixture or steam) to and from the cylinder of an engine. The assembly is made up of a housing mounted on top of an engine of the piston and cylinder type. The working fluid is conducted to the cylinder and out of the cylinder via an intake and exhaust passages respectively. The rotating valve member has a window that opens and closes the intake and exhaust passages as the valve member is rotated. This opening and closing of the valve system is well timed and it happens periodically. The valve member in the form of a cylinder or disc is accommodated in the housing which has the intake and exhaust passages in it. The valve member is driven by an axle which is coupled for rotation with the engine crankshaft. (US patent #6,158,467A, Dec. 2000)

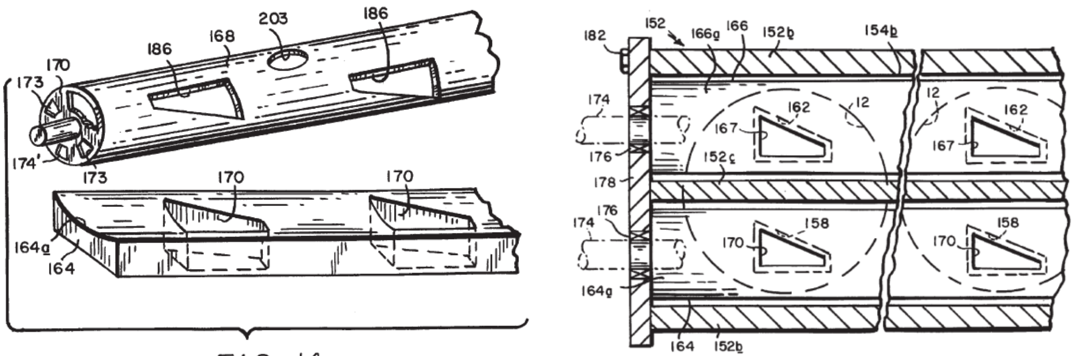

FIG. 14 FIG. 13

Figure 12: Part of the valve assembly, shown here is the valve member. (US patent #6,158,467A, Dec 2000)

This chapter focuses on developing and modelling the feasible solutions to the given problem. The above-mentioned patents are used as the foundation for generating the concepts.

The immediate goal of the overall project is to develop a prototype and test the functionality of the whole engine. A single-cylinder engine with a crankshaft perpendicular to the axis of the cylinder is to be used for testing. The concepts developed here are to be applied both on the single-cylinder engine and the five-cylinder axial piston expander, with some minor modifications on the designs. In the case of a single-cylinder engine, the valve rotational axis is perpendicular to the cylinder axis and parallel to the crankshaft, whereas in the case of the multi-cylinder engine, the valve is parallel to the axis of the cylinder. In the case of a one-cylinder engine, the valve mechanism is driven by the engine's crankshaft via a timing belt and pulley transmission. On the multi-cylinder engine, the valve shaft is coupled directly to the engine's z-shaft, hence there is no transmission required in this case. Much emphasis was put on the one-cylinder engine which is the immediate goal of the overall project, but the overall concept of the five-cylinder engine layout was kept in mind. Figure13 depicts the application of the rotary valve on the multi cylinder and the single cylinder engine. This project focuses on the case of a one-cylinder engine.

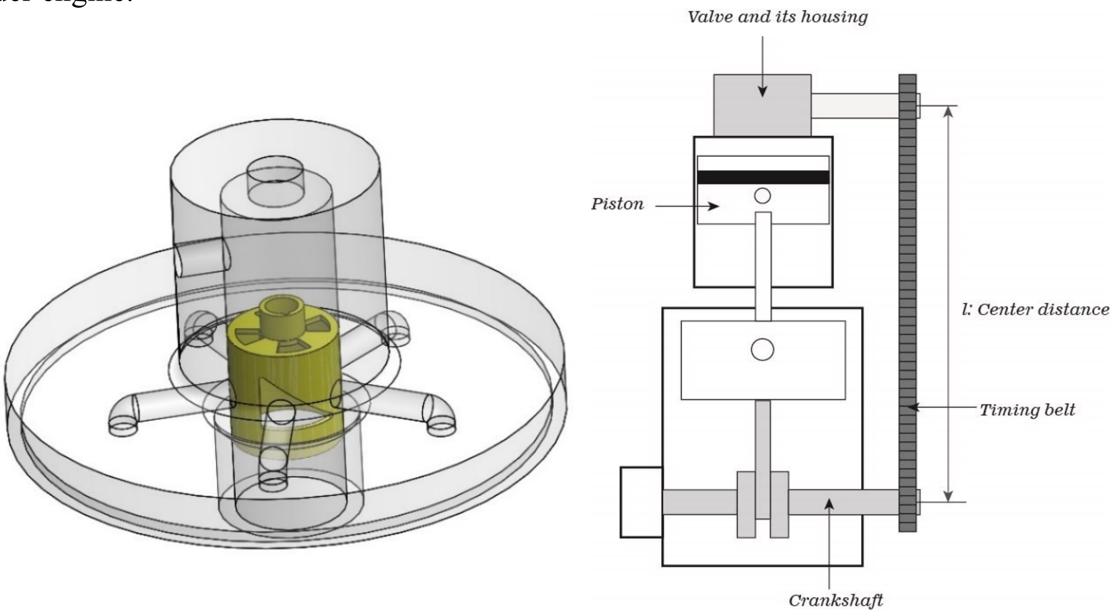

Figure 13: Illustration of the rotating valve in a multi-cylinder engine (left) and on a single-cylinder engine(right)

4.1. Product requirements specification

Table 1 gives the requirements that the solution should meet. The case of a single cylinder engineer is the main focus of this project. On top of the tabulated requirements, the valve should operate in an oil-free environment and should be able to provide a variable cut-off time.

Table 1: Project Requirements

Condition	Specification	
	Multi-cylinder engine	Single cylinder prototype
Temperature *(°C)*	450	450
Pressure *(bar)*	250	50
Cut-off time *(ms)*	0.4	5
Engine speed *(rpm)*	6000	3000
Outer diameter *(mm)*	40	40
Cylinder displacement *(cm^3)*	250	50.26

Timing characteristics of the expander

Figure 14 illustrates the general timing characteristics of the piston expander.

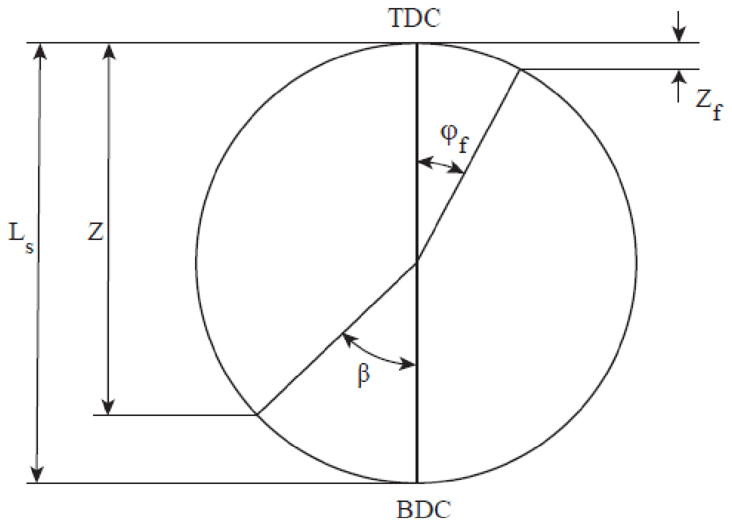

Figure 14: General timing diagram for the expander

With the following designation;

TDC: *Top Dead Center*; BDC: *Bottom Dead center*; L_s: *stroke length*;
Z: *instant stroke length*; β: *crankangle*; φ_f: *the filling angle from TDC*

$$Z_f: opening\ length;\ \ F = \frac{Z_f}{L_s}: filling\ ratio$$

Table 2: Timing parameters

Parameter	Specification			
Speed (rpm)	6000		3000	
F (%)	1,7	50	1,7	50
$\boldsymbol{\varphi_f}$	15°	90°	15°	90°
Time for one revolution (*ms*)	10	10	20	20
Cut-off time (*ms*)	0,4	2,5	0,8	5

Table 2 gives the general timing parameters of the engine. For this project a cut-off time of 5ms, filling angle of 90°, filling ratio of 50 % and a speed of 3000 rpm were the targets.

4.2. Description of concepts

This section describes the working principle of the developed valve mechanisms.

Concept 1: Discoid Rotary Valve

This valve mechanism (represented in Figure 15) is composed of the valve member (5) which is in form of a disc with an opening port (10) (as indicated in Figure 16) that allows the passage of pressurized steam from the inlet to the outlet passage of the valve block. The valve housing is made up of two blocks which is to be positioned on top of the engine cylinder. The upper block (7) of the valve assembly has an inlet passage which is connected to the steam generator. The lower block (1) of the valve assembly has an outlet passage which conducts steam to the cylinder of the engine. The valve member (5) which is in form of a disc is keyed to a valve shaft (3) and is locked axially by circlips (4). The shaft is supported by two bearings (2), and on one of its end, a pulley (8) is keyed to the shaft. The pulley is connected to the crankshaft of the engine via a timing belt (not represented in the figure), thus the valve shaft is driven by the crankshaft via this transmission.

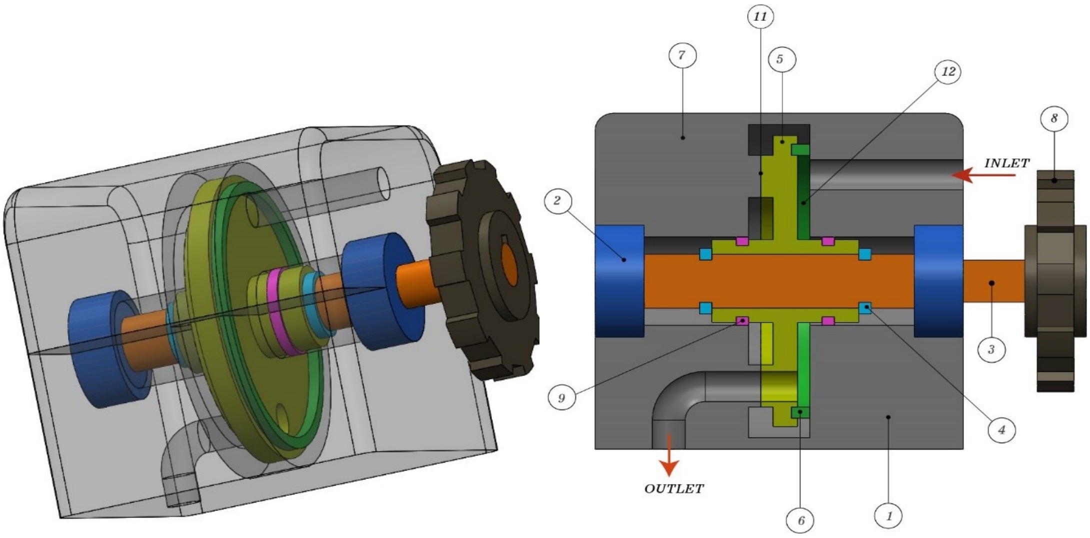

Figure 15: Representation of the Discoid valve assembly

The valve shaft (3) rotates with the valve member (5), and the opening port (10) on the valve member periodically opens and closes the outlet passage, thus the flow of steam to the cylinder is regulated this way. The size and shape of the port on the valve member can come in different variations. Sliding seals (9) are used to stop pressurized steam from escaping to unwanted areas. The sliding seal (6) on the valve member is used to trap pressurized steam in a compartment (12), in such a way that this compartment is always filled with steam which is periodically communicated to the engine cylinder via the outlet passage. The leakages between the valve member and the outlet passage are minimized by the sealing effect provided by the force acting on the valve member due to the pressurized steam. The frictional forces between the contact surfaces (11) of the valve member and the two blocks can be reduced by coating the protruded parts on the two blocks and the protruded part on the valve member (13, represented in Figure 16). The coatings can be made out of graphite which can operate well as a lubricant in an oxidizing environment and also at high temperatures of 450° C. (A, van Beek) The reason of having protruded surfaces on both the valve member and the two blocks (11) is to minimize the area to be coated.

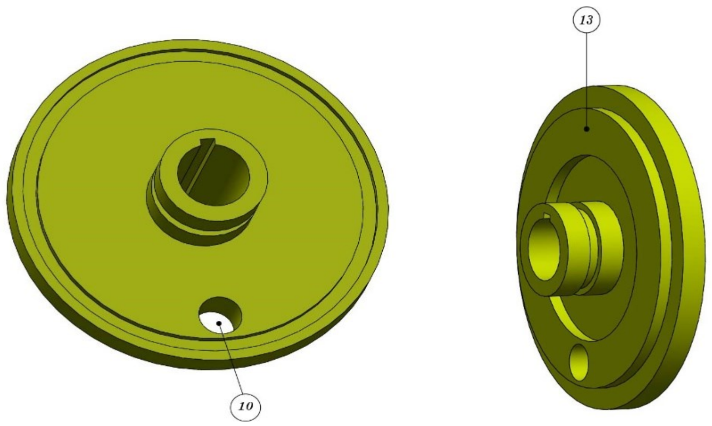

Figure 16: Valve member

Advantages

- Compact and simple design i.e. few parts
- Can be easily adopted to a five-cylinder engine
- Ease of manufacture i.e. no complex geometry hence parts can be mass-produced

Disadvantages

- Low reliability; likely to have higher leakages
- Can only provide constant cut-off time

Concept 2: Poppet valve

This valve mechanism (represented in Figure 17) is a typical conventional poppet valve which is comprised of the poppet valve (2), the tappet (7), valve keepers (5), spring retainer (6), washer (3), compression spring (4), and the camshaft (10). The valve mechanism is housed in the valve block (1) which can be positioned on top of the engine cylinder. The camshaft is supported by two bearings (9) which are housed in the bearing brackets (8). A pulley (11) is keyed to one end of the camshaft and is connected to the engine's crankshaft via a timing belt. The valve is opened by the camshaft and is closed by the spring. The valve keepers have a conical shape which allows them to be wedged into the spring retainer. The spring is supported by a washer which sits on a seating machined in the valve block.
In the case of a five-cylinder piston expander, the poppet valves are disposed radially and each cylinder has its own poppet valve which is opened by the camshaft and is closed by the spring. (as represented in Figure 18)
The spring is preloaded so that it exerts a force on the poppet valve at its rest position. The force exerted by the spring is acting on the valve stem because of the valve retainer which is attached to the valve by the valve keeper. The valve keepers are wedged to the valve retainer because of the conical mating between these two parts. The tappet connects the camshaft and the valve, and it transfers the motion of the cams to the valve thus the valve is opened by the cam and closed with the aid of the spring.

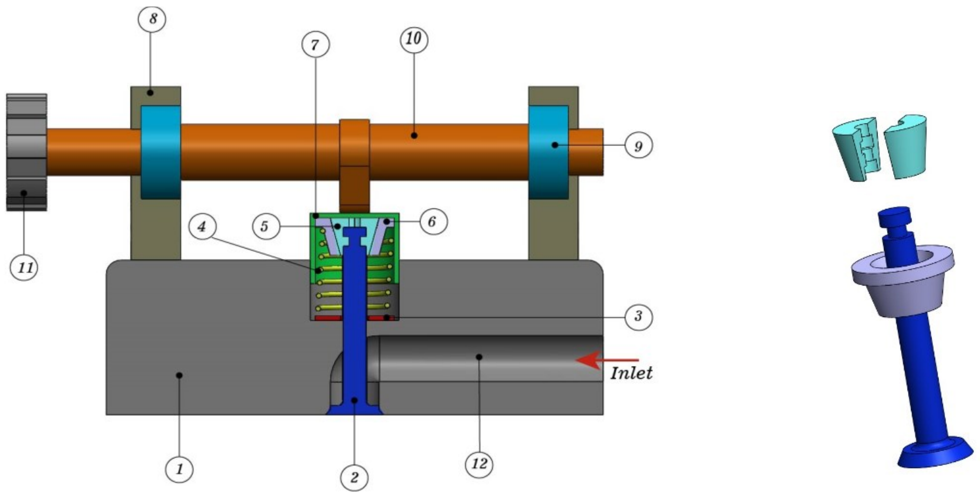

Figure 17: Poppet valve on a single cylinder engine (left) and layout of the valve components (right)

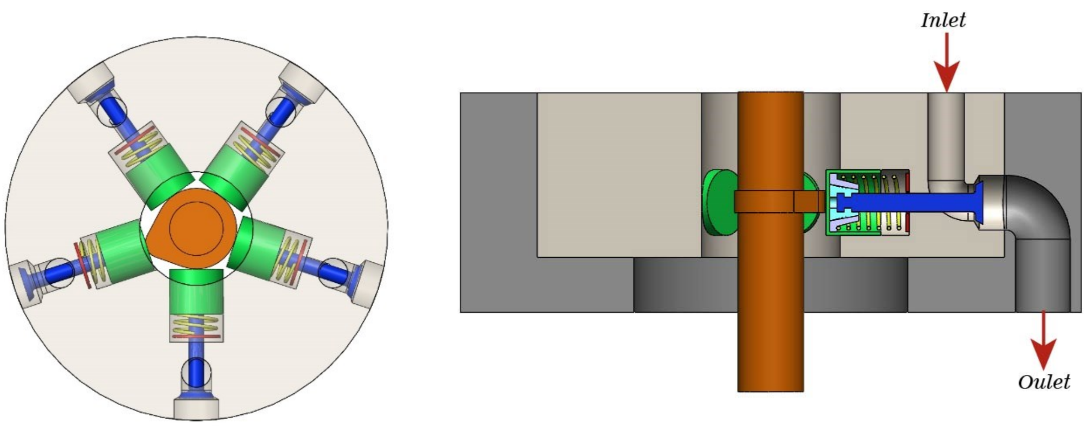

Figure 18: Layout of poppet valves on a multi-cylinder engine

Advantages

- Provides tight sealing; virtually no leakages
- Simple system
- Minimal frictional losses
- Easy to control

Disadvantages

- Noisy operation
- Valve floating at high rpm
- Many components are required in the case of radial poppet valve system
- Fluid flow is obstructed by the presence of the poppet valve

Concept 3: Cylindrical Rotary valve

Note: The details of the working principle of this design could not be revealed due to confidentiality issues, as a result the presented information has been filtered. See Lambert's rotary valve (Section 3.3.2) for basic details on which this design was built upon.

This embodiment (as represented in Figure 20) consists of a cylindrical valve member represented in Fig 19 which is keyed to a rotating shaft (2) in the mechanism. The valve member has inlet ports (22) which allows steam from the inlet passage of the block to pass through it and communicated to the cylinder via the circumferential outlet port on the valve member (23). As the valve member rotates around its axis, the outlet port (23) alternately communicates with the outlet passage of the valve block and the valve is opened and closed periodically. One of the shafts, (the motor shaft (2)) is connected to a pulley (3) via a key and a keyway and the pulley is connected to the crankshaft via a timing belt (not shown in the figure).

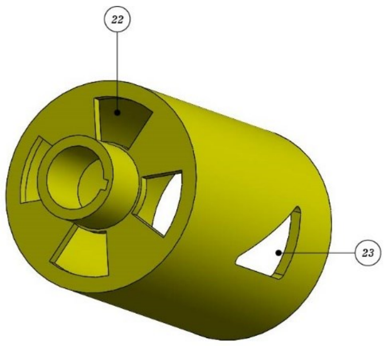

Figure 19: Illustration of the valve body for concept 3

The pulley is locked axially by a locknut (5) and lock washer (4) on one side, and on the other side it is locked by a shoulder on the shaft. The motor shaft (2) is supported by rolling bearings, which are housed in the lid of the valve block (1). The lead is mounted to the valve block by screws.

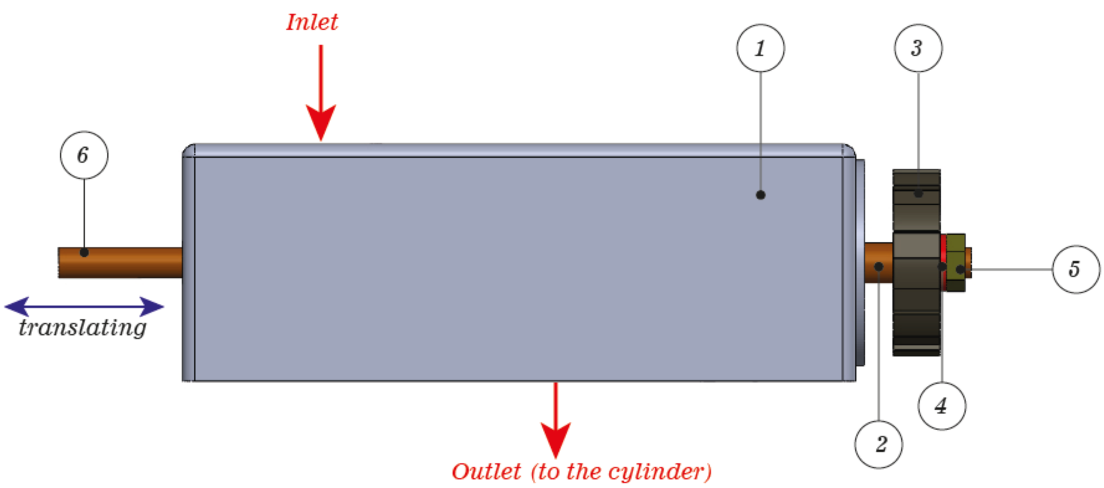

Figure 20: Illustration of Concept 3: The Rotary Valve

The other shaft (6) is connected to an actuator (not shown here) and can only translate without rotating. It is supported by sliding bearings. The actuator shaft (6) is connected to the motor shaft in such a way that the motor shaft can rotate without causing rotational movement to the actuator shaft.

Whilst the actuator shaft can translate, it cannot cause the motor shaft to translate. The sliding seals help to minimize leakages of the pressurized steam. The valve lining is coated inside with graphite as this can act as a lubricant and can minimize frictional forces. The lining is screwed to the housing to prevent it from rotating and cause misalignment of the inlet and outlet passages.

As the actuator shaft can translate along its axis, it causes the valve member to translate (axial shifting) with it thus the cut-off time is varied this way as the amount of steam to enter the cylinder is regulated by moving the actuator shaft. The axial shifting of the valve member (which is caused by the actuator), will change the relative position of the valve thus the valve mechanism can provide the throttling effect this way.

Advantages

- Allows variable cut-off time
- Individual parts can be mass produced i.e. no complex geometry involved
- Compact and lightweight design
- Reliable at high rpm
- Can be easily adapted to a five-cylinder piston expander

Disadvantages

- Great demands on sealing
- Mechanical and thermal distortion of the valve member

4.3. Concept selection

To choose the best and optimal solution, out of the developed concepts, the Pugh matrix was used for selection. Comparisons of the three concepts was made in order to determine the best candidate that meet a set of criteria described below. The final selection was made taking multiple interlinked factors into account. The Pugh matrix was used to make an analysis on the design alternatives so that a robust decision could be established. The Pugh decision matrix was used to rank the concepts to choose the most optimal and best suitable concept.
The three designs have been evaluated against a number of criteria as described below. A linear 5-point scale was used as shown below. The scale is symmetrical about the origin (0), and linear. The weighted score is simply the product of the rating and the weighting. The weighted scores are then summed and the concept with the highest score is selected:

+2: much better

+1: better

0: satisfactory

-1: worse

-2: much worse

The highest ranked score was selected as the winning candidate.

Selection criteria

The following factors were used to determine the best design solution among the three solutions. Lambert's rotary valve was used as a reference.

Reliability: at high rpm reliability of the mechanism becomes an issue as some valve mechanism 'float' and do not close and open as they are supposed to do
Manufacturability: due to complex geometries and tight tolerances, some parts are uneconomical to manufacture in a mass production environment.

Sealing: one of the most important factor in valve design is to have minimal leakages at low cost. Leakages determine how effective a valve can be.
Variable cut-off time: this was one of the important factors to consider as it has a greater impact on the operation of the engine as a whole. A constant cut-off timing can be used for prototyping but a variable cut-off timing is crucial in a multi-cylinder engine.
The number of parts: the number of parts required to mount the valve assembly was also considered as this could have an impact on the overall cost and compactness of the valve mechanism.
Frictional losses: high forces due to friction result in higher power demands and uneconomical actuation energy.
Cost: the best design is the one that fulfils the requirements at a low cost. The cost of mass-producing the parts was also taken into account.
Table 3 illustrates how the concepts have been evaluated and ranked.

Table 3: Decision Matrix for Concept Selection

			Design concept			
		Weighting	**Lambert's Rotary Valve**	**Concept 1**	**Concept 2**	**Concept 3**
Selection Criteria	Reliability at high rpm	5	0	+1	0	+2
	Manufacturability	4	0	+2	+2	+2
	Sealing	5	0	0	+2	+1
	Variable opening time	5	0	-2	-2	+2
	Number of component	1	0	+2	0	+1
	Frictional forces	3	0	-2	+1	-2
	Cost	2	0	+1	+1	+1
	Total score	50	0	1	13	30

From the Pugh concept selection matrix, it can be noted that the cylindrical rotary valve (concept 3) has emerged as the winning candidate as it has the highest ranked score.

4.4. Material selection

The selection of material for a machine component is one of the most important decision that will affect the functionality and reliability of the entire mechanism. Each component was assigned a specific material basing on the operating conditions and the loads acting on it. The software CES Edupack was implemented in the process of choosing material. The operating conditions for the valve are severe on account of high temperatures and high pressure.
The valve assembly is subjected to thermal stresses in both the circumferential and longitudinal directions, higher temperatures, and severe oxidizing conditions. Basing on the operating conditions mentioned in the specification, the following factors influenced much on the decision made on the material assigned to each component.

- High strength and hardness to resist wear and loads
- High fatigue and creep resistance
- Resistance to corrosion
- Least coefficient of thermal expansion
- High thermal conductivity, for better heat dissipation

- Least coefficient of friction
- Manufacturability
- Cost

Based on these factors, each component was assigned a specific material in order to resist the severe operating conditions of high pressure and temperatures. Superheated steam is known to be very aggressive in terms of its strong oxidizing power, hence the environment is very oxidizing and corrosive. Due to high temperatures and pressure, the components are most likely to deform and the thermal effects could be of greater magnitude.
The software Ansys was used to determine the thermal expansion of the valve body which is the most critical part of the mechanism. A simulation was run to determine the deformation of each of the potential materials and the results are presented in the following sections. The simulation considered a pressure of 50 bars and a temperature of 450° C. The goal for simulating the thermal expansion of each material was to determine the clearance between the valve body and its casing. The clearance will avoid the valve body from getting stuck inside its housing.
The following group of materials were potential candidates that could meet the requirements mentioned above.

Titanium Alloys

Titanium alloys can be used at maximum service temperatures ranging from 450-500 °C. They also have unusually low thermal conductivity and expansion coefficient. The alloy Ti-6Al-4V is a typical example and a potential candidate, often used in valve bodies and compressor blades of aircraft turbines. Titanium alloys have extraordinary corrosion resistance, have very high tensile toughness and toughness even at extreme temperatures (CES Edupack). Figure 21 shows the results of the deformations of titanium and a maximum deformation of 0.93 mm was observed.

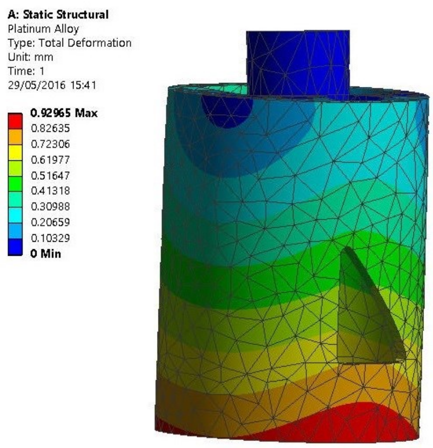

Figure 21: Ansys results of static deformation of Titanium alloy

Alumina

Alumina is an advanced technical ceramic that is cheap and easy to process. It has a broad set of properties which include high mechanical strength, temperature resistance of up to 1650°C, excellent chemical stability, high thermal conductivity and corrosion resistance. It is often used for high temperature components, wear resistant components, in plain bearings, mechanical seals and spark plug insulators (Plansee). The following figure illustrates the deformation of alumina in the given operating temperature and pressure. A maximum deformation of 0.23 mm was observed.

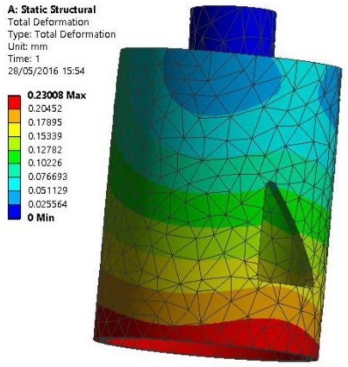

Figure 22: Ansys results for static deformation of Alumina

Chromel

This is a nickel-chromium alloy that can be used at temperatures as high as 1200 °C. It can resist corrosion and oxidation at high temperatures (CES Edupack). Figure 23 represents the results from Ansys simulation and a maximum deformation of 0.52 mm was observed.

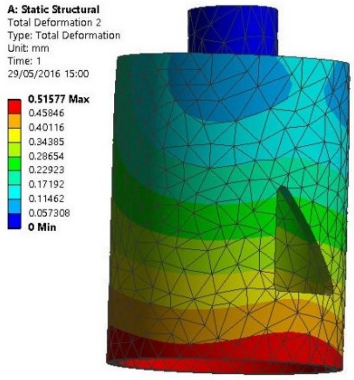

Figure 23: Ansys results for static deformation of Chromel

Molybdenum Alloys

Molybdenum alloys have the ability to withstand severe temperatures without considerable deformation, which makes them useful in applications that involve intense heating. Typical molybdenum alloys are TZM and MHC, and they can resist corrosion up to temperatures of 760 °C with the help of a Sibor® protective layer. These alloys possess very high melting points, and a low coefficient of thermal expansion (Plansee). Figure 2 represents the results from Ansys simulation and a maximum deformation of 0.27 mm was observed.

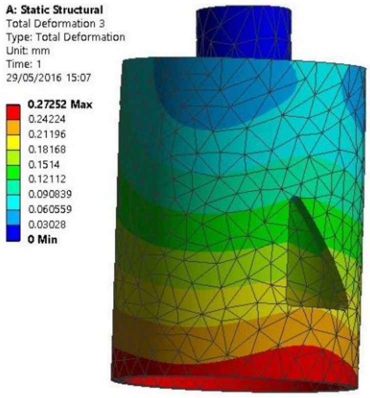

Figure 24: Ansys results for thermal deformation of Molybdenum alloy

Tungsten Alloys

Tungsten alloys are characterized by low coefficient of thermal expansion therefore suitable for very high temperature applications. Tungsten alloys have extraordinary corrosion resistance and high tensile strength and toughness even at elevated temperatures. However, these alloys are relatively very expensive and are difficult to process, which limit their use in general applications (Plansee). From Figure 25, a maximum deformation of 0.23 was noticed for Tungsten alloys.

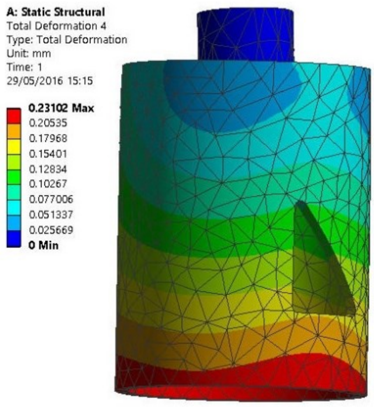

Figure 25: Ansys results for static deformation of Tungsten alloy

Inconel

This is a nickel based high temperature alloy, that can resist corrosion and oxidation due to high level of Chromium and Molybdenum. Its melting temperature (ranging from 1370-1450 ° C) makes it a good candidate well suited for services in extreme temperatures and pressure. However, it is a difficult metal to machine using conventional methods (Plansee). Figure 26 gives the Ansys results of the deformation of Inconel. A maximum deformation of 0.44 mm is noticed.

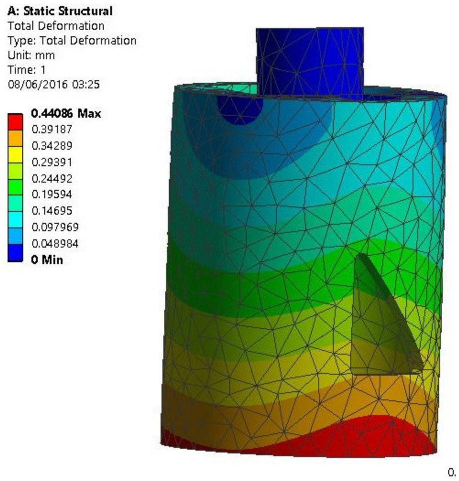

Figure 26: Ansys Results for thermal deformation of Inconel

Decision Matrix

Since there is a good number of potential materials that can be used in the said operating environment, it is difficult to make the best selection given that there are interlinked factors that should be accounted for. A decision matrix was established to make the most robust decision. A selection criteria was established and a set of factors was considered.
A linear 5-point scale was used as shown below. The scale is symmetrical about the origin (0), and linear. The weighted score is simply the product of the rating and the weighting. The weighted score is then summed and the material with the highest score is selected:

+2: much better

+1: better

0: satisfactory

-1: worse

-2: much worse

Titanium Alloy was used as the reference

Table 4: Decision Matrix for Material Selection

			Material					
		Weighting	Titanium Alloys	Alumina	Chromel	Molybdenum Alloys	Tungsten Alloys	Inconel
Selection Criteria	Corrosion resistance	5	0	+2	+1	+1	+2	+1
	Manufacturability	4	0	-1	0	+1	-2	-1
	Cost	4	0	+1	0	-1	-2	0
	Wear resistance	3	0	+1	0	+1	+2	+2
	Thermal expansion	5	0	+2	+1	+2	+2	+1
	Total score	42	0	23	10	13	10	12

From the Decision Matrix, the winning candidate emerged as Alumina followed by Molybdenum alloys. Basing on these results, each component of the mechanism was assigned a specific material as indicated in BOM (not presented for confidentiality issues)

4.5. Concept Analysis

The analysis and modelling of the chosen concept was carried out in order to verify the feasibility of the concept. The concept modelling serves as the 'proof of concept' as it demonstrates the potential of the concept at least in principle.

The technical feasibility of the concept was analytically validated and potential 'stumbling blocks' were identified as well as the other work necessary to put the concept into real world application.

The scope of the analysis was focusing on the kinematics and feasibility of the system. The following sections gives the calculations and the analysis carried out for individual components and for the assembly as a whole. Detailed thermodynamics and computational fluid dynamics was reserved for future work.

4.5.1 Preliminary flow analysis

Preliminary calculations on the dynamics of steam have been carried out with the goal of determining the area of the outlet port of the valve which is required to give the maximum power targeted. For simplicity of calculations the flow was considered incompressible, meaning the pressure was assumed to be constant. However, for future work, detailed Computational Fluid Mechanics methods should be employed were all variable parameters like temperature and pressure will be taken into account.

Volumetric flow rate

The volumetric flow rate (*Q*), (also known as the volume flow rate, rate of fluid flow or volume velocity), is the volume of fluid which passes per unit time. It is calculated using equation (1)

$$Q = \frac{V}{t} = v.A = \frac{\dot{m}}{\rho} \tag{1}$$

Where *V*: volume, *v*: velocity, *t*: time, *A*: cross sectional area, ρ: density, $\dot{m}$: mass flow rate

The volumetric flow rate was determined from the mass flow rate formula:

Knowing that the peak power aimed for is *30 kW*. The mass flow rate can be determined from equation (2):

$$P = \dot{m} * \Delta h \tag{2}$$

With *P* as the power (*kW*), and *Δh* is the change in enthalpy (*kJ/kg*), $\dot{m}$: is the mass flow rate (kg/s).

The conditions for both the condenser and the admission were determined from steam tables (Spirax Sarco).

For the admission conditions (steam admitted to the engine), the temperature and pressure were set at 450°C and 50 bars. For condenser conditions (steam vented out of the engine), the pressure was set at *1 bar* and the required corresponding temperature was set at 100 *°C*. For an isotropic expansion, the entropy is the same for both the admission and the condenser, hence the entropy found using admission conditions was used to determine the enthalpy for the condenser conditions. The admission enthalpy and the condenser enthalpy were used to determine Δh

Table 5: Data from Steam tables

Variable	Admission	Condenser
Temperature *(°C)*	450	100
Pressure *(bar)*	50	1.0
Enthalpy *(kJ/kg)*	3314.9	2476.6
Entropy *(kJ/kg K)*	6.808	6.808
Density *(kg/m^3)*	16.14	0.655
Δh *(kJ/kg)*	838	

Table 5 gives the data for the admission and condenser condition and the change in enthalpy. Using the above data, the mass flow rate and the volumetric flow rate were found to be:
$\dot{m}$=0,0358 kg/s and Q=0.002218 m^3/s=2218 cm^3/s
The corresponding volume of steam was calculated using equation (3), knowing that the cut-off time targeted is 5 ms;

$$Q = \frac{V}{t} \tag{3}$$

With *V* as the volume, which is equal to 11.09 cm^3. The engine to be used is of 40/40 mm bore/stroke which correspond to an engine displacement of 50.2 cm^3.
The maximum targeted speed for the prototype is 3000 rpm which means one revolution takes 0.02s.

4.5.2 Dimensioning of the valve outlet port

In order to reach the targeted peak power, the valve outlet port (23) was dimensioned to ensure that the required volume enters the cylinder within the given cut-off time.

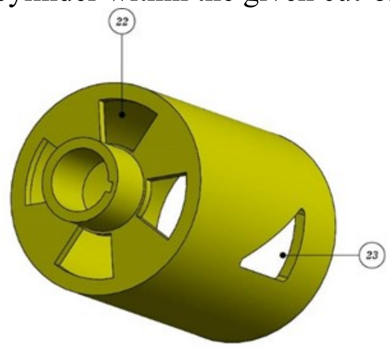

Figure 27: Illustration of the valve outlet port

The goal of the prototype is to have a maximum speed of 3000 rpm at a valve cut-off time of 5 ms. The corresponding time for one revolution at this targeted speed is 0.02s, which implies that the filling angle (φ) required is given by equation (4):

$$\varphi = \frac{t_c}{t_{rev}} * 360 \tag{4}$$

Where t_c and t_{rev} are the cut-off time and the time for one revolution respectively.
This gives a filling angle of 90°.
Given the filling angle, the circumferential length *(s)* of the valve outlet port can be determined using equation (5):

$$s = r\theta \tag{5}$$

Where θ is the filling angle in radians and r is the radius of the valve body.
This gives the circumferential port length of 31.41 mm which is equivalent to a chord length *(b)* of 28.28 mm

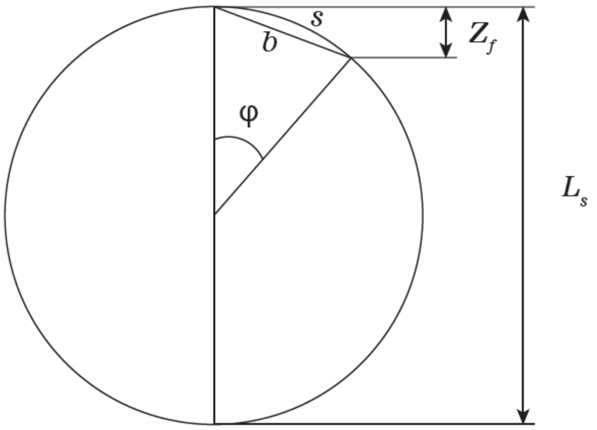

Figure 28: Illustration of the geometry

The filling ratio (F) is given equation (6);

$$F = \frac{Z_f}{L_s} * 100 = \frac{r - r * cos\varphi}{L_s} * 100 \tag{6}$$

With L_s as the stroke length which is equal to 40mm.
This gives a value of 50 %. Figure 28 illustrates how the filling angle varies with the filling ratio. It is also indicated in the graph that a filling angle of 90° corresponds to a filling ratio of 50 % which agrees with the specifications given. The opening and closing effects from the rotational motion of the shaft are timed out precisely to work in conjunction with the gas flow requirements of the engine.

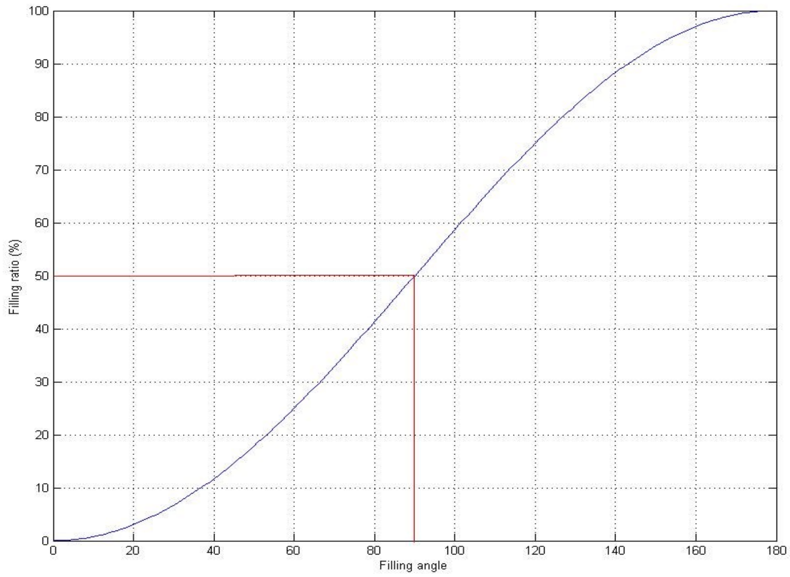

Figure 29: The graph of filling ratio as a function of the filling angle

The filling ratio was also plotted as a function of the filling angle as illustrated in Fig 29. It can be noted that the rotational speed is inversely proportional to the cut-off time. A filling ratio of 50 % correspond to a valve cut-off time of 5 ms. Figure 30 illustrates the variation of filling ratio with cut-off time for speed of 3000 and 6000 rpm.

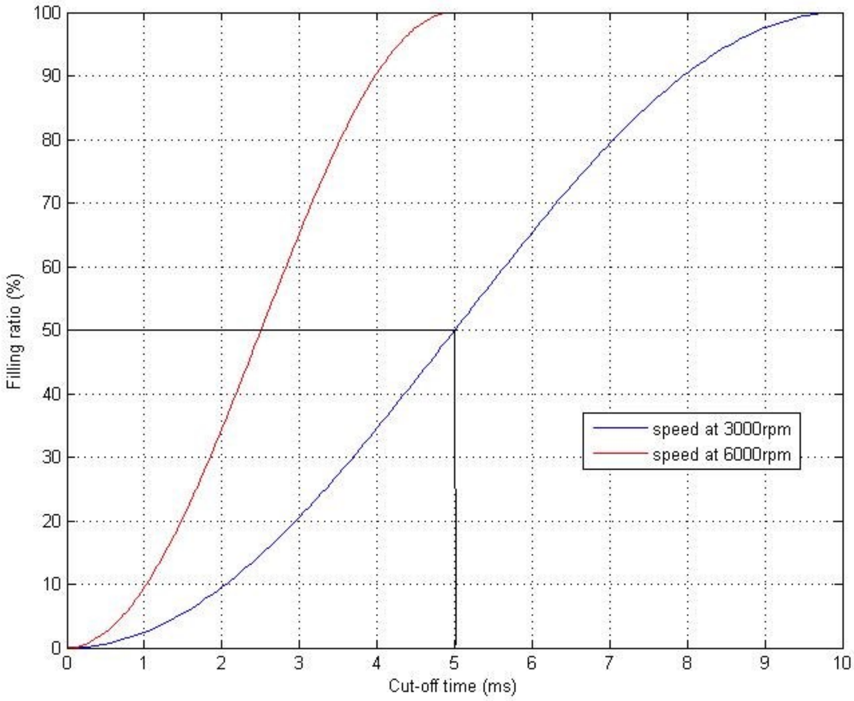

Figure 30: The graph of Filling ratio as a function of Cut-off time

In order to calculate the flow area of the outlet port, the velocity (v) of the steam is required. Equation (7) was used to calculate steam velocity;

$$Q = A * v \tag{7}$$

Where A is the flow area and v is the velocity of steam.
From an existing valve from the company, Ranotor, the flow area was measured to be 1.495 cm^2 which correspond to steam velocity of 1483.61 cm/s. The area in the design was doubled, which implies that the speed is halved to 741.81 cm/s. This gives an area of 2.99cm^2 and a length of 2.11cm for the port of the valve
Hence according to the above calculations, the valve outlet port should have dimensions of 21.1*28.3 *mm (h*b).*

4.5.3 Variable cut-off timing

As one of the requirements, the valve mechanism should be able to provide variable cut-off timing. By appropriately selecting the size and shape of the opening valve port, the cut-off timing can be controlled as well as the volume of steam injected into the cylinder. The triangular geometry chosen for the valve outlet port allow variable timing. As described in Section 4.2.3 an actuator is required to shift the valve body axially. The axial shifting of the valve body will change the relative position of the outlet valve port which thus allows the opening time to be varied. This mechanism serves as the throttle for the engine.
The way in which the cut-off timing varies with the axial displacement was analyzed and is illustrated in the following figure. Figure 31 is a graph of the filling ratio with the varying cut-off timing and it conforms to the graph represented in Figure 29

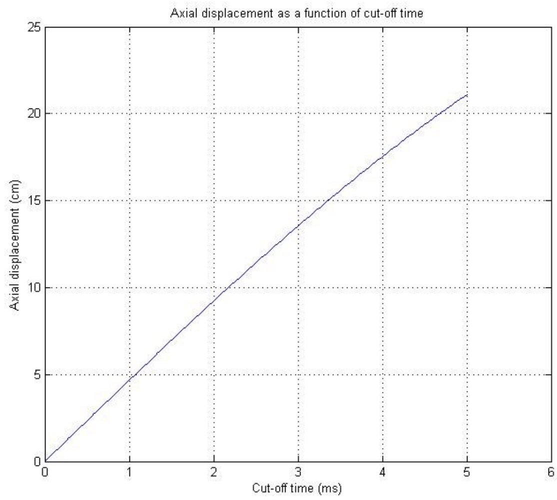

Figure 31: Graph of valve axial displacement in function of cut-off time

The variation of the filling ration with cut-off timing is illustrated in Figure 32.

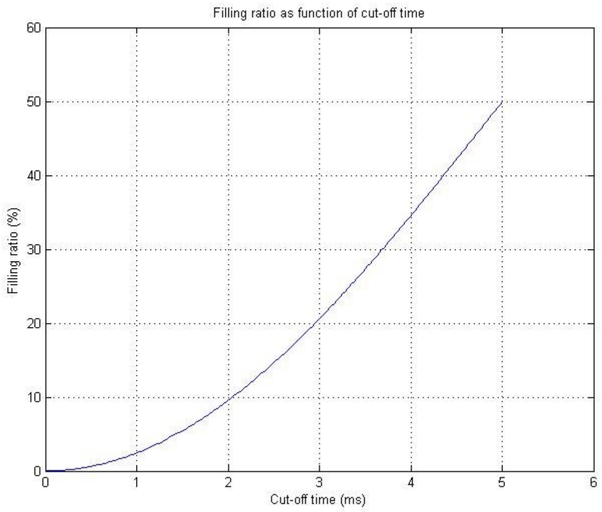

Figure 32: Variation of filling ratio with cut-off time

4.5.4 Dimensioning of the valve body

The valve body is subjected to high pressure, hence determination of its thickness is crucial. The thickness was calculated to ensure that intense stresses induced by pressurized steam can be supported without causing considerable deformation of the valve body. As the valve body is subjected to internal pressure, the most significant stress is the tensile stress acting in a direction tangential to the circumference, and this is known as circumferential or hoop stresses.

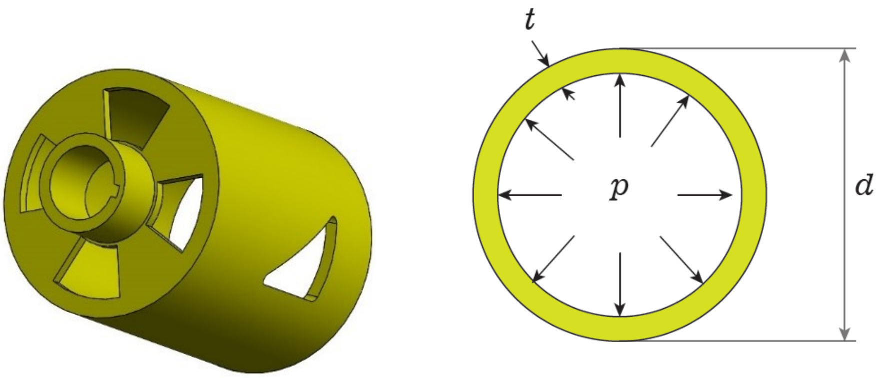

Figure 33: Valve body and its cross section

The thickness of the valve body (illustrated in Figure 33) is determined using Hoop's stress formula for pressure vessels as indicated in equation (8),

$$t = \frac{p * d}{2 * \sigma} \tag{8}$$

Where *t:* thickness, *d:* diameter, σ: hoop stress, and *p*: pressure
According to the results from material selection, Alumina is the best candidate hence a hoop stress of 380 MPa was considered at 450 °C which gives a thickness of 0.29 mm.
A thickness of 2 mm was assigned to the valve body in order to reduce the thermal deformation.

4.5.5 Dimensioning of the transmission system

The rotating valve on a single cylinder engine is driven by the crankshaft as illustrated in *Figure 34*. To ensure a smooth operation of the valve mechanism, the transmission system was dimensioned according to the given requirements of 30kW power and a speed of 1500-3000 rpm. The main purpose of the transmission system is to drive the rotating valve at the same speed as the crankshaft. Timing belts were chosen over chains because of their following advantages:

- Damps and isolates noise and vibrations
- Need no lubrication and minimal maintenance
- Timing belts run much faster than chains

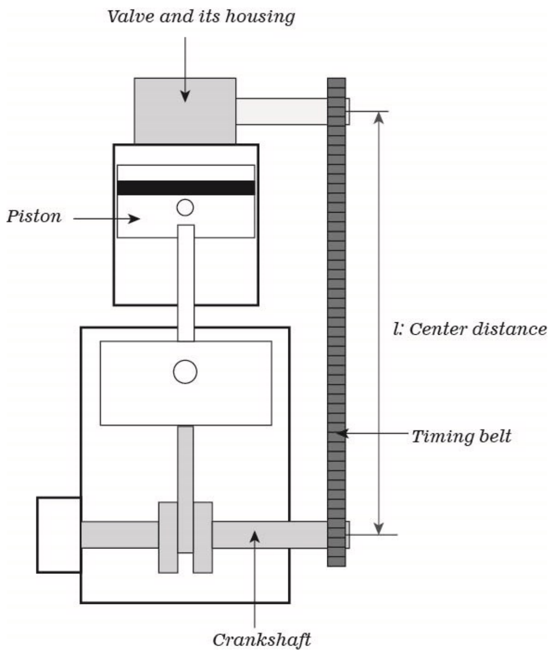

Figure 34: Illustration of the transmission system on a single cylinder engine

The dimensioning of the transmission system was carried out as recommended by SKF which is a world leader in the design and manufacture of transmission belts. (SKF Power Transmission belts). Since the targeted maximum power (30kW) is quite considerable, the drive conditions can be considered as heavy duty, considering the electric motor to be used as soft start, a service factor of 1.7 was selected;

$$C_2 = 1.7$$

The design power (P_d) was calculated as follows:

$$P_d = P_r * C_2 = 51kW \tag{9}$$

With P_r as the drive power.
The required speed ratio (I_r) was set at 1 since the two pulleys are rotating at the same speed.

$$I_r = 1$$

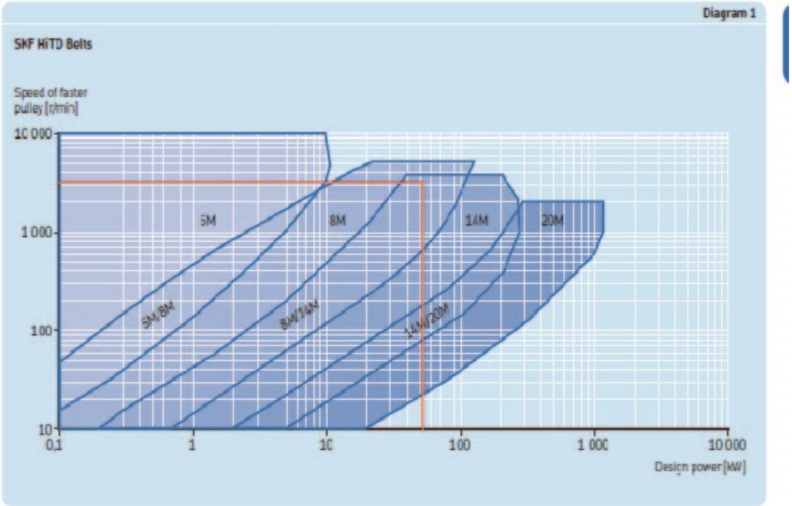

Figure 35: SKF Belt cross section diagram. (SKF Group)

According to SKF belt section diagram (Figure 35) based on speed and design power, 14M section was selected for the given drive parameters for heavy duty.

The following corresponding parameters for section *14M* were selected from SKF tables:

- Speed ratio (I_r): 1
- Belt length: 1610
- Centre distance (Cd): 609.0
- Number of teeth on pulley *(Z):* 28
- Pulley pitch diameter *(D):* 124.78

Power correction factors (C_1, C_4)

From the given Correction factor tables, C_1 is not given for the chosen diameter. To determine C_4, the number of teeth in mesh *(TIM)* is required and is calculated using equation (10):

$$TIM = \frac{0.5 - (D - d)}{6 * C_d} * Z_s = 3.8 \tag{10}$$

With D: large pulley diameter, d: small pulley diameter, C_d: Design center distance, Z_s: number of teeth on small pulley.
From the Correction factor tables (Figure 36), the corresponding value of C_4 is 0.6

Table 6

Teeth in mesh (TIM) Correction factor C_4

TIM	C4
–	
6 and above	1
5	0,8
4	0,6
3	0,4
2	0,2

Table 7

Belt length correction factor C_1

Belt length	Correction factor 5M	8M	14M	XL L H XH T2.5 T5 T10 AT5 AT10
mm	–			
425	0,80	–	–	1,00
535	0,90	–	–	1,00
600	1,00	–	–	1,00
800	1,10	–	–	1,00
890	1,20	–	–	1,00
1 050	1,30	–	–	1,00
1 190	–	0,80	–	1,00
1 200	–	0,90	–	1,00
1 420	–	1,00	–	1,00
1 610	–	1,10	–	1,00
1 760	–	1,20	–	1,00
1 890	–	–	–	1,00
2 000	–	–	0,90	1,00
2 450	–	–	0,95	1,00
2 500	–	–	1,00	1,00
3 150	–	–	1,05	1,00
3 400	–	–	1,10	1,00

Figure 36: SKF Correction factor table. (SKF Group)

Corrected belt power rating and respective belt width (P_{corr}, W_b)

The corrected power rating is determined by equation (11) with the basic power rating (P_b, given in power rating tables) multiplied by power correction factors C_1, C_4. P_b is equal to 31.21 kW according to the given power basic tables.

$$P_{corr} = P_b * C_1 * C_4 = 18.72 \tag{11}$$

The corrected power rating must be equal or greater than the design power P_d (equation 12)

$$P_b * C_1 * C_4 \geq P_d \tag{12}$$

The minimum required basic power rating is calculated by the following equation:

$$P_b = \frac{P_d}{C_4 C_1} \tag{13}$$

$$P_b = 85\ kW$$

Remarque: The calculated corrected power rating is less than the design power meaning the above condition is not fulfilled. However, after careful considerations and a higher degree of iterations, it was found that to fulfil the above condition, an unreasonably bulky and longer mechanism was required. Since the purpose of the transmission system in this case is to drive the rotating valve which has no loads on it, fulfilling the above condition was not a necessity and as a result the selection process was continued without fulfilling the condition.

A belt of width 40mm was finally selected

Timing pulley parameters:	Timing belt parameters
Section 14M (RSB),	Designation: PHG 1610-14M
Number of teeth:28;	Number of teeth per side: 115
Belt width: 40mm ;	Pitch length: 1610

4.5.6 Shaft dimensioning

The shaft carries the timing pulley which exerts forces on the shaft in the transverse direction (perpendicular to the shaft axis). Bending moments are induced in the shaft due to these transvers forces and this bending calls for analysis of stress in the shaft. The main purpose of sizing the shaft was to determine the minimum diameter required to meet the strength requirements that can support the loads without any considerable shaft deformations.
The general layout of the shaft to accommodate shaft elements was determined earlier. The beam theory was used to determine a constant shaft diameter that can accommodate the SKF specified bearing misalignment. Figure 37 represents the general layout of the shaft.

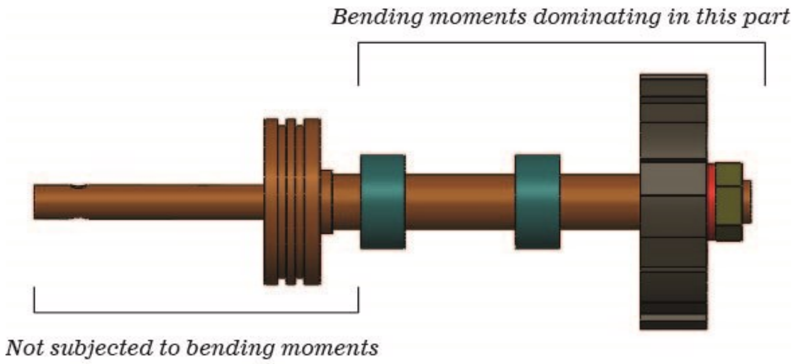

Figure 37: Shaft general layout with bearings and timing pulley

Determination of reaction forces and the bending moment

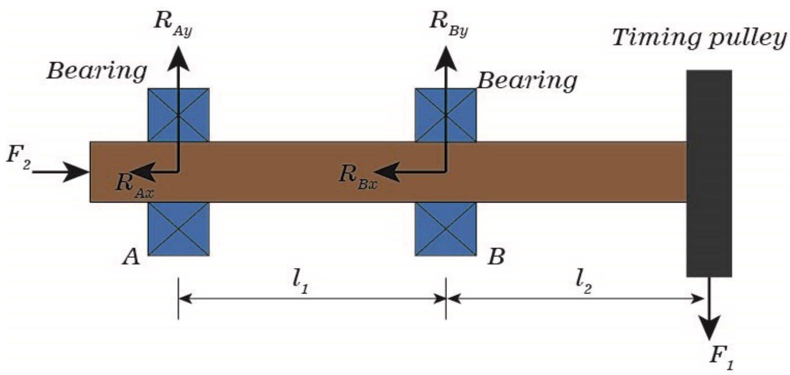

Figure 38: Simplification of the shaft and the forces acting on it

Figure 38 illustrates the simplified diagram used to determine the reaction forces.
The force F_1 is due to the torque transmitted by the tight side of the timing belt. The following equations have been used in determining the required parameters.
Note: See more details on how the equations have been employed in the appendixed Matlab code (Appendix A)
The slack side of the timing belt exerts no force on the timing pulley hence the total bending force on the shaft is given by equation (14):

$$F_1 = \frac{2 * T}{D_p} = 3.06\ kN \tag{14}$$

With T as torque transmitted by the belt and D_p as the pitch diameter of the pulley
The torque transmitted by the belt is given by the equation (15);

$$T = \frac{P}{\omega} = 191\ Nm \tag{15}$$

Where P is the power and ω is the speed. The engine is to operate at speeds ranging from 1500-3000 rpm hence to get the maximum torque a speed of 1500 rpm was used.
The axial force F_2 is exerted by the pressure acting on the shaft and is given by equation (16):

$$F = p * A \tag{26}$$

$$F_2 = 4.96\ kN$$

with P as the pressure and A as the area on which the pressure is acting upon.
Equation (17) and (18) gives the reaction forces on the bearings;

$$R_{Ay} = \frac{F_1 * l_2}{l_1} \tag{17}$$

$$R_{Ay} = 10.2\ k\ (\downarrow)$$

$$R_{By} = \frac{F_1 * (l_1 + l_2)}{l_1} \tag{18}$$

$$R_{By} = 13.3k$$

$$R_{Ax} = R_{Bx} = \frac{F_2}{2} = 2.5kN$$

Total reaction forces at the bearings:

$$R_A = 10.5\ kN; \qquad R_B = 13.5kN$$

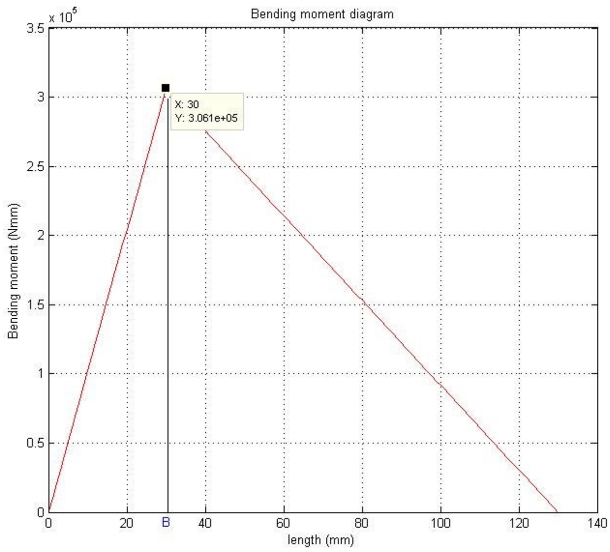

Figure 39: Bending moment diagram

From the bending moment diagram (Figure 39) it can be noted that the maximum bending moment is at point B and has a magnitude of $3.06*10^5$ Nmm

Shaft sizing

The smallest safe shaft diameter was determined using von Mises yield criteria. The material assigned to the shaft is Alumina which has a yield strength of 550 MPa and elastic modulus of 390 GPa.
Equation (19) is the Von Misses yield criterion:

$$\sigma_e = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - \sigma_x\sigma_y - \sigma_y\sigma_z - \sigma_x\sigma_z + 3\tau_{xy}^2 + 3\tau_{yz}^2 + 3\tau_{xz}^2} \tag{19}$$

With σ_e as the equivalent stress.
Maximum bending stress is calculated using equation (20):

$$\sigma_{max} = \frac{M_b}{W_b} \tag{20}$$

Where M_b is the bending moment and W_b section modulus defined by equation (21):

$$W_b = \frac{\pi * d^3}{32} \tag{21}$$

Maximum shear stress is given by equation (22):

$$\tau_{max} = \frac{T}{W_v} \tag{22}$$

With d as the shaft diameter and W_v as the torsional section modulus defined by equation (23);

$$W_v = \frac{\pi d^3}{16} \tag{23}$$

Von Misses criteria transforms to equation (24);

$$\sigma_e = \sqrt{\sigma_{max}^2 + 3\tau_{max}^2} \tag{24}$$

Safety factor (SF): The safety factor was taken as 3 and is defined by equation (25):

$$SF = \frac{\sigma_y}{\sigma_e} \tag{25}$$

: See Matlab code (Appendix A, shaft sizing) for detailed use of formulae
The shaft diameter is given by equation (26):

$$d = \sqrt[3]{\frac{SF}{\sigma_y * \pi}\sqrt{(32M_b)^2 + 3(16T)^2}} \tag{26}$$

The smallest safe shaft diameter is 26.83 mm, say 35 mm *(to fit the selected bearing)*

Deflection of the shaft

The deflection of the shaft and the misalignment of the shaft was determined using the double integration method as defined by equation (27):

$$EI * \frac{d^2y}{dx^2} = M \tag{27}$$

With *y*: the deflection of the beam, *E*: modulus of elasticity, *I*: the moment of inertia of the shaft cross section about the neutral axis, *M*: bending moment at the distance *x* from one end of the shaft. For details see the appendixed Matlab code.

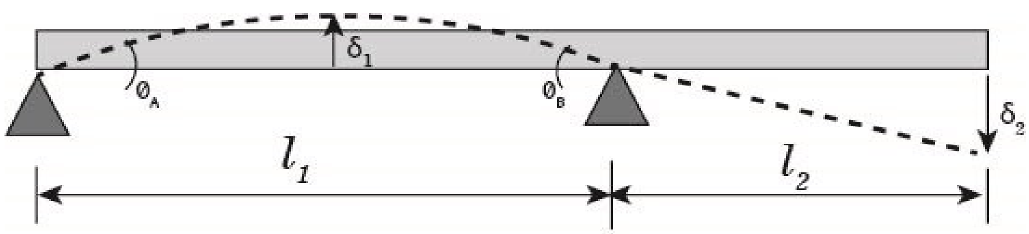

Figure 40: Illustration of shaft deflection

Figure 40 illustrates the deflection of the shaft under loads.

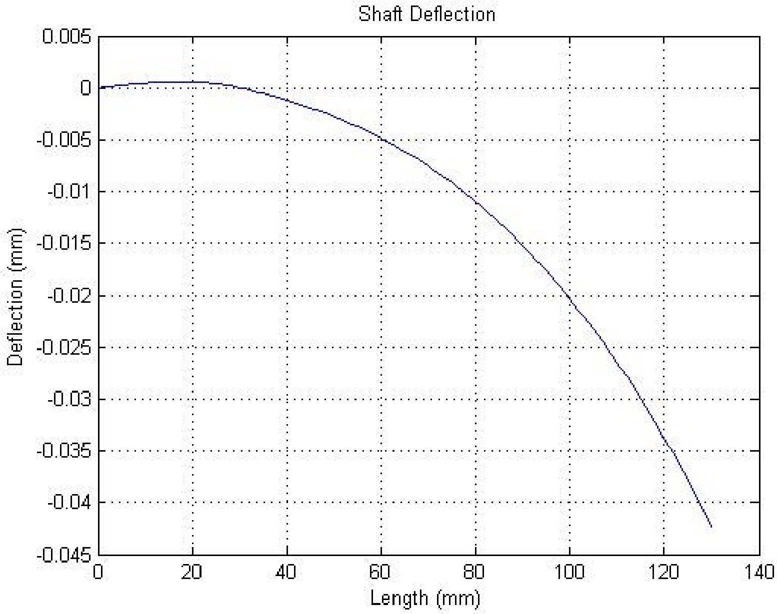

Figure 41: Deflection of the shaft

From figure 41, the maximum deflection of the shaft is 0.042 mm which is reasonable and acceptable.
The calculated misalignments are as given below in minutes of arc:

$$\theta_A = 0.17', \quad and\ \theta_B = 0.35'$$

4.5.7 Dynamic Analysis

A rotating shaft will always deflect due to rotation, even if its unloaded. At certain speeds known as critical speeds, the unbalanced mass of the shaft caused by rotation, will cause resonant vibration. The deflection of the shaft depends upon the following factors;

- The stiffness of the shaft
- The imbalance of the shaft
- The damping in the components
- The total mass of the shaft and other components attached to it

The theoretical angular velocity of the shaft that excites the natural frequency of the shaft is known as the critical speed. Critical speed occurs when the rotational speed is equal to the numerical value of the natural vibration. To reduce vibrations and noise of a rotating system, it is always important to calculate the critical speed.
A rotating shaft can rotate in different mode shape with corresponding but the first natural frequency is needed in the case of a rotating shaft.
The Rayleigh-Ritz equation (equation 28) is used here to calculate the first natural frequency of a shaft which has been divided into *n* segments:

$$\omega_1 = \sqrt{\frac{g * (\sum_{i=1}^{n} \omega_i y_i)}{(\sum_{i=1}^{n} \omega_i y_i^2)}} \qquad (28)$$

Where g is the acceleration due to gravity, ω_i are the weight of each segment, y_i are the gravitational static deflections of the canter of each segment; g is the acceleration due to gravity.

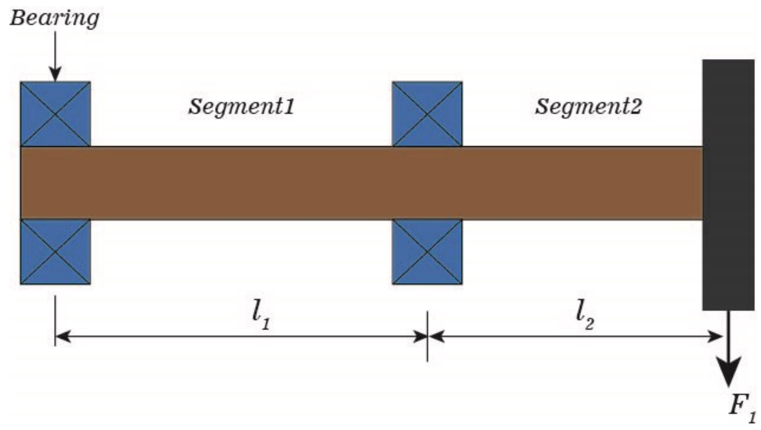

Figure 41: Shaft segments for vibration analysis

For Segment 1:
Deflection from shaft weight is determined by equation (29):

$$\delta_{11} = \frac{5wl_1^3}{384EI} \tag{29}$$

$$\delta_{11} = 9.4 * 10^{-6}mm$$

The deflection from load only (δ_{12}) was calculated in the above section and was found to be $5.6*10^{-4}$mm
Total deflection (δ_1):

$$\delta_1 = \delta_{11} + \delta_{12} = 5.6 * 10^{-7}m$$

Critical speed (N_{c1}) is calculated from equation (30):

$$N_{c1} = \frac{30}{\pi}\sqrt{\frac{g}{\delta_1}} \tag{30}$$

$$N_{c1} = 4 * 10^4 rpm$$

The operational speed (N_1) is taken as 75% of the critical speed (as per recommendations by Kruger)

$$N_1 = 30\ 000\ rpm$$

For Segment 2:
Deflection from shaft weight is given by equation (31):

$$\delta_{21} = \frac{wl_2^3}{8EI} \tag{31}$$

$$\delta_{21} = 1.93 * 10^{-5}mm$$

Deflection from load only (δ_{22}) was calculated in the above section and was found to be 0.04 mm
Total deflection (δ_1):

$$\delta_2 = \delta_{21} + \delta_{22} = 4 * 10^{-5}m$$

Critical speed (N_{c2}) is determined by equation (29):

$$N_{c2} = \frac{30}{\pi}\sqrt{\frac{g2}{\delta_2}} = 4730rpm$$

The operational speed (N_2) is taken as 75% of the critical speed (as per recommendations by Kruger in the Technical Bulletin, TBN017, 1998)

$$N_2 = 3546\, rpm$$

The minimum operating speed of the two segments is 3546 rpm which is well above the maximum speed of 3000 rpm.

4.5.8 Bearing selection

The bearing to be selected should withstand the operating conditions given in Table 6:

Table 6: Bearing Requirements

Condition	Specification
Temperature	450 °C
Speed	3000 rpm
Radial load (F_r)	13.5kN
Axial load (F_a)	4.96 kN

According to the study and evaluation on high temperature bearings, there is no available angular contact bearings that can work at the given temperatures. High temperature bearings that are available on the market are mainly deep groove ball bearings. As a result of this, the mechanism was modified to take axial loading by itself hence deep groove ball bearings can be applied.
The VXB high temperature deep groove ball bearing 6007 (35*62*14) was selected for the application. (VXB Group)

4.5.9 Preloaded fasteners in tension

The modified design employs a lid fastened to the housing as illustrated in Figure 43. The lid is now taking all the axial loading and the bearing is only loaded radially. The formulae used here are from. (R. L. Norton)
The axial force exerted on the lid was calculated above as 4.96 kN (the axial force)
If four bolts are used, then each bolt is carrying a force of 1.24 kN
A hexagonal head bolt *M4*, with shank and coarse threads, class *9.8* was selected.

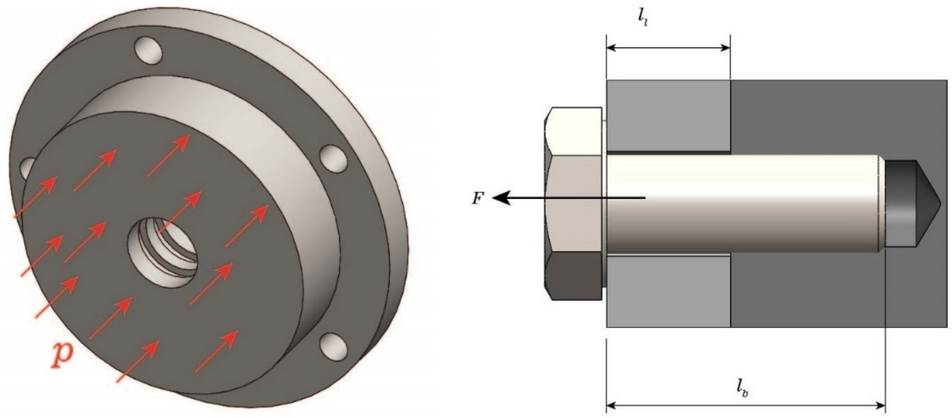

Figure 43: Illustration of the pressure acting on the lid (left) and a bolt joint in tension (right)

The yield stress force *(F_y)* of the bolt is given by equation (32):

$$F_y = \sigma_y * A_s \tag{32}$$
$$F_y = 5.92\ kN$$

Where σ_y is the yield stress of the bolt and A_s is the tensile stress area defined by equation (33),

$$A_s = \frac{\pi * d_p^2}{4} \tag{33}$$

With d_p as the pitch diameter of the threads.
A preload of *85%* of the bolt's yield stress force was set as first trial.

$$\Rightarrow F_0 = 0.85 * F_y$$
$$F_0 = 5.03\ kN$$

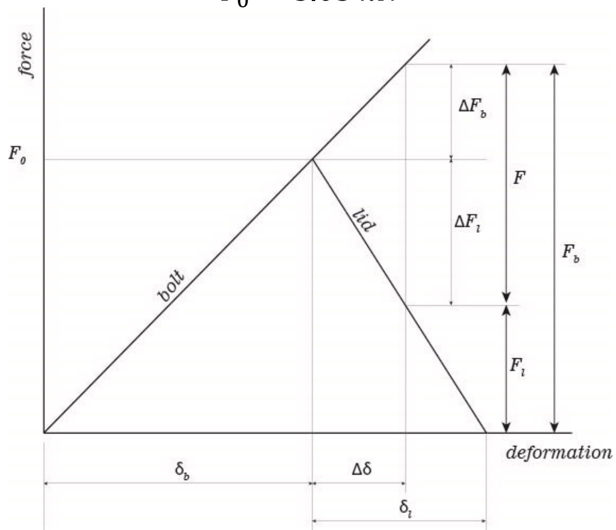

Figure 44: Load deflection and resulting forces in a preloaded bolt

Figure 44 represents the force and deflection diagram in a preloaded diagram. The designation in the above diagram are the same as described in the following sections.
The stiffness of the bolt *(k_b)* was calculated from equation (34):

$$\frac{1}{k_b} = \frac{l_t}{A_s * E_b} + \frac{l_b}{A_b * E_b} \tag{34}$$

Where A_s and l_t is the tensile stress area and length of the threaded part respectively; l_b and A_b is the length of the unthreaded part and the cross sectional area respectively, E_b is the Young modulus of the bolt material.

The stiffness of the lid was calculated by equation (35):

$$\frac{1}{k_l} = \frac{\pi(D^2 - d^2)}{4} * \frac{E_l}{l_l} \tag{35}$$

Where E_l is the young modulus of the material of the lid and l_l is the width of the lid.

The joint stiffness factor *(C)* is then defined by equation (36):

$$C = \frac{k_b}{k_l + k_b} \tag{36}$$

$$C = 0.0032$$

It then implies that, the respective forces felt by the bolt and the lid are defined by equation (37) and (38)

$$Portion\ of\ applied\ force\ on\ belt: \quad \Delta F_b = C * F \tag{37}$$

$$Portion\ of\ applied\ force\ on\ lid: \quad \Delta F_l = (C - 1) * F \tag{38}$$

$$\Delta F_b = 3.9\ N;\ \Delta F_b = 1.2kN$$

The resulting forces on the bolt and lead due to the applied force *(F)* are defined by equation (39) and (40):

$$F_b = F + \Delta Fb \tag{39}$$

$$F_l = F - \Delta F_l \tag{40}$$

$$F_b = 1.24\ kN; \quad F_l = 3.9\ N$$

Maximum tensile stress in the bolt (σ_b) is given by equation (41);

$$\sigma_b = \frac{F_b}{A_s} \tag{41}$$

The principal stresses and the von Misses are the same as the applied force in this case since it's a uniaxial stress situation. This implies that the factor of safety *(SF_y)* against yielding is given by equation (42):

$$SF_y = \frac{\sigma_y}{\sigma_b} \tag{42}$$

σ_y is the yield strength

$$SF_y = 4.8$$

Equation (43) defines the load required to separate the joint *(F_j)*:

$$F_j = \frac{F_0}{1 - C} \tag{43}$$

$$F_j = 5.04\ kN$$

Equation (44) is the Safety factor against joint separation *(SF_j):*

$$SF_j = \frac{F_0}{F} \tag{44}$$

$$SF_j = 4.05$$

The two safety factors calculated are acceptable and their values indicate that the joint can resist yielding and failure.

4.5.10 Thermomechanical analysis

The change of dimensions and the change of mechanical properties of the components was analyzed since they are subjected to high temperatures of 450°C. The major goal for the thermomechanical analysis was to determine the thermal expansion of each component in order to assign tolerances to the parts. Thermal expansion of the components might cause jamming of the mechanism.

In this mechanism, there is a small radial clearance between the valve's periphery and the housing, which ensures that any thermal or mechanically induced distortion of the valve body can be accommodated without the mechanism getting stuck.

The clearance between the valve body and the housing should be sufficient enough to accommodate thermal expansion. Due to high temperatures, the components are most likely to deform and the thermal effects could be of greater magnitude. Figure 45, 46 and 47 illustrates the thermal deformation of the critical components of the assembly.

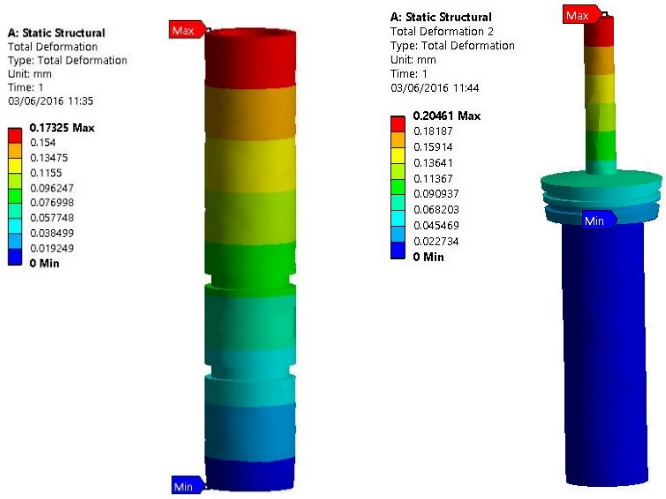

Figure 45: Thermal expansion of the intermediate shaft(left) and the motor shaft(right*)*

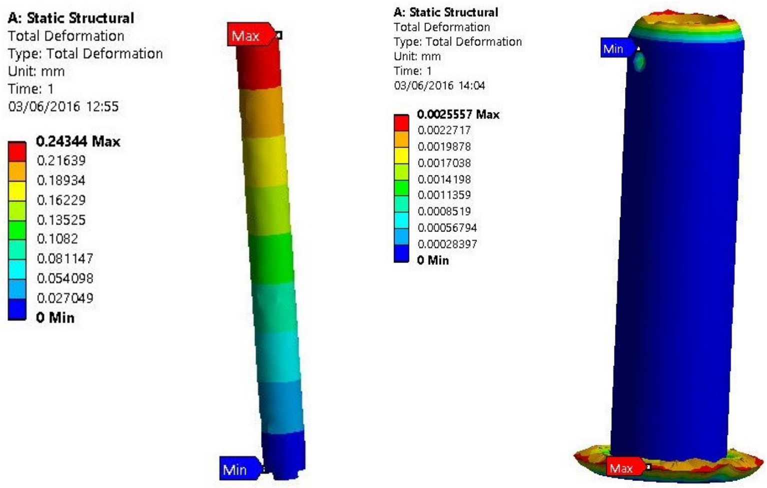

Figure 46: Static deformation of actuator shaft (left) and the cylinder lining(right)

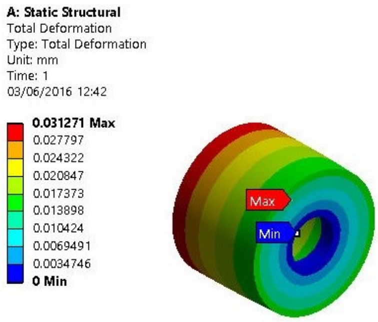

Figure 47: Static deformation of the connector

4.6. Selection of a seal design

Sealing methods have always been a major problem with the rotary valve mechanisms. For this reason, the sealing is one of the most critical component for the operation of the valve. Various latest sealing solutions have been studied and evaluated and an optimum design was selected. The major important primary factors taken into consideration when selecting seals were:

- The operating temperature (450 °C)
- The operating pressure (50 bars)
- Maximum speed of the shaft (3000 rpm)
- The nature of the liquid to be contained (super-heated steam)

According to the study and evaluation carried out on different sealing solutions, it's very difficult to find out an off-shelf sealing that can meet the demands on the designed valve mechanism. As a result, most seals that operate in high-temperature-high-pressure (HTHP) are custom manufactured according to the seal design specifications. The global leader in sealing technologies that operates in critical and demanding conditions, Parker has been conducted as a potential manufacturer of the gas-tight seal required for the efficient operation of the valve mechanism. According to the company's catalogue, (Parker Hannifin Corporation), there a is a guarantee of producing a sealing solution that works well in the given conditions since the company specializes in seals that can operate at even higher temperatures and pressures than demanded in this case.

The sealing solution suggested for the valve is a gas tight and double arrangement seals, that has an active sprung sealing mechanism that can accommodate large thermal distortions. The seal should be metallic, dynamic, operating between a rotating shaft and a stationery component, gas-tight with tight tolerances. Figure (48) is an illustration of a gas-tight dynamic sealing from Parker.

Figure 48: An example of a dynamic, HTHP gas-tight seal. (Parker Hannifin Corporation)

This chapter summarizes the final engineered results from conceptual analysis and modelling. The interpretation of the results is hereby presented.

Three concepts were developed, studied and evaluated. Pugh's decision matrix was used to select the best optimum solution, taking into account a number of interlinked factors. The cylindrical rotary valve (concept 3) emerged as the winning champion because of its inherent advantages over other concepts. The Pugh matrix was not only used to determine the winning candidate but also to identify the weaknesses of the best solution. The concept was further analyzed and modelled and the outcome is presented below.

Material selection;

Potential materials that can meet the demanding operating conditions of high temperature and high pressure have been identified, listed and evaluated. Pugh's decision matrix was established to compare the materials and evaluate them against a set of criteria. The most important factors were the ability to resist corrosion and oxidation at high temperatures and also the ability to work at high temperatures and pressure without considerable thermal deformation. Ansys was used to simulate the thermal expansion of each material under the given operating conditions. The valve body is the most critical component in the valve mechanism hence it was used for thermal analysis. From a list of potential material candidates, Alumina came out as the winner because of its optimum characteristics. The following figure depicts the results from Ansys simulation to determine the thermal expansion of Alumina;

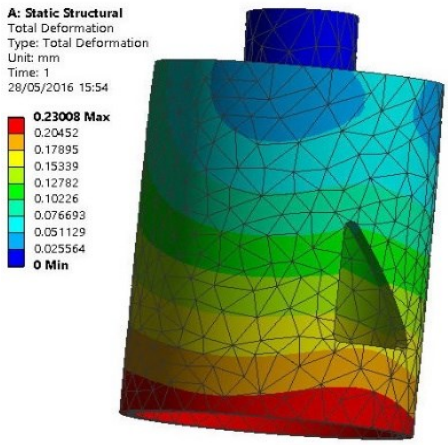

Figure 49: Static deformation of Alumina under high temperature and pressure

Concept analysis:

In order to determine the feasibility of the chosen concept, analysis and modelling was carried out and the following results were obtained:
The outlet port of the valve body was dimensioned in a way that it meets the required cut-off time at its maximum operating speed of 3000 rpm. The analysis carried out has demonstrated that the chosen concept has the full potential to be applied in the steam expander.

Table 7: Verification of design specifications

Parameter	Specification	Condition
Speed (rpm)	3000	verified
Filling ratio, F (%)	50	verified
Filling angle, $\boldsymbol{\varphi}_f$	90°	verified
Time for one revolution (*ms*)	20	verified
Cut-off time (*ms*)	5	verified

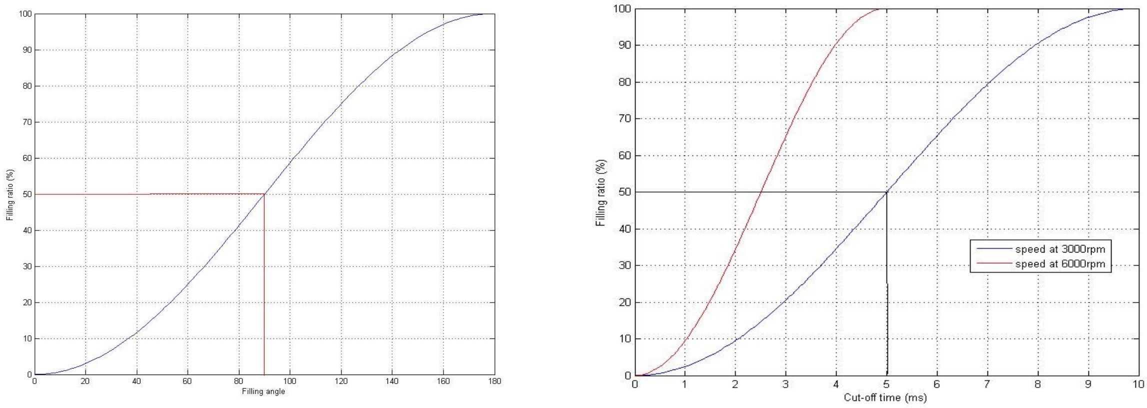

Figure 50: Design specification verification

The above results show that the design specifications have been met. The developed concept also offers a variable cut-off time as represented below.

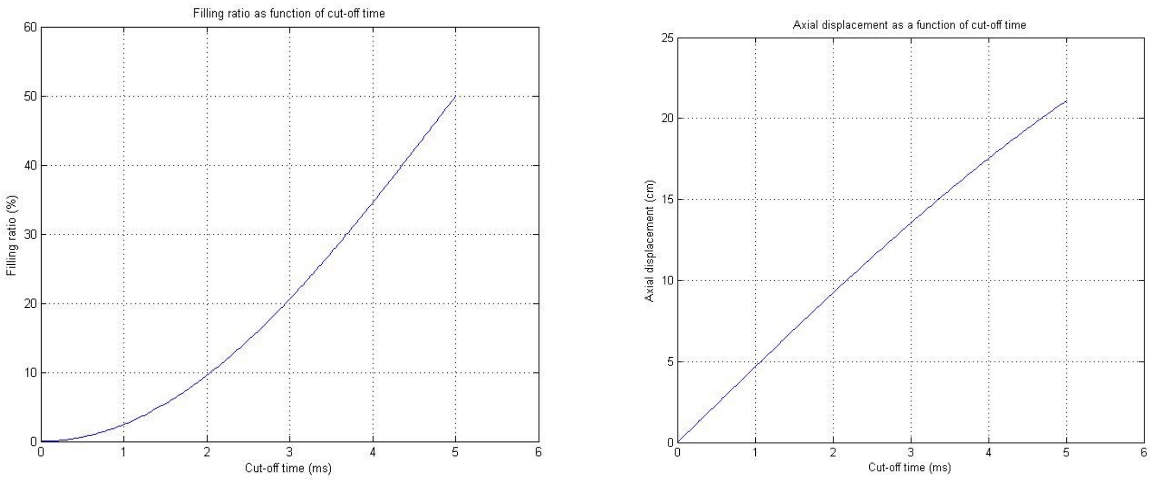

Figure 51: Variable cut-of timing

Transmission system

The transmission system has been dimensioned according to SKF recommendations. However, the corrected power rating was found to be less than the design power which is contrary to the conditions set by SKF. Since the purpose of the belt transmission in this case is to only rotate the valve which has no other resisting torque besides the negligible friction, the set condition was neglected because there are no high torque demands on the valve.

Shaft sizing

The sizing of the shaft was done using von Mises criterion. A minimum diameter of 28 mm is required for the shaft to support the given loads. However, the shaft was assigned a diameter of 35 mm in order to fit the chosen bearing.

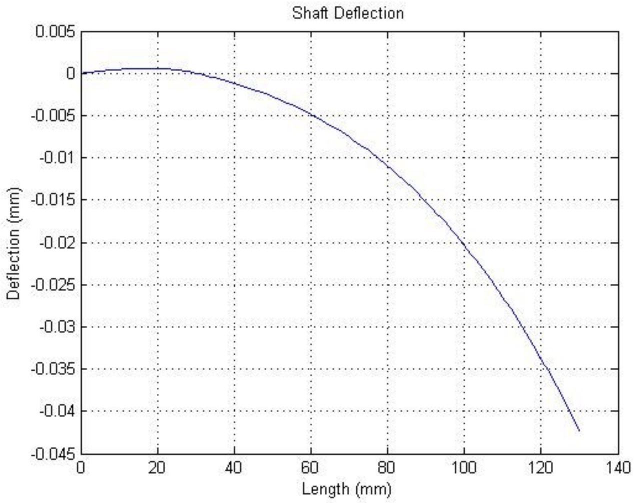

Figure 52: Deflection of the shaft

A maximum shaft deflection of 0.042 mm was obtained and this is acceptable.
The bearing misalignments are less than 1 minute of arc and this is also accepted since its far less than the maximum misalignment of 10 minute specified by the manufacturer.
The selected bearing is a VXB high temperature, deep groove ball bearing 6007 which can operate at 500 °C. Angular contact bearings that can operate at this temperature couldn't be found in the identified supplier's catalogues. As a result of this, the design was modified to take axial loads and allow the use of the available deep groove ball bearings.
The modified design employs the use of a lid fixed to the housing by bolts which are to be preloaded. The analysis of this bolted joint indicated that the joint is safe and can support the acting pressure.
The safety factors of joint separation and joint yielding were calculated as 4.05 and 4.8 which is satisfactory and acceptable.
The dynamic analysis has determined the critical speed of the shaft using Rayleigh-Ritz equation. From the critical speed, the maximum operating speed was found to be 3546 rpm which is a way higher that the given 3000 rpm in the requirements. This implies that the rotating system is safe from significant vibrations.
The results from thermomechanical analysis indicated that the expansion of the components is within the expected range hence the dimensioning of the components was carried out taking the thermal deformation into consideration. According to Ansys simulation results, the chosen materials will not cause jamming of the mechanism.
These results confirm that the application of the chosen concept in the given material is feasible, and the concept is full of potential.

6. Discussion and conclusion

This chapter presents the discussion on the process and outcome of the project and the conclusion drawn.

Discussion

The work carried out in this project has provided a baseline for further development of the rotary valve. The targets of this project have been achieved within the given timeframe. However not all of the problems have been solved but at least the goals of this thesis work have been delivered.

The main target of the project was to provide a conceptual solution that works according to the design specifications. Different solutions have been provided and the best solution was selected and further developed. The analysis and modelling carried out has also proved that the solution has a greater potential to be applied on the engine and other industrial applications.

The selected material has proved to work well in the given operating conditions, at least in theory. The design is to operate in severe conditions of high temperature and high pressure which can distort the components considerably. It is also well known that superheated steam is very aggressive when it comes to reacting with metals. The chosen material, Alumina which is a technical ceramic material has proved to withstand these conditions.

The main challenges in the history of rotary valve are high friction and wear between surfaces in contact, and sealing problems.

The high frictional forces in the selected concept are due to the valve body rubbing against the housing. This has been addressed by providing a graphite lining inside the valve housing. Graphite is a very good lubricant at the given temperature and in such an oxidizing environment. Also in case of wear, the lining can easily be replaced instead of replacing the whole housing. Tribological and wear analysis were not part of this project because of the limited timeframe. This was recommended for further work as its critical to study the wear properties, tribological aspects and to determine the acting frictional forces.

A leading sealing manufacturer has been identified and conducted and there is guarantee of efficient sealing solutions since they produce seals that operate at more demanding conditions than those specified in this project.

This project has not investigated computational fluid dynamics of the steam due to the lack of expertise of the author in this field. Preliminary calculations on the flow of the steam have been carried out, but the modelling was simplified. Detailed analysis on computational fluid dynamics is recommended, taking into consideration all variable parameters like pressure and temperature.

Though the analysis and modelling carried out in this work have proved that the solution is feasible, it is recommended to validate the concepts through physical prototyping. Even though there is still more work to be completed, the analysis has shown that the developed concept has a full potential to be applied in practice

Conclusion

The work carried out in this project has provided a conceptual solution to the given problem. The results from the analysis have proved that it is feasible to realize the designed rotary valve in practice. Though not all the problems in the design solution have been solved, but at least this thesis has provided a solid foundation upon which further development of a rotating valve can rest upon. The developed solution has greater potential to be applied on a modern steam expander and other industrial applications. The goals of the project have been achieved and the solution developed is very promising. The results obtained in this thesis work provide a theoretical foundation for the development and application of a rotary valve in high-temperature-high-pressure environment.

7. Future work

This section describes the work that need to be carried out for further development of the rotary valve.

The following need to be further studied in order to provide a complete analysis of the selected concept

- Computational fluid mechanics
- Tribological and wear analysis
- Actuator selection and sizing
- Imbalance analysis and rotor dynamics
- Prototyping and testing

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APPENDIX A: Matlab Code

```
% rotary valve calculations...thesis...Nyawo talent
clear all
close all
%% dimensioning of the valve outlet port
Ls=40 % stroke length in mm
d=40 % valve diameter in mm
dh=838; % delta enthalpy in kJ/kg
P=30; % power in kW
m_dot=P/dh; %mass flow in kg/s
rho=16.14; %steam density kg/m^3
Q=m_dot/rho; % flow rate in m3/s
tc=5e-3; % cut-off time in s
trev=0.02; % time in ms for one revolution for a speed of 3000rpm
phi=tc/trev*360 %the filling angle in degrees
theta=phi*pi/180 % filling angle in radians
r=d/2; % valve body radius in mm
s=r*theta % circumferential port length of the valve in mm
V=Q*tc*1e6 % volume in cc
beta=(180-phi)/2 % the other angles of the triangular opening port
b=(r/sind(beta))*sind(phi); %length of chord in mm
v=741.81; % velocity of steam in cm/s
A=Q*1e6/v;%flow area in cm2
h=2*A/(b/10);
%% filling angle and filling time
Phi=[0:180]; % filling angle
Zf=r-(r.*cosd(Phi));
F=(Zf./Ls)*100;
figure(1)
plot(Phi,F)
grid on

ylabel('Filling ratio (%)')
xlabel('Filling angle')
%%
Tc3=Phi.*trev/360; % cut-off time for a speed of 3000rpm
figure(2)
plot(Tc3,F)
hold on
Trev=10; % time in ms for one revolution for a speed of 6000rpm
Tc6=Phi.*Trev/360; % cut-off time for a speed of 6000rpm
plot(Tc6,F,'r');
legend('speed at 3000rpm','speed at 6000rpm')
ylabel('Filling ratio (%)')
xlabel('Cut-off time (ms)')
grid on
%% variable cut-off timing
Beta=atand(28.3/21.1);
x=[0:0.1:21.1]; % axial displacemrnt
z=x.*tand(Beta);
A=acosd((1-z.^2./(2*r^2))); %filling angle
tcc=A.*trev./360;% cut-off time
figure(3)
plot(tcc,x)
title('Axial displacement as a function of cut-off time')
xlabel('Cut-off time (ms)')
ylabel('Axial displacement (cm)')
grid on
Zf1=r-(r.*cosd(A));
F1=(Zf1./Ls)*100;%filling ratio
figure(4)
plot(tcc,F1)
```

```
title('Filling ratio as function of cut-off time')
ylabel('Filling ratio (%)')
xlabel('Cut-off time (ms)')
grid on
%% Valve body thickness
p=5; %MPa
sigma=350 % Mpa
t=(p*d)/(sigma*2)
%% Timming belt selection
pr=30; % kW
c2=1.7 % service factor
pd =pr*c2 % design power
Dp=124.78/1000; % Large Pulley diameter
ddp=124.78/1000; % small Pulley diameter
cd=0.609; %design center distance
z=28; %number of teeth
TIM=(0.5-(Dp-ddp))*z/(6*cd)
c1=1.0; %correction factor
c4=0.6; % correction factor
pb=31.21;
pcorr=c1*c4*pb;
pbb=pd/(c1*c4);
%% Shaft sizing
P=30e3 ;% Power in W
p=5 ;%Pressure in MPa
omega=1500*2*pi/60; % rotational speed in rad/s
T=P/omega ;%torque in Nm
F1=2*T/Dp;
A=pi*(19.5^2-8^2); %area in mm^2
F2=p*A;
l1=30;
l2=100;
l=l1+l2;
Ray=(F1*l2)/l1;
Rby=(F1*l)/l1;
Rax=0.5*F2;
Rbx=Rax;
Ra=sqrt(Rax^2+Ray^2)%total reaction force at A
Rb=sqrt(Rbx^2+Rby^2) %total reaction force at B
x1=[0:l1];
M1=Ray.*x1;
plot(x1,M1,'r')
hold on
x2=[30:l];
M2=F1.*(l-x2);
plot(x2,M2,'r')
ylabel('Bending moment (Nmm)')
xlabel('length (mm)')
title('Bending moment diagram')
%%
% von Misses criterion
Mb=3.06e5; % bending moment Nmm
SF=3; %safety factor
sigmay=700; %yield stregnth MPa
d=(SF/(pi*sigmay)*sqrt((32*Mb)^2+3*(16*T*1000)^2))^(1/3);
% shaft deflection
E=186e3 ; % ypumg modulus Mpa
I=(pi*30^4)/64 % moment of inertia
y1=-((Ray.*x1)./(6*E*I)).*(x1.^2-l1^2)
plot(x1,y1)
hold on
y2=-(1/(E*I))*((F1.*x2.^3./6)+(Ray*l1^2/3-F1*l1^2/2).*x2+(F1*l1^3/3-
Ray*l1^3/3));
plot(x2,y2)
```

```
grid on
ylabel('Deflection (mm)')
xlabel('Length (mm)')
title('Shaft Deflection')
thetaA=-atand(-Ray*l1^2/(6*E*I))*60% misalaignment at A in min of arc
thetaB=atand(Ray*l1^2/(3*E*I))*60 % misalaignment at B in min of arc
%% bolt pretensioning
n=4; %number of bolts
Fx=4.96e3;%axial force
F=Fx/n;%force per bolt
sigmaby=610; % the yield stress of the bolt MPa
As=9.7;  % tensile stress area mm2
Fy=sigmaby*As;%yield stress force
F0=0.85*Fy;% preload force
lt=14;
lb=11;
Eb=210e3;% Mpa
Ab=pi*2^2/2;
kb=((lt/(As*Eb))+(lb/(Ab*Eb)))^(-1);
d=40;
D=54;
El=120e3 ;%Mpa
ll=6;
kl=(pi*(D^2-d^2)/4*El/ll);
C=kb/(kl+kb);
dFb=C*F;
dFl=(1-C)*F;
Fb=F+dFb
Fl=F-dFl
sigmab=Fb/As
SFy=sigmaby/sigmab
Fj=F0/(1-C)
SFj=F0/F
```