Concept proposal of a new oil distribution system for controllable pitch propellers

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Preface

This bachelor thesis is written at the Department of Mechanical and Marine Engineering at Western University of Applied Sciences (WNUAS). The report is written by three students in the study program General Mechanical Engineering at campus Kronstad, in the spring of 2020.

The bachelor thesis is written in cooperation with Wärtsilä Norway, that has provided the problem statement, and guided us throughout the semester. Thank you for allowing us to come visit your facilities at Rubbestadneset and giving us the opportunity to do our thesis with you. A special thank you to Anders Hedin and Lukasz Bartnik for outstanding knowledge and guidance.

We also want to thank Associate Professor Mariusz Domagala for great feedback along the way, especially for the help in modelling.

Abstract

In some vessels that use controllable pitch propellers there is a need for an oil distribution shaft that delivers hydraulic oil from the hydraulic power pack to the propeller to control pitch position. Therefore, there is the need for an oil distribution system capable of transmitting oil from a stationary part into a rotating shaft, and that can measure pitch position which can be transmitted to a control panel.

This thesis focuses on the concept proposal of a new Oil Distribution shaft, computational fluid dynamics analysis (CFD) of a journal bearing and finite element method (FEM) analysis of the shaft for Wärtsilä. One of the main goals of the concept proposal was to reduce the overall cost, this was done by focusing on the main cost drivers. Several different concept proposals focusing on oil distribution functionality and feedback of pitch position were created and evaluated. Through discussions with Wärtsilä and within the group, some ideas from each of these concepts were further developed into a finalised concept proposal.

The thesis also explores journal bearing design calculations and the function of journal bearings. In this part there is also a CFD analysis to get a visualisation of pressure distribution in a journal bearing.

Sammendrag

I noen fartøy som bruker kontrollerbare pitch propellere er det nødvendig å ha en olje distribusjons aksling som leverer hydraulisk olje fra den hydrauliske kraftenhet for å kontrollere propellenes pitch posisjon. Derfor er det nødvendig å ha et olje-distribusjonssystem for kan overføre olje fra en stasjonær del inn i en roterende aksling, og som kan måle pitch posisjon som kan sendes til et kontrollpanel.

Denne bacheloroppgave fokuserer på et konseptforslag av en ny olje distribusjons aksling, numerisk fluiddynamikk analyse av glidelager og endelig element-metode analyse av akslingen til Wärtsilä. Et av hovedmålene med konsept forslaget var å redusere den totale kostnaden. Dette var gjort ved å fokusere på hoved kostnadsdriverne. Flere forskjellige konsept forslag fokusert på olje distribusjons funksjonalitet og avlesning av pitch posisjon ble utviklet og evaluert. Gjennom diskusjoner med Wärtsilä og innad i gruppen ble ideer fra flere av konseptene videre utviklet til et ferdigstilt konsept forslag.

Rapporten utforsker også kalkulasjoner av glidelager design og det funksjonelle ved glidelagre. I denne delen er det også gjort en numerisk fluiddynamikk analyse for å visualisere hvordan trykkfordelingen er i et glidelager.

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Nomenclature

A = Area [m^2]

ρ = density [kg/m^3]

$P_{projected}$ = projected pressure [N/m^2]

P_{max} = Maximum pressure in hydrodynamic film [N/m^2]

n = Speed [rpm]

l = Length [m]

D_B = Bearing diameter [m]

C_r = Radial clearance [m]

ν = Kinematic viscosity [mm^2/s]

μ = Dynamic viscosity [Pa s]

R = Radius [m]

h_0 = Minimum film thickness [m]

ε = Eccentricity ratio

f = Coefficient of friction

Q = Volumetric flowrate [m^3/s]

Q_s = Volumetric flowrate of sideways leak [m^3/s]

F_T = Tangential force [N]

F_N = Normal force [N]

P_{loss} = Power loss [J/s]

T = Torque [Nm]

ω = Angular velocity [rad/s]

α_t = Stress concentration factor

d_h = Hole diameter [m]

d_i = Internal diameter [m]

e = Width of slot [m]

d = Shaft outer diameter [m]

S = Sommerfeld number

1. Introduction

Controllable pitch propellers (CPP) make it possible for optimal efficiency and high performance while still having low levels of vibrations and noise. CPP enable the vessel to control its propulsion by changing the orientation of the propeller blades instead of changing the revolutions of the engine, this is called pitching the propeller blades. Pitching is done by increasing the hydraulic pressure in different channels that terminates in the propeller hub and operates the pitch mechanism. To distribute the hydraulic oil for this mechanism, there are mostly three methods that are used; oil distribution done inside the reduction gearbox, a box mounted on the cross-section of the propeller shaft (only applicable with reduction gear), or a bearing around the propeller shaft with actuation channels to the centre. This report will mainly focus on the latter, referred to as the oil distribution shaft, or OD-shaft.

The shaft-mounted oil distribution unit that Wärtsilä currently uses is extremely reliable and rarely the cause for major overhaul of the propulsion system. However, because of limitation of the computer programs used when designing the unit, there should be possibilities to streamline the unit and reduce its cost using stronger drawing and analysis programs, whilst maintaining the strengths of today's design. The focus will be to reduce the diameter of the shaft, as this will allow most of the cost-drivers of the unit to naturally decrease in complexity and/or size, while keeping the functionality and strengths of the current unit.

The operator must be able to know at which angle the propellers are stationed at any moment. This is today solved by having a mechanical movement inside the propeller shaft that translates the angle of the propellers into linear movement inside the shaft, that further is transported to the outside of the shaft. The shaft mounted oil distributor and measuring unit is subject to improvements in method of measuring, size, and cost.

This report will also be discussing journal bearings, together with a journal bearing design calculation and computational fluid dynamic analysis.

See Figure 1 and Figure 2 for an overview of today's design.

1.1 Wärtsilä

With operations in over 200 locations in 70 different countries and about 18 000 employees Wärtsilä Oyj Abp is an international company with origins in Finland, they are global leaders within marine and energy solutions. Wärtsilä has four daughter companies in Norway: Wärtsilä Norway AS, Wärtsilä Moss AS, Wärtsilä Valmarine AS and Wärtsilä Gas Solutions AS. In total 1 000 employees in the Norwegian sector, that specialises in sale of ship design, propulsion systems, electro and automation systems and navigation systems for ship and offshore instalments to customers around the world. This thesis is written in cooperation with Wärtsilä Norway AS on Rubbestadneset on Bømlo in western Norway, which focuses on propulsion systems. [1]

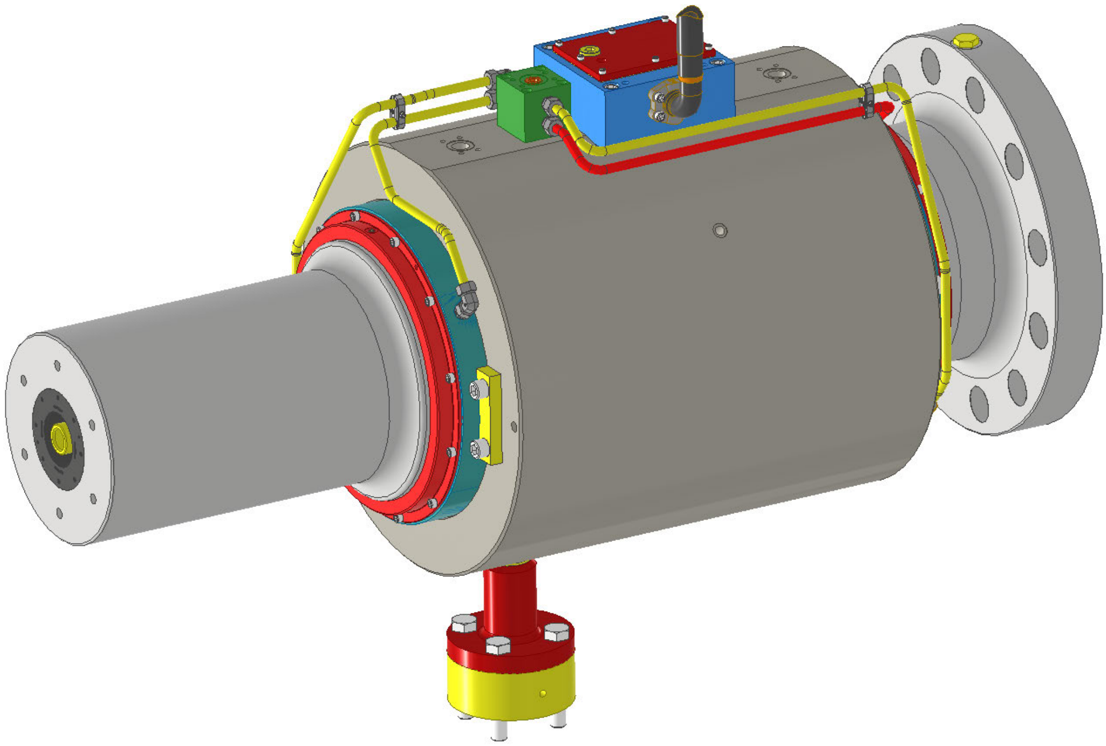

Figure 1: RO-unit

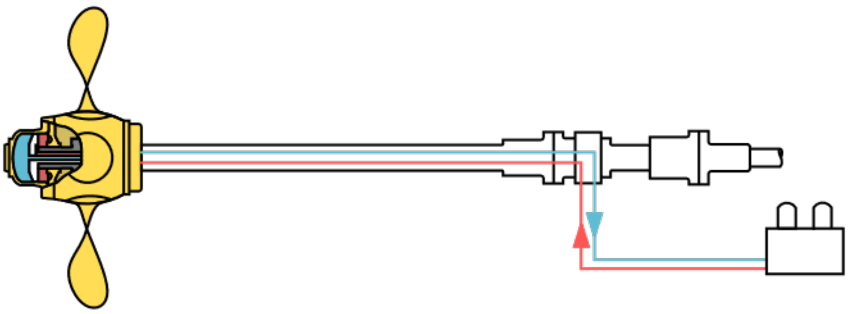

Figure 2: CPP system overview

2. Method

This chapter explores the different methods used to answer the given problem statement. How and why they are used.

2.1 Theoretical approach

A theoretical approach was used to get a better understanding of mainly two areas of this report.

2.1.1 Theory of stress concentrations

To get an understanding in how a hole or a slot would affect the stress concentrations in the shaft a theoretical approach was used. There were done calculations of the stress factors caused by holes and slots in various shaft diameters.

2.1.2 Theory of journal bearings

For journal bearing calculations a theoretical approach was used. This was first to get an understanding of how and why a journal bearing works. The theoretical approach was further used to be able to calculate the various properties of a journal bearing given certain parameters.

2.2 Tools

Digital tools are regularly used in any engineering task done today, supporting both calculations and visualisations. This report used several different tools to analyse, draw, and visualise and this chapter will give a brief explanation of the main methods and tools used.

2.2.1 3-dimensional Computer-Aided Design software

To be able to present an understandable solution proposal, a 3D model was created with the help of Computer-aided design (CAD) software. 3D CAD software was used throughout the design process as a measure to make 2D sketches into solid models. These models were then used for FEM-analysis and further tweaking of the geometry. The software used for the thesis is called Autodesk Inventor. Autodesk inventor is a professional CAD software for 3D mechanical design, simulation, and documentation [2].

2.2.2 Finite Element Method (FEM)

When designing a new geometry of the OD-shaft, fine element method analysis software was used. The software used is called Ansys. This is an Ansys workbench-bundle composed of Ansys Mechanical, Ansys CFD, Ansys Autodyn, Ansys SpaceClaim and Ansys DesignXplorer. Ansys Mechanical and Ansys CFD has been used to test load capability and limits in terms of different types of forces applied to structural components, as well as used for fluid analysis in journal bearings. Information gattered was then used to further improve the design [3].

The version of Ansys used in this report is a student version. This means that the software is limited in the sense that fine mesh and complex geometry is highly restricted. It is used however to get an understanding of the behaviour in the material as geometry changes. The main function of the software is therefore a tool for doing comparison studies of the existing and new geometry.

To counteract the limitations of the student license, mesh method is highly important. Concentrating the mesh around the sections or geometry have high stress values. The models that are studied are also simplified, where areas of the geometry which is assumed to have low stress concentration have simplified geometry to be as efficient with the node allocation as possible.

The studies in this thesis that are looking at maximum stress will use the same worst-case scenario boundary conditions; these values can be found in Table 1.

The preliminary FEM analysis showed that force acting on the shaft from the weight of parts like housing and insert had a small impact in maximum stress.

Load	Value
Torque	401,1 kNm Applied on cross-section of flange
Fixed support	Applied at cross-section of shaft, propeller side

Table 1: Boundary conditions

2.2.3 Computational Fluid Dynamics Analysis (CFD)

From the information acquired in the theoretical approach to journal bearings a CFD model was developed for the bearing that was calculated in chapter 3.5.5. The program used to do the CFD analysis is called Ansys Fluent and is in the bundle of software in Ansys Student. Originally the plan was to develop a method for CFD analysis of the current journal bearing insert in use today. This turned out to be too time consuming and was therefore changed to be just a simple journal bearing analysis.

3. Theoretical background for the report

In this chapter will the important background information be presented. The function of important parts, and information that gives an understanding for the system in which this report is about.

3.1 Description of CPP

Controllable pitch propeller or CPP is a system in which the propeller pitch can be change depending on how much thrust the operator want. The pitch is defined as the distance the propeller screws itself forward in the water with non-slip conditions. [4, p. 15] The engine turns with constant speed, measured in revolutions per minute. When the propeller blades change to a higher pitch the resistance to rotation follows to become higher as well. To compensate for this the engine torque output rises and keeps the revolutions constant, and vice versa for lower pitch angle. One key advantage for CPP is that the operator can switch from ahead to astern without shutting off the engine and restarting it. Fixed pitch propellers need to change the turning direction of the propeller to switch between ahead and astern. For FPP this is done by changing the turning direction of the engine itself. FP-propellers is also depending on the engine or gearbox to change its revolutions to then have a change in thrust consequently. Here the CP-propellers have a great advantage as they allow the engine to operate at any desired load and/or speed, which allows for the engine to operate at its most efficient state. It is also the best choice to achieve swift manoeuvring. This can be compared to a continuously variable transmission found in some cars.

CPP can operate with direct input from the engine which reduces the direct cost of expensive equipment. There is however a slight loss in efficiency at lower speeds when operating with constant rpm and CP-propellers. If the CPP was combined with a synchronous gearbox and followed a combinator curve, i.e. variable pitch combined with variable rpm output, the efficiency would be higher. This is done in today's system.

It must be said that with CPP over FPP there is a slight loss in propeller efficiency, typically 1-2%. [4, p. 17] This comes as a consequence of the larger propeller hub needed to operate the rotation of the propeller blades.

3.2 Today's OD-shaft system

To operate the CPP there is a need for oil pressure going out to the hub. This report will look at the system in which the hydraulic oil is delivered to the propeller shaft through an oil distribution unit mounted directly on the propeller shaft. The system in place must therefore be functional while rotating. This is achieved with the help of journal bearings. With journal bearings a stationary part mounted to a rotating shaft is possible. Through these journal bearings a hydraulic oil flow from stator, meaning the stationary component, to the rotor, meaning the rotating component is possible. Hydraulic flow is lead into the shaft distribution line through a composition of holes in the housing, journal bearings, feedback rod, and guiding bush seen in Figure 3.

In today's system the oil distribution uses two journal bearings for ahead and astern actuation. In between these lays the system for reading the pitch position. When the pitch changes, an inner pipe moves inside the shaft. Attached to this pipe is the feedback rod. In the middle of that feedback rod the feedback ring fixture extends out to a sliding ring called feedback ring that rotates with the shaft. As the pitch changes this ring slides back and forth along the shaft. In the slot of the feedback ring lays a sliding block which is attached to an adjusting lever. When the feedback ring slides along the shaft this sliding block makes the adjusting lever move to either side. In the space where the feedback ring, ring fixture and sliding block there is oil always used for lubrication of the components. From the adjusting levers movement, a "knob" connected at the top of the lever rotates, this is then translated into an electrical signal. The electrical signal is then sent to the control panel so that the operator can check the pitch and adjust accordingly.

Figure 3: Original RO-unit nomenclature

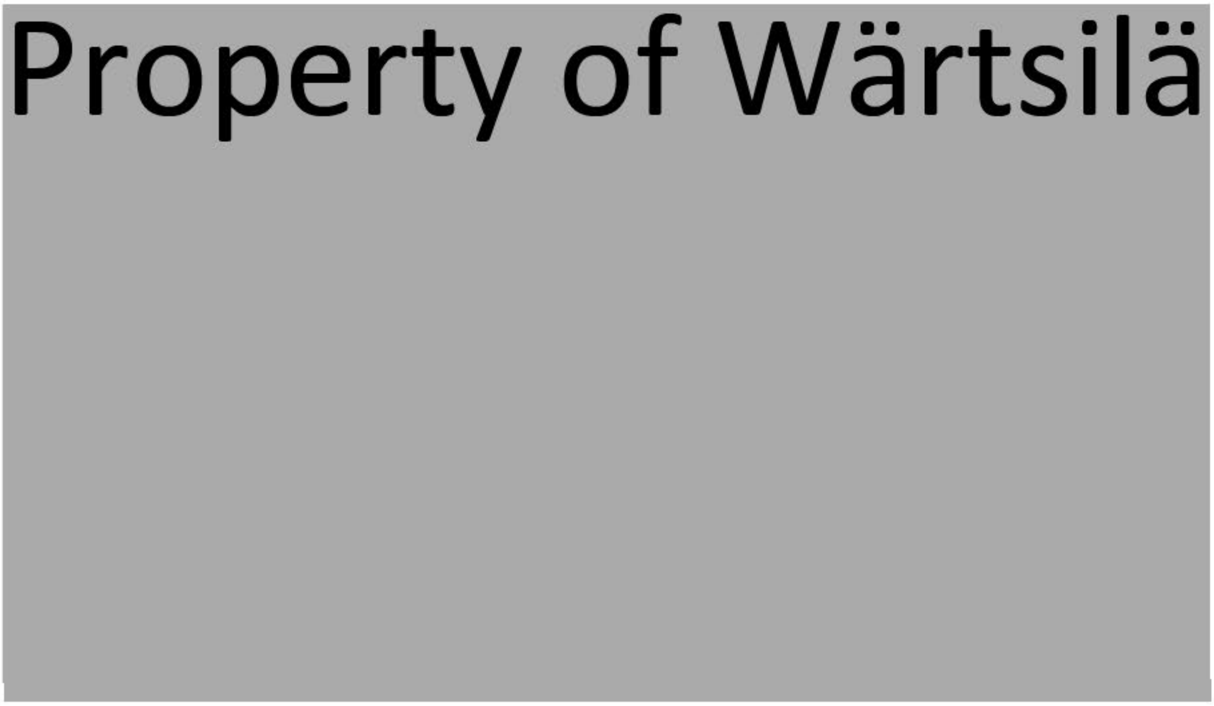

Figure 4: Original RO-unit nomenclature

3.3 Materials

Part	Material	Tensile strength R_m [N/mm^2]	0.2% proof Stress $R_{p0,2}$ [N/mm^2]	Possible alternatives
Journal Bearing Insert	GGG40	≥ 400	-	ASTM A536, GRADE 60-40-18 ASTM A536, GRADE 65-45-12 ASTM A536, GRADE 80-55-06
Housing	C45E N	≥ 560	≥ 275	EN8
Servo Shaft	C45	≥ 560	≥ 370	EN8

Table 2: Material properties [5] [6]

Material	Tensile strength, min, MPa	Yield strength, min, MPa	Elongation in 2 in. or 50mm, min, %	0.2% proof Stress $R_{p0,2}$ [N/mm^2]
ASTM A536, GRADE 60-40-18	414	276	18	-
ASTM A536, GRADE 65-45-12	448	310	12	-
ASTM A536, GRADE 80-55-06	552	379	6	-
EN8 Normalised	550	280	-	-

Table 3: Alternative materials [7] [8]

3.3.1 Bearing, housing, and inserts

Component properties				
Component	***Material***	***Density, kg/m^3***	***Volume, m^3***	***Weight, kg***
Housing	C45E N	7850	0,16543	1298,6
Journal Bearing Insert	GGG40	7300	0,01926	149,6
White metal	Babbitt 80	7300	N/A	N/A

Table 4: Component properties [9] [10] [11]

3.4 Shaft geometry

The geometry of the shaft will determine where the stress will be concentrated. Abrupt transitions, holes, slots are all examples of geometry which will impact the overall strength of the shaft. Production irregularities such as scratches, impact damage, and cracks will also shift the concentrations points of the stress. To keep the structure of the shaft, it is important that the stress does not exceed the maximum allowed stress, however, features like holes and slots are crucial to the design of the shaft in some cases. This means that the shaft needs to be analysed with these features and dimensioned to accommodate them or rework the design by reducing the impact of the stress concentration [12, p. 77]. Figure 5 shows a shaft which has a section with increased diameter; it is possible to see how the stress is concentrated around the fillet, which is the only stress concentration geometry on this shaft.

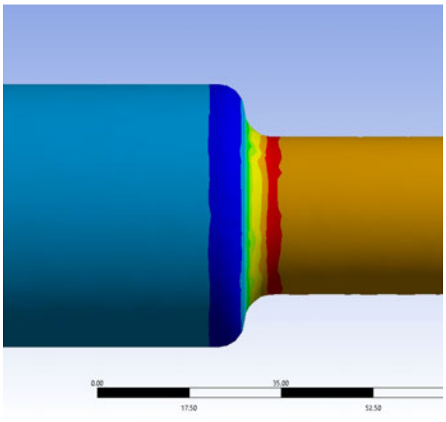

Figure 5: Stress concentration in a shaft

3.4.1 Stress concentration factors

Det Norske Veritas and Germanischer Lloyd (DNVGL) has determined which geometrical features has the greatest impact on the structure of a shaft typically used in marine application as; Shoulder fillets and flange fillets, U-notch, step with undercut, shrink fits, keyways, radial holes, longitudinal slots, splines, and square grooves (circlip). For this thesis, it is particularly important to note that shoulder and flange fillet, u-notch, radial holes, and longitudinal slot, are geometrical features that are key to the functions required by the oil distribution shaft.

As Wärtsilä's current design utilises longitudinal slots for their feedback mechanism, it is interesting to determine what the impact of a longitudinal slot is compared to a radial hole. Using formulas found in DNVGLs guidelines [13, pp. 34-40], it is possible to calculate the stress concentration factor of each of these geometrical features and this will give a great indication on which features should be used when proposing a new concept. It should be noted that these calculations do not take into consideration the magnitude of load or torque applied on the shaft, this will be added in computer assisted studies later in this thesis.

$$\alpha_{t,\ slot} = \alpha_{t,\ hole} + 0{,}8\frac{(l-e)/d}{\sqrt{\left(\left(1-\frac{d_i}{d}\right)*\frac{e}{d}\right)}} \tag{1}$$

$$\alpha_{t,\ hole} = 2{,}3 - 3\left(\frac{d_h}{d}\right)^2 + 10\left(\frac{d_h}{d}\right)^2\left(\frac{d_i}{d}\right)^2 \tag{2}$$

$$\alpha_{t,\ hole1} = 2{,}3 - 3\left(\frac{53}{500}\right)^2 + 10\left(\frac{53}{500}\right)^2\left(\frac{120}{500}\right)^2 = 2{,}273 \tag{3}$$

$$\alpha_{t,\ slot} = 2{,}273 + 0{,}8\frac{\frac{(369-50)}{500}}{\sqrt{\left(\left(1-\frac{120}{500}\right)*\frac{50}{500}\right)}} = 4{,}124 \tag{4}$$

$$\alpha_{t,\ hole2} = 2{,}3 - 3\left(\frac{50}{410}\right)^2 + 10\left(\frac{50}{410}\right)^2\left(\frac{242}{410}\right)^2 = 2{,}307 \tag{5}$$

By replacing Wärtsilä's current longitudinal slot with a radial hole Ø50mm, there is a significant decrease in stress concentration factor from 4,124 to 2,307. This heavily indicates that if it is possible to make a feedback mechanism that can function through a radial hole, it will be very beneficial for the structure of the shaft. The current design has an increased diameter around this section to combat the increased concentration of stress, by reducing the stress concentration factor, it should be possible to also reduce the diameter of this section. This means that it is possible to remove the remaining stress concentration factors of the shaft, excluding the flange fillet and radial holes used for feedback and actuation. The flange fillet currently used has a fillet radius of 45mm, which gives a stress concentration factor of 1,4867, furthermore, by keeping the same radius on the Ø410mm shaft this factor is reduced to 1,3757. This is mainly due to the combinations of small relative flange thicknesses (t/d) and small relative fillet radii (r/d) as e.g. (r + t)/d < 0.35 the α_t increases. This shall be taken into account by multiplying α_t with 1 + (0.08 d/(r + t))^2. [13] So, by decreasing the diameter of the shaft, the flange fillet also become less of a stress concentration point.

$$\alpha_{t,\ 500} = 1 + \frac{1}{\sqrt{6{,}8\frac{r}{D-d} + 38\frac{r}{d}\left(1 + 2\frac{r}{d}\right)^2 + 4\frac{d}{D}\left(\frac{r}{D-d}\right)^2}} * \left(1 + \left(0{,}08\frac{d}{r+t}\right)^2\right) \tag{6}$$

$$\alpha_{t,\ 500} = \left(1 + \frac{1}{\sqrt{6{,}8\frac{45}{770-460} + 38\frac{45}{460}\left(1 + 2\frac{45}{460}\right)^2 + 4\frac{460}{770}\left(\frac{45}{770-460}\right)^2}}\right) * \left(1 + \left(0{,}08\frac{460}{45+100}\right)^2\right) = 1{,}4867 \tag{7}$$

$$\alpha_{t,\ 410} = 1 + \frac{1}{\sqrt{6{,}8\frac{r}{D-d} + 38\frac{r}{d}\left(1 + 2\frac{r}{d}\right)^2 + 4\frac{d}{D}\left(\frac{r}{D-d}\right)^2}} \tag{8}$$

$$\alpha_{t,\ 410} = 1 + \frac{1}{\sqrt{6{,}8\frac{45}{770-410} + 38\frac{45}{410}\left(1 + 2\frac{45}{410}\right)^2 + 4\frac{410}{770}\left(\frac{45}{770-410}\right)^2}} = 1{,}3757 \tag{9}$$

As seen in appendix 9; when using a radial hole, the stress concentration factor will be more stable when changing the diameter of the shaft. This is beneficial when scaling the unit, as the required geometry of the functionality can stay mostly the same when using a radial hole.

3.4.2 Finite Element Method

Finite element method was used to conduct a static structural study on how the geometry affects where the maximum von-mises stress equivalent in the shaft will occur, and at which magnitude. All the studies in this thesis concerning maximum stress uses the same boundary conditions; 401,1kNm torque applied on the flange, and a fixed support on the surface of the cross section. There were also some studies where loads also included the weight of the bearing and housing, but this will be specifically mentioned if it is the case. The student license only allows for a total of 50 000 nodes and elements with a 32 000 cap on nodes. Because of the limitations of the student license, it is crucial to have an efficient mesh, and to concentrate the nodes and elements around the areas which is presumed to have the highest stress. It is also important to exclude node-hubs because this could create a singularity where the average stress of the surrounding elements indicates of false stress in these nodes, usually higher than expected. To avoid these issues, a hexahedral mesh was used around the critical geometry of the shaft as this type of mesh is very efficient and easy to avoid singularities.

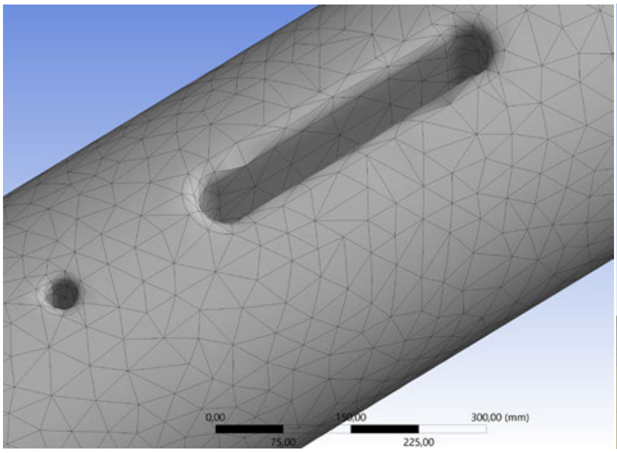

Figure 6: Meshing before rework

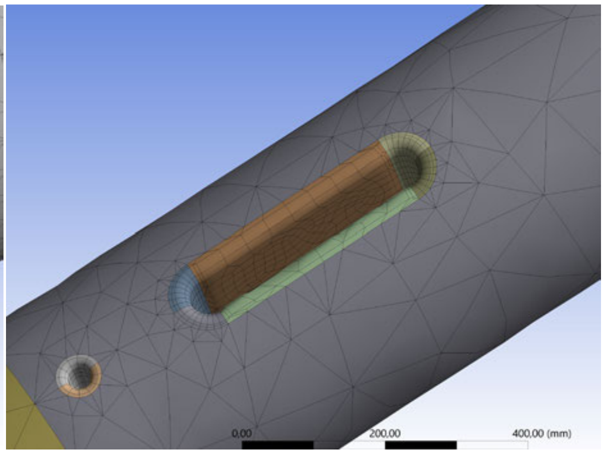

Figure 7: Meshing after rework

The first finite element method study mainly consisted of two identical structural steel shafts with a diameter of 410mm, the difference being the feedback geometry. This will indicate how a slot affects the structural integrity of the shaft compared to a hole, and which geometry should be used to reduce the shaft diameter from 500 mm to 410 mm which is the same diameter as the propeller shaft. As seen in Appendix 2 and 3, there is a significant higher value of von-mises equivalent stress in the slot compared to the hole, 225MPa in the slot and 115MPa in the hole. This reinforces the information gathered of stress concentration factors, and even with the limitations of the analysis, it can be concluded that a hole is significantly better to use in the shaft.

Finite element method was also used to find a reference point of maximum stress allowed in the shaft. The same method used in the last paragraph was applied to Wärtsilä's current shaft, as seen in appendix 1, and it is interesting to note that it is possible to see how the increased diameter and the fillets takes high stress to protect the slot, and there is a significant stress concentration in the slot on the side where the actuation channel is closer to the feedback slot. When further developing the shaft geometry, a maximum allowed stress of 115 MPa will be used as a reference point, this reference point is well within the fatigue limits of most variations of C45 steel which is currently used in a shaft.

3.5 Journal bearing

This part of the chapter looks at calculations of journal bearings, which is used in the RO-unit as load bearings to hold the housing and other components on the shaft, and for oil distribution into the shaft for controlling propeller pitch. For easier hand calculations there has been a simplification of the journal bearings. There will also be a CFD analysis done in Ansys Fluent.

3.5.1 Today's journal bearing geometry

Bearing geometry is in large depicted by the ability to transfer hydraulic oil in and out of the OD-shaft. Each of the two journal bearing inserts has groves in between two babbitts or white metal strips. This allows for distribution of oil into the shaft as well as provide oil to operate the journal bearings. The journal bearings have a L/D ratio of 0,248. The white metal strips have a small angle to them, this is a crucial design element to ensure stability in the journal bearing.

3.5.2 When to choose journal bearings

Journal bearings are generally acknowledged as a robust bearing solution in many applications. In between the journal and the bearing is a thin film of hydraulic oil supporting the radial load. Journal bearings have better surface separation then a roller bearing, this will normally lead to them being longer lasting than roller bearings, given normal circumstances. The bearing will experience metal to metal contact with the journal during start and stop, the reason is that the hydraulic film does not develop before there is a rotational speed of the journal. When stood still the journal will rest on the bearing surface, therefore, causing metal to metal contact. This together with contaminations in the fluid film, excessive vibrations, poor installation, and selection of bearing will affect the longevity of the bearing. For large applications, where rotational speed and vibrations gets higher the journal bearing is preferred as it has a dampening characteristic. Using hydraulic fluid, the heat generated in the bearing can be transported away.

3.5.3 L/D ratio

The bearings in the RO-unit does not carry any of the load from the weight of the propeller shaft and is not responsible for dampening vibrations in the propeller shaft. Length to diameter ratio determines a great portion of the bearing's characteristics. The ratio varies typically between 0,25-1. Diameter of the bearing is for the most part predetermined by the amount of torque and bending moment the shaft must handle. Therefore, it is the length that is subject to change for the designer of the bearing. When calculating L/D ratio, it is the active babbitt material that dictates the axial length of the bearing. Extra space for oil distribution is therefore not in this calculation. This factor is tuned to give the desired properties of that bearing. For example, a lower L/D ratio will have low damping characteristics compared to a larger ratio. If the ratio exceeds 0,75 it will show little to no gain in dampening. For a ratio under 0.3 there is poor dampening. For high radial loads the L/D ratio can exceed 1, this is however not a consideration for the bearings discussed in this report as they carry small loads [14]. In today's design there is two journal bearings with each having a L/D ratio of 0,248. These inserts have low dampening, but they are there mainly to distribute oil and support the housing for the feedback mechanism.

3.5.4 Material options

The supporting material for a journal bearing is usually steel, as this is preferred for its strength. The babbitt material itself is a white metal alloy which varies from place of use and from manufacturer to manufacturer. Often the exact alloy composition used by manufacturer is proprietary. The babbitt material is a softer material than the journal, this is to prevent damaging the journal during start and stop, since there will be metal to metal contact during this procedure. Another characteristic considered is that the dry friction between the journal and bearing is low so that there is less damage during start and stop.

3.5.5 Journal bearing design

To do the hand calculations the journal bearings was simplified. The geometry of the actual bearing is complex, so this was change to a smooth sleeve as seen in Figure 8. Calculations was simplified further by removing the oil filled cavities.


Figure 8: Simplified journal bearing

Simplification method removes the gap in between the two white metal parts of the bearing. Calculations is then done looking at the two parts as one large bearing with an inlet in the middle. This is combined with the other simplifications mentioned above. This chapter will look at the journal bearing design method. This is to get an understanding of how they are sized when looking at today's design. When designing the journal bearing some desired functions must be determined. The first calculation was calculating the "Sommerfeld" number. From that number a handful of graphs can be used to determine various design properties like, maximum pressure, flow of leak, etcetera. See appendix 6 for the various graphs used in this part of the report.

The graph for oil characteristics in terms of heat and dynamic viscosity is shown with SAE oil. SAE20 is an equivalent oil to VG-68. For the calculations done in this chapter SAE20 is therefore used. The following calculations is for maximum oil temperature, 60℃.

$$S = \left(\frac{R_j}{C_r}\right)^2 * \mu \frac{n}{P_{projected}} \tag{10}$$

$$S_{60} = \left(\frac{0{,}250\,m}{0{,}0001\,m}\right)^2 * 18\,mPa\,s * \frac{2{,}5\,rps}{127\,kPa} = 2{,}2$$

The calculation of Sommerfeld number can then be used in graphs found in appendix 6 for further calculations regarding bearing properties.

$$\frac{h_0}{C_r} = 0{,}54 \rightarrow h_0 = 0{,}054\,mm \tag{11}$$

Equation (11) calculates the minimum film thickness with the help of a graph. This can then be used to calculate the eccentricity ratio, shown in equation (12).

$$\frac{h_0}{C_r} = 1 - \varepsilon \rightarrow \varepsilon = 0{,}46 \tag{12}$$

$$\frac{R_j}{C_r} * f = 50 \rightarrow f = 0{,}02 \tag{13}$$

$$\frac{P_{projected}}{P_{max}} = 0{,}38 \rightarrow P_{max} = 334\,kPa \tag{14}$$

Equation (14) calculates the maximum pressure in the hydraulic film.

$$\frac{Q}{R_j C_r n l} = 4{,}6 \rightarrow Q = 2{,}14\ \frac{dm^3}{min} \tag{15}$$

$$\frac{Q_s}{Q} = 0{,}62 \rightarrow Q_s = 1{,}35\ \frac{dm^3}{min} \tag{16}$$

Equation (16) calculates Q_s which is the sideways flow. This sideways flow can be assumed to be the flow of leakage from the bearing. For the bearing to have enough hydraulic fluid this amount should be replenished.

$$F_T = f * F_N = 0{,}02 * 7848\ N = 156{,}96\ N \tag{17}$$

The tangential force calculated in equation (17) is due to the friction forces in the hydraulic fluid. This force acts against the rotation of the journal. From the tangential force the torque that acts against the rotation of the journal can be calculated as shown in equation (18). From this the power loss due to the journal bearing is calculated in equation (19).

$$T = F_T * R_j = 157\ N * 0{,}25\ m = 39{,}24\ Nm \tag{18}$$

$$P_{loss} = T * \omega = 39{,}24\ Nm * \frac{5\pi}{s} = 616{,}4\ \frac{J}{s} \tag{19}$$

See appendix 7 for other values calculated for the journal bearing at different temperatures.

Parameters used in calculation		
Parameters	***Unit***	***Value***
Journal diameter, d	mm	500
Shaft speed, n	rpm	150
Angular velocity, ω	rad/s	5π
Radial clearance, C	mm	0,1
Length of white metal, L	mm	124
Projected pressure, $P_{projected}$	kPa	127

Table 5: Parameters used for journal bearing calculations and CFD analysis

3.5.6 CFD analysis

This CFD-analysis (Figure 9) is done with the same parameters as in the calculations above for SAE20 hydraulic oil at 60℃, see Table 5 for overview of parameters. The inner wall is set as a rotational wall at the shaft speed. The outer wall is set as a stationary wall. The line going through the centre is the inlet, and the lines on each side of the oil film is set as outlets. Gauge pressure at inlet and outlet is set at 0. Inlet velocity is set from Q_s calculated in equation (16) above. See equation (20) for calculation of velocity of the oil at the inlet.

$$Q_s = A * v \rightarrow v = \frac{Q_s}{A} = \frac{1{,}35 \, \frac{dm^3}{min}}{314{,}2 \, mm^2} = 0{,}07161 \, \frac{m}{s} \tag{20}$$

In Figure 9 the analysis there is a zone with negative pressure. This is the cavitation region. Where the flow pressure drops under the ambient pressure, a phenomenon called gaseous cavitation occur. The gases dissolved in the hydraulic fluid are released. The model does not take cavitation into consideration, that gives the sub ambient pressures in the model [15].

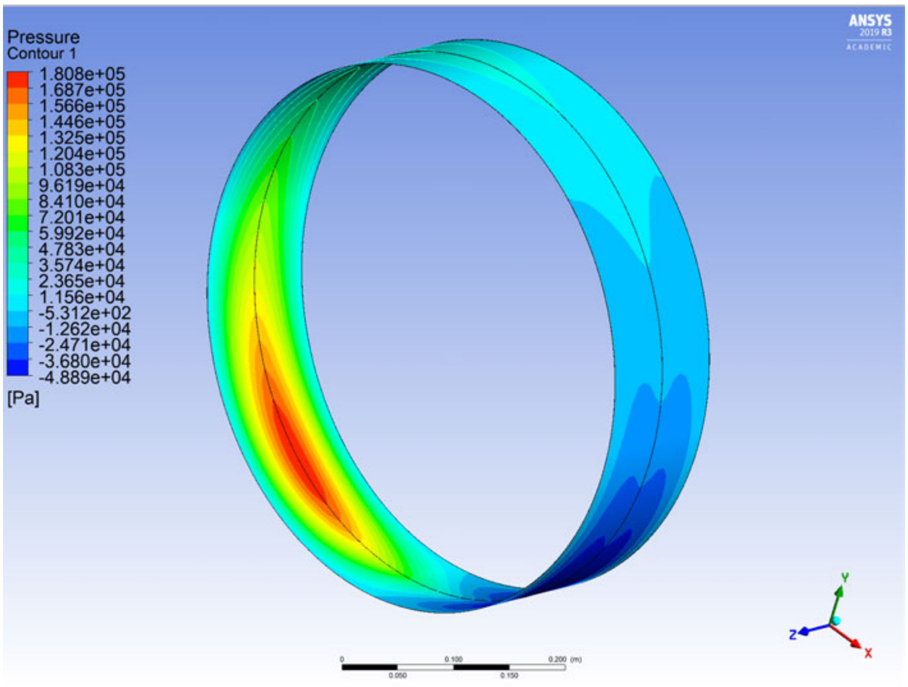

Figure 9: CFD analysis of journal bearing with velocity and gauge pressure 0

The red area is where there is the highest pressure in the hydraulic fluid film. This comes from the eccentricity ratio. The eccentricity comes from the applied radial load on the bearing and is counteracting that load. It does also change if the viscosity of the oil changes as seen in equation (10). A lower viscosity or higher applied radial load lowers the Sommerfeld number, and consequently, lowers the minimum film thickness and increases the eccentricity ratio.

The analysis done in Figure 10 have outside gauge pressure set to 250 kPa and without inlet velocity. Here the minimum pressure is above ambient pressure. There is still one high pressure zone and one low pressure zone. The reason behind the outlet pressure is that the housing of today's design is oil filled with absolute pressures under 350 kPa.

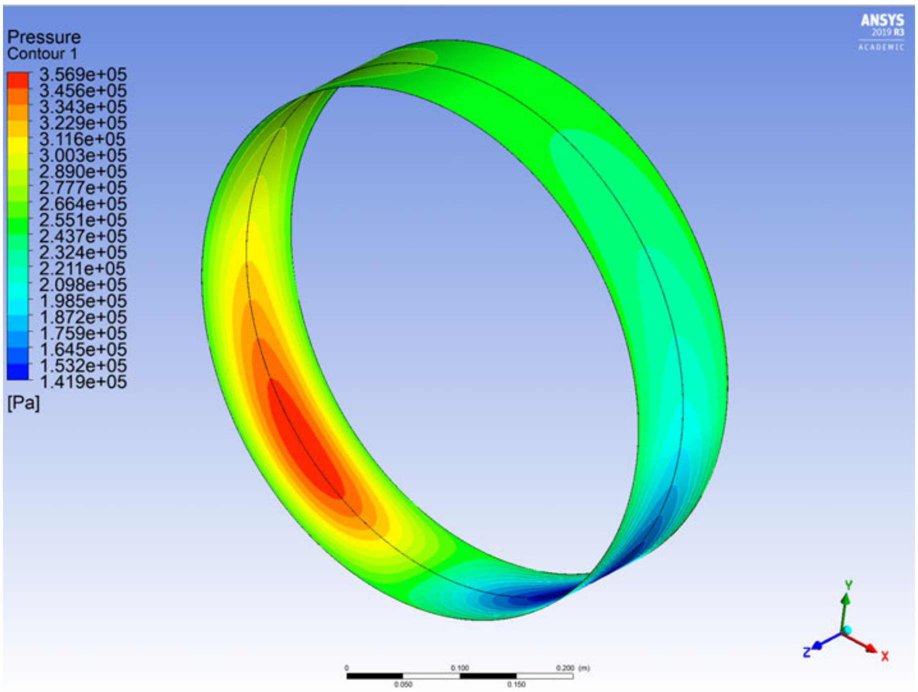

Figure 10: CFD analysis of journal bearing without velocity and gauge pressure set to 250 kPa

The analysis done in Figure 11 have a gauge pressure on the outlets set to 250 kPa, and inlet velocity as calculated in equation (20). The outside pressure comes from, as mentioned above, the pressure in the oil filled housing. In this analysis the minimum pressure is not under ambient pressure as the analysis seen in Figure 9.

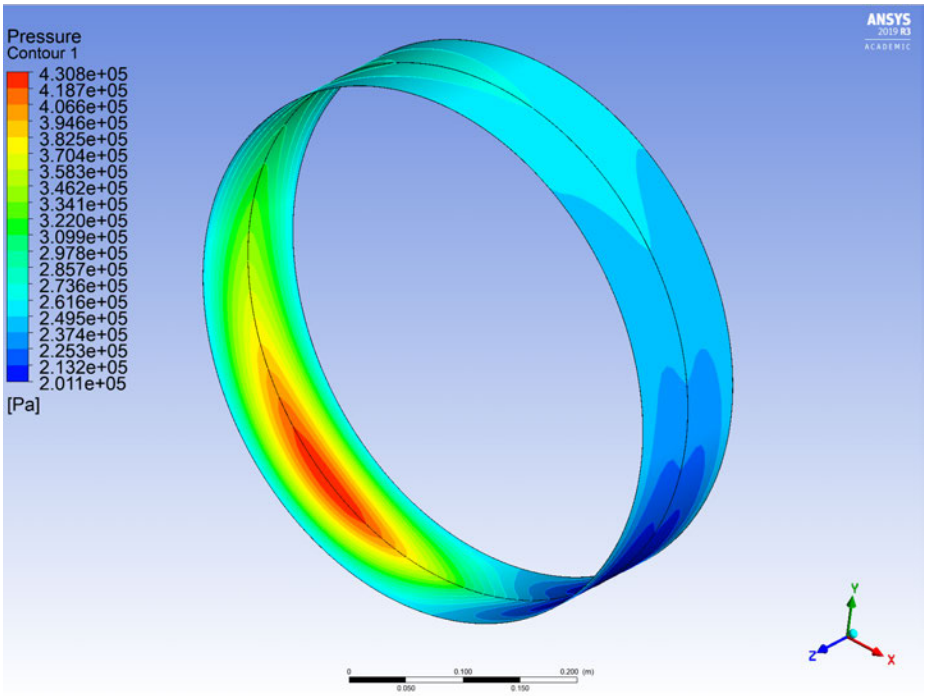

Figure 11: CFD analysis of journal bearing with velocity and gauge pressure 250 kPa

The goal with doing CFD analysis was originally to develop a method for doing CFD analysis on the journal bearings used in today's system. This however turned out to be too time consuming and led to simplifications to be able to do an analysis for this thesis.

3.6 Hydraulic actuation system

In the current design the ability to retain a specific pitch position is achieved with a blocking valve inside the feedback rod. This valve is an inhouse design from when the original RO-unit was developed. The valve is a pilot operated valve, meaning it allows free flow ahead only, unless the pilot pressure increases above the set value, then it opens and allows flow to astern. In this case the pilot port is connected to the opposite line so when one line is pressurised by the hydraulic power pack, or HPP for short, it releases the pressure from the other line. Having these installed allows control over the current propeller pitch, since the hydraulic oil controlling the pitch is retained in the shaft until the pilot port is pressurised allowing the oil to flow back.

HPP is responsible for generating hydraulic flow when a command is sent from the bridge, or the engine control room in some cases, to change the propeller pitch. There is also a proportional valve, which is a valve that provides a change in output flow or pressure, this valve allows one HPP to output different pressures to different parts of the hydraulic system, delivering high pressure to the activation of the OD-shaft and simultaneously lower pressure to the lubrication line.

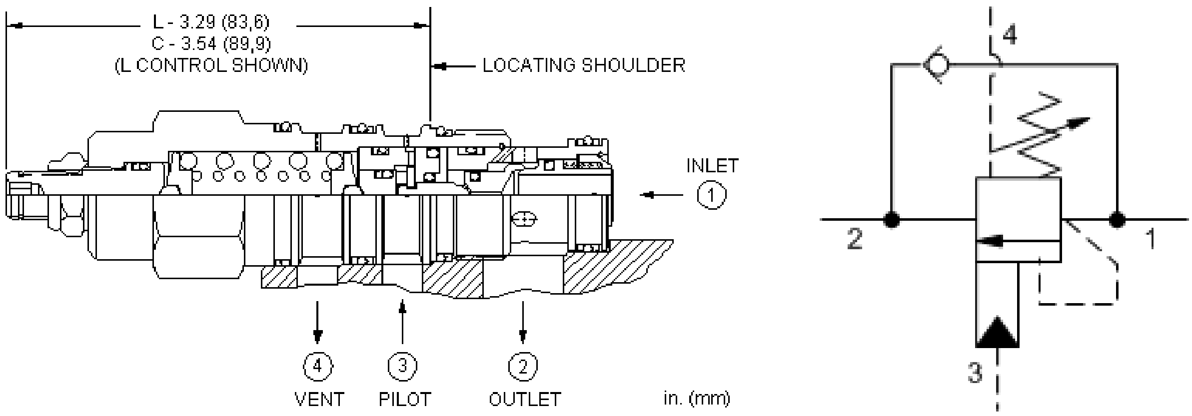

Figure 12: Counterbalance valve [16]

Figure 12 shows a SUN hydraulics vented counterbalance valve, these valves function by only allowing flow from inlet ① to outlet ②. However, if a pressure is applied to the pilot ③ this is switched and the media running through the valve can pass back from outlet ② to inlet ①. This valve also has a safety feature built into the valve, if the pressure on the outlet build above a set pressure this will also trigger the pilot and allow pressure release back into the inlet. In this case the pressure acting on the pilot will come from the opposing channel. The vent is also equipped with positive seals between the different ports, this means no leakage should appear, on reseating of the valve however there is a small leakage of a maximum of 0,3 cm^3/min, therefore a vent port ④ is added, this allows the leakage to dissipate from the vent. Without the vent port there would be a pressure build-up over time in the spring compartment and in the end freezing the valve, making it impossible to open. At the pilot port ③ there is also a pilot assist, this is added to lower the amount of pressure needed to open the pilot, this is a predetermined ratio. In this case it means that the valve opens before the opposing line has reached the maximum operating pressure, allowing for a smoother movement of the pitch. [16]

The cost of counterbalance valves is determined by flow capacity, meaning there is the possibility to use four valves with the same total capacity as two, without an increase in price. Having four valves instead of two creates a form of redundancy, so that in case of a malfunction in one of the valves, the vessel should still be able to retain control over the pitch and therefore manoeuvrability. Wärtsilä uses SUN hydraulics products in different parts of today's product line-up, meaning Wärtsilä has knowledge and contacts with SUN hydraulics. The valve in SUN hydraulics products that is most relevant for this thesis is CWEALHN vented counterbalance valve shown in Figure 12, this valve satisfies the need for flow, maximum pressure, and pilot setting (Appendix 9).

3.7 Class guidelines and standards to consider

In general, any commercial vessel must be compliant with a specific set of class rules. The supplier of equipment to a vessel must ensure compliance and the class society certifies the compliance for every commercial vessel. For this thesis, the DNV class society has been considered. Det Norske Veritas and Germanischer Lloyd (DNVGL) is an independent expert in risk and quality assurance. In their own words they are the leading classification society, and they are a recognized advisor in multiple fields, this includes the maritime industry. [17]

To ensure best practice and safe operations of vessels during their lifetime, DNVGL has certain rules and guidelines that must be followed in order to get approval. When a vessel follows DNVGL's guidelines it acts as a quality stamp that reaffirm customers and insurance companies that the vessel is safe. The guidelines that are applicable to this thesis are mainly:

404 Propulsion machinery orders from the navigation bridge shall be indicated in the main machinery control room and at the maneuvering platform. (SOLAS Ch. II-1/31.2.4) [18, p. 17]

5.13.4(a) Pitch indicators. A pitch indicator is to be fitted on the navigation bridge. In addition, each station capable of controlling the propeller pitch is to be fitted with a pitch indicator. [19, p. 303]

DNVGL also states the different shaft diameters for the different classes of vessels, for the thesis an existing design has been studied, in that design the shaft diameter is 410 mm, this diameter is then checked to be compliant with class rules for every individual equipment delivery.

3.8 Cost-drivers

This report's focus is to propose a concept of a solution for a new oil distribution system. However, one of the reasons Wärtsilä is looking at new solutions for this system, is to explore the possibilities of a different and possibly smaller and simpler system expected to have a reduced cost. So, to be able to propose a cheaper unit, it is important to know the different factors and parts that have the largest impact on the total cost of the unit. Looking at the current design, Wärtsilä has conducted internal studies and made a cost break-down of the unit and in general there are three big cost-drives: Servo-shaft, housing, and bearing inserts.

The servo-shaft is a substantial cost-raiser because of the material required, but the production cost is also quite significant because of the geometry of the current shaft. By reducing the diameter of the shaft and removing or changing the geometry, this cost could be reduced both in terms of material required, and general machining cost. This decrease of the diameter would also naturally decrease the cost of the bearing inserts, by reducing the size, but there seems to be opportunities to also lower the required load capacity needed from the bearing, i.e. removing the housing around the feedback mechanism entirely. Throughout this report, the parts mentioned above are the three main parts that will be considered when making cost-saving decisions.

4. Development process

This chapter will show the development process and explain the different concepts that were explored throughout the process. The development process used in this report is a very dynamic approach, meaning that the process is very open to new ideas and the drafts may seemingly not have a red thread from the first draft to the final proposal. This approach was used because of the great complexity of the unit, where practical knowledge and experience is very important to determine which solutions are pragmatic in terms of assembly and production. Discussions with Wärtsilä greatly impacted the route the concept development took, as each meeting they shared their experience which would lead to new ideas and interesting options to explore. It is evident when comparing the first drafts to the final proposal that there is little academic structure that guides the flow of development, however, to utilise the information and knowledge gathered from Wärtsilä, this approach was a pragmatic way of conducting the development process.

4.1 Design methodology

The dynamic development process is constrained by some requirements, to ensure that the final concept proposal is a viable and usable concept. By setting general goals and criteria of design before starting the first draft it was possible to have a structured way to revise and critique the different drafts. These goals and criteria were decided through talks with Wärtsilä, and internally in the group. Wärtsilä had some requirements for the feedback-mechanism, mainly that the mechanism had to be mechanical and there should be an option to read the position manually. This requirement also coincides with the class guidelines from DNVGL. The main goal set by Wärtsilä was a smaller, and simpler geometry on the shaft diameter, as this could lead to a more cost-efficient unit due to the reduction of bearing size and general machining cost of the shaft and various parts.

The internal goals consisted mainly of; simplicity, space efficiency, and reliability. The unit should be easily assembled and overhauled, although the greater goal was that this unit should not be the cause of major overhaul of the propulsion system on board a vessel. To accomplish this, a simplistic method was employed; if there are fewer parts that endured stress and movement there are fewer things that could go wrong. A ground-up approach was used to build the first draft as simple as possible, only the bare necessities were included in the draft to make the unit function. This also gave a greater understanding of the current design method, as the draft was carefully built in a simple fashion. This draft route also leaves room for the dynamic approach, where new knowledge and feedback can easily be incorporated.

A critical part of the unit is the shaft geometry, concluding which geometry is the most expedient for this unit is important as this could lead to a very space and cost-efficient shaft. Analysis done in chapter 2.4 geometry; showed that the critical factor was the geometry around the feedback mechanism, and the shaft diameter. This indicates that a solution to the feedback mechanism should be able to function through a round hole around Ø50mm, which means the shaft diameter could be significantly reduced. Analysis also showed that the shaft takes most of the stress on the outskirts of the perimeter, this factor gives more room for parts inside that shaft.

To ensure that these requirements and goals are accomplished, every draft will be critiqued using a pros-and-cons list. This list will include paraphrased feedback from Wärtsilä and general statements on each concept.

4.2 Revision of current design

The current shaft that is being used for the RO-unit is a variable diameter shaft, meaning that its diameter is increased to counteract, or handle, the stress incurred in the feedback-slot. When redesigning the RO-unit one of the goals should be to make it a constant diameter across the board, and to do this it is necessary to change the geometry of the feedback slot. To illustrate the impact of the diameter, a simple analysis was conducted in ANSYS; as seen in appendix 1 through 4, the difference in the stress is very much connected to the diameter and the geometry of the feedback slot.

By changing the geometry of the feedback slot, the preliminary analysis showed a decrease in stress concentration in the slot, which indicated that a smaller slot should result in the change of the diameter without exceeding the maximum stress allowed. For the purpose of this deduction, the maximum stress allowed is determined to be 115 MPa, originating from the preliminary analysis conducted in ANSYS, however, as seen in chapter 2.2.2 Finite Element Method (FEM) the limitations of the student licenses means that this is probably the maximum stress in reality. The method used in ANSYS to get the best result possible, was to create a mesh concentration around the critical geometry. Having close to 90% of the nodes around the geometry which takes most of the stress, the result becomes more accurate and reliable.

Another big cost section of this RO-unit is the bearing mechanism, and the parts that comes with this. This could potentially be simplified to reduce the materials needed, this will also affect the weight which in turn affects the load capacity needed by the bearing, which means there is possibilities making a very cheap unit comparably. The housing could potentially be removed, if the feedback mechanism is made to not require an oil bath, however, this comes with its costs as the most feasible ways to accomplish this would be through the use of sensors or similar, which is currently an expensive option. So, one solution would be to find a cheap alternative.

The blocking-valve in this design helps maintain the pressure in the system, which decreases the power-consumption of the RO-unit by removing the need to constantly run power-packs to maintain the pressure, however the current valve that is used could use some upgrades. Property of Wärtsilä [redacted] The placement of the valve also makes it hard to run diagnostic and overhaul, because it is placed inside the shaft and might require dry-dock.

The current design of the RO-unit gives the impression that it rarely fails mechanically, and that longevity is key to the design. It does however give an impression that it is unnecessarily bulky and complex. The feedback mechanism requires a variable shaft diameter, large housing supported by two bearings, and an additional bearing inside the housing. So, the first step towards revising the design was to look at the core functions of the unit and ask simple questions of what parts you need to carry out the required functions. To make this easier, it was decided to split the functions of the unit in to two categories: Oil distribution, and feedback mechanism.

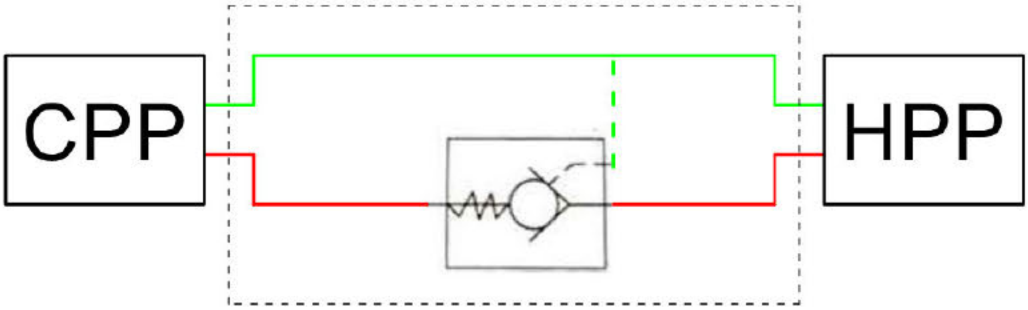

Figure 13: Hydraulic scheme of today's unit

When looking at the oil distribution, a hydraulic scheme (Figure 13) was created to get a sense of what was needed in terms of piping, channels, and valves. The system is fairly simple, there are two main lines that has variable flow to change the pitch position according to said flow, or pressure. A valve is also needed to maintain the pressure in the system, by maintaining the pressure it will reduce the power consumption because the power pack does not have to continuously compensate for hydraulic fluid leaving the system. In terms of key functions; the oil distribution needs two actuation points, one for ahead and one for astern, lubrication channels that runs through the shaft towards the propeller hub, a channel for manual pitch setting by hand held power pack, supply hydraulic oil for the oil film, and a valve to maintain pressure in the system. The valve should also have a safety release, which activates when the pressure in the system exceeds maximum pressure allowed to protect the system.

The feedback mechanism measures the position of the hydraulic line which pivots the propeller blades, this means that the position of the blade is directly connected to the position of the hydraulic line. The current design uses a radial rod mounted on this line to transfer the movement to the outside of the shaft, the rod is connected to a sliding ring bearing – mechanism which in turn transfers the axial movement to rotational movement, which is static on the housing. This solution is what makes the shaft have a variable diameter, requires a lot of space in terms of housing, and an additional bearing. The goal when revising this mechanism was to make it space efficient both in terms of shaft diameter and housing, reliable, and accurate. The first ideas were based on separating the two core functions of the RO-unit and read the feedback in an oil-free environment, this could potentially mean that the housing required by the last solution could be removed entirely.

4.3 Drafts

When starting to design a revised concept, it was decided to do the bulk of drawing in three stages. The first stage should be very concept based; the unit would not necessarily have to function properly, but it should give a concept which could be developed further. [13] This draft was then presented to Wärtsilä and discussed, so that Wärtsilä could assist in which concept ideas they would be interested in and could be feasible. The feedback given by Wärtsilä was then incorporated into the design, including some new ideas to be presented to Wärtsilä. The third draft will build on the other concepts, and implement the information and knowledge gathered throughout the thesis. The final proposal will be an accumulation of the work in the thesis.

To evaluate the concepts a grading table was made, the table was used to look at each individual concept and evaluate to which degree the concepts achieves the goals and requirements set in chapter 4. It should be noted that this table will be used mainly for internal concept development, and the grades will be very subjective as the premise for the grading is the internal goals and requirements, and not necessarily a grade which indicates how the concepts will work in reality. Because of the subjective grading it is difficult to not compare earlier designs and grade the concepts accordingly.

Draft #		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	The complexity of geometry.	
Feedback system	Evaluate precision, reliability	
Oil distribution	Simplicity and functionality of oil distribution.	
Design robustness	Evaluation of the longevity of the system.	
Assembly	The complexity of assembly.	
Sum	Total score of design (out of 25)	**0**

Table 6: Concept evaluation table

4.3.1 Draft 1 – Concepts

The main idea behind draft 1 was to separate the two functions of the unit. The oil distribution function was created by using a hydraulic scheme to place the necessary parts and channels which is required for the unit to function. However, the detailing of the design was kept to a minimum as this was just an illustration of the concept. As seen on Figure 14, the oil distribution is almost entirely separated from the feedback mechanism which meant that it was possible to allocate space to the feedback mechanism without infringing on the design of the oil distribution. This meant that the oil distribution would have no effect on the dimensions of the shaft, which is one of the main design goals. The feedback mechanism was then created separately from the oil distribution, however, because of the separation it was important that neither of the concept had to be entirely compatible as this could lead to good ideas being lost due to incompatibility. The thinking behind this was basically that whatever the concept was, it could be changed to be compatible – if not, the design could be altered to make ideas applicable.

Draft 1.1

As the current RO-unit that Wärtsilä uses has a shaft geometry that impacts the bearing diameter and length, the first concept tried to rearrange the placement of the functions. This rearrangement was made possible because of the idea to separate the two main functions of the RO-unit. The shaft geometry is still not optimal; however, the impact of the geometry is less severe than the current unit because it is possible to redesign the bearing and housing. This would reduce two out of three major cost-raiser. The feedback mechanism uses a mechanical solution to get the position of the feedback rod from the inner shaft to the outside of the shaft, where it would be measured by a stator unit similar to what Wärtsilä currently uses for other oil distribution systems.

Figure 14: Draft 1.1 nomenclature

Draft 1.1		
Criteria	***Description***	***Score (1-5)***
Complexity drivers	Diameter variation, multiple slots and angled hole for oil.	2
Feedback system	High precision, but in need for a sensor. Mechanism in it self is good, but the consequence is larger diameter at the feedback mechanism.	4
Oil distribution	In the current draft the distribution would not work. Does not have emergency pitch separate from journal bearing. Missing valves. Have potential.	2
Design robustness	Few vulnerable parts.	4
Assembly	Hard to assemble in the state of the draft, as the feedback mechanism is not fully developed.	2
Sum	Total score of design (out of 25)	**14**

Table 7: Draft 1.1 evaluation

Draft 1.2

This draft builds on draft 1.1 by using a different solution for the feedback mechanism it is possible to further change the geometry of the shaft. Having a feedback mechanism that does not affect the shaft geometry would generally be a great solution, however, the way this is accomplished is very important. The feedback mechanism in this design uses an optical sensor to measure the position of the feedback rod inside the shaft, and transmits the result out of the shaft through cables – the sensor would then need a way to transmit this signal to a stator-unit, i.e. Wi-Fi-, radio-, radio-frequency identification (RFID) signal. A solution like this would also require certain redundancies to protect the unit from unexpected situations like blackout and similar. Which means that it would require a battery pack that should be possible to charge. This is judged to make the feedback mechanism more expensive than current design, and even though it reduces the cost of the other cost-driver, it could create a new – quite large – cost driver.

Property of Wärtsilä

Figure 15: Draft 1.2 nomenclature

Draft 1.2		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	same diameter for the whole shaft, still has angled hole for oil.	3
Feedback system	Based on sensors as main feedback system, does not fulfil requirement of mechanical feedback, and sensors of this kind is expensive.	1
Oil distribution	In the current draft the distribution would not work. Does not have emergency pitch separate from journal bearing. Missing valves. Have potential.	2
Design robustness	Few vulnerable parts, but maintenance of sensor if needed could be challenging.	3
Assembly	Might be hard to assemble sensor.	3
Sum	Total score of design (out of 25)	**12**

Table 8: Draft 1.2 evaluation

Draft 1.3

In the current design there is a large slot. With the idea in shown in Figure 16 the slot is replaced by having a feedback rod going horizontally from one chamber and through to the feedback ring. Diameter of the shaft can then be reduced where the feedback ring is located, and somewhat reduced at the journal bearing inserts diameter. Functionally the feedback disk is connected to the feedback pipe which moves with change in propeller pitch. This axial motion is then transferred to the feedback ring via the feedback rod. The rest of the feedback mechanism is discussed in closer detail in chapter 3.2 as this is the same for this draft as the original design. The thinking here was that a large portion of today's shaft components could be reused in this new design. Parts such as housing, feedback mechanism, and journal bearing inserts seen in Figure 3.

This is however a complex shaft design. Where the feedback disk is located, the shaft diameter is increased by a fair amount. The coupling size depicted in Figure 16 is not to scale for what is needed to transfer the torque. It would be a much larger coupling and is therefore a design flaw of this draft. To counteract this, it was discussed if the feedback disk could be located at the opposite side of the flange.

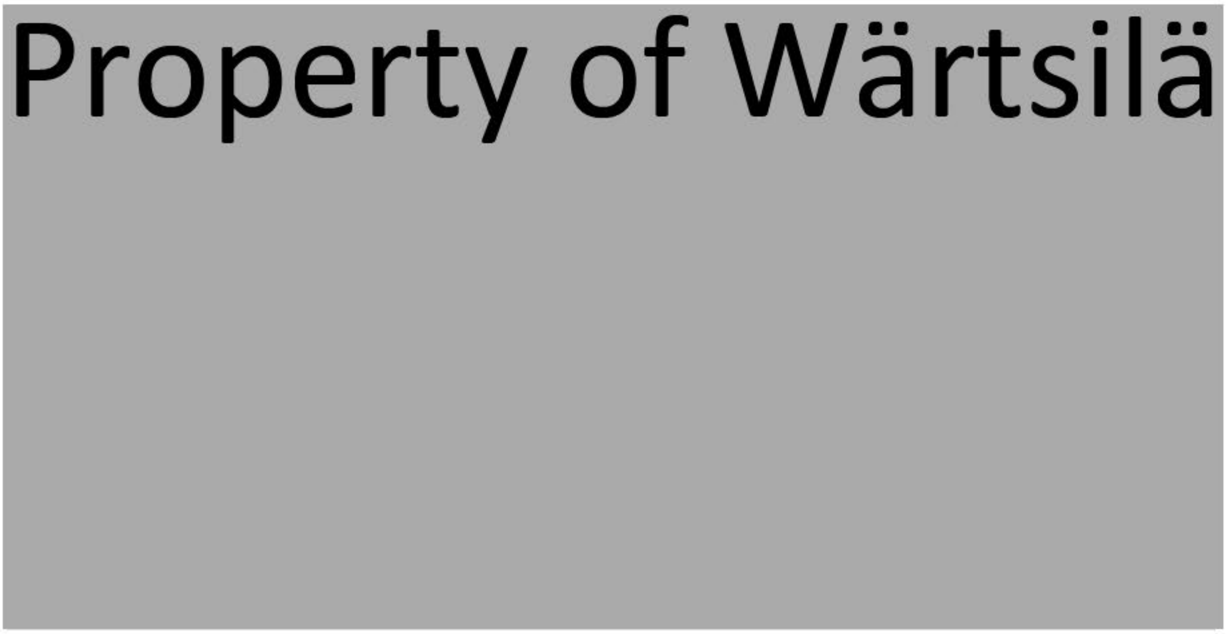

Figure 16: Draft 1.3 nomenclature

Draft 1.3		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	High complexity of shaft geometry	1
Feedback system	Possibility for jamming and requires correct assembly and high tolerances. It is an interesting idea, but in reality hard to accomplish.	2
Oil distribution	Focus has not been allocated to oil distribution in this draft. In current state it would not work.	N/A
Design robustness	Feedback-disk and -rod could be sources of failure.	2
Assembly	Requires correct assembly and high tolerances.	1
Sum	Total score of design (out of 25)	**6**

Table 9: Draft 1.3 evaluation

Draft 1.4

This draft uses the same basic concept as draft 1.3.Draft 1. The difference here is the separation of oil distribution and feedback mechanism, as this was one of the main goals with the new design. In this design the feedback ring and adjusting lever would be lubricated with grease instead of using the hydraulic oil, the method used in the original design. Here the housing is a tool for placement of the feedback mechanism, there is no oil filled chamber here.

With this design the same design problematics occurred as in draft 1.3. One additional drawback with the design is that the feedback movement happens at two places. One time at the feedback disk and another at the feedback ring.

Property of Wärtsilä

Figure 17: Draft 1.4 nomenclature

Draft 1.4		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	High complexity of shaft geometry, but slightly better then draft 1.3.	1
Feedback system	Possibility for jamming and requires correct assembly and high tolerances. It is an interesting idea, but in reality hard to accomplish.	2
Oil distribution	Focus has not been allocated to oil distribution in this draft. In current state it would not work.	N/A
Design robustness	Feedback-disk and -rod could be sources of failure.	2
Assembly	Requires correct assembly and high tolerances.	1
Sum	Total score of design (out of 25)	**6**

Table 10: Draft 1.4 evaluation

Draft 1.5

Conceptual design revolving around the wires pulling on springs where it is possible to measure the distance the spring is activated. In reality, the spring would have to be quite long to accommodate the travel range needed to cover the feedback rod whole range of motion (300mm) so this solution could require some pragmatic solutions like a coil spring to get a more linear force-curve and reduce the travel distance of the "pull-back"-mechanism. Furthermore, the structural integrity of the fillet on the flange is also compromised in this concept, so the mounting point of the spring would also require a different solution.

Property of Wärtsilä

Figure 18: Draft 1.5 nomenclature

Draft 1.5		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	Shaft geometry is low in complexity for feedback mechanism.	4
Feedback system	Low precision. In reality the spring would be much longer to achieve desired function.	1
Oil distribution	Focus has not been allocated to oil distribution in this draft.	N/A
Design robustness	Jamming of feedback ring is a possibility.	2
Assembly	Should be relatively easy to assemble, but could be difficulties with installing the wire.	3
Sum	Total score of design (out of 25)	**10**

Table 11: Draft 1.5 evaluation

4.3.2 Draft 2 – Revision and implementation of feedback

After presenting the first draft to Wärtsilä, the feedback was implemented into the design. Feedback mechanism was the greatest subject of change, the new draft took the idea of a sliding bearing to transfer the axial movement of the feedback rod, to an axial movement on the sliding bearing. To do this, a sprocket rod would be the transfer medium - where one side would be connected to the feedback rod by a rack gear which would rotate the sprocket rod and this rotation would then be used as the driving force of the feedback ring by means of a belt mechanism. However, the method used for transferring this movement through the shaft, was not satisfactory as using gears for this mechanism is not expedient. Gears could slip, break, and the precision of the gear is not very good as most gears require some lashing which would naturally affect the precision. Furthermore, there was also some changes to the oil distribution unit, the housing and bearing was changed to be one part compared to the first draft, reducing the material and machining cost, but the idea remained very much the same as draft 1.1 and draft 1.2.

Property of Wärtsilä

Figure 19: Draft 2 nomenclature

Draft 2		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	Low complexity of shaft. Some complexity of feedback system.	4
Feedback system	Gearing could be source for lower precision. Could also potentially damage teeth of rack or pinion gear. Is not space efficient.	2
Oil distribution	In the current draft the distribution would not work as it is missing valves. Have potential.	2
Design robustness	Teeth of rack and pinion gear is vulnerable.	2
Assembly	difficulties in assembly of lining up gearing correct.	2
Sum	Total score of design (out of 25)	**12**

Table 12: Draft 2 evaluation

4.3.3 Draft 3

Further implementation of feedback from discussions with Wärtsilä is incorporated in to draft 3, the feedback mechanism is somewhat altered to be more pragmatic and precise, however, this draft also focuses very much on the oil distribution function. Placing the counter-balance valves turned out to be more elaborate than expected and a pragmatic solution was needed to get a satisfactory placement and functionality. To make a more space efficient proposal, the actuation channels were angled to enter the shaft "in-line", this enabled the possibilities of making a part that combined several different functions in a small package. The valve box functions as a separation seal between the ahead and astern lines, the valves maintains the pressure in the system and the design of the valve box includes channels for piloting between the valves. This draft does not include the venting channels required by most vents of this flow and load, but there are many possibilities to include these channels.

Some changes were also made to the geometry of the bearing, because of the decrease in weight of the insert, the load capacity of the bearing could be greatly reduced, preliminary calculations showed the load was almost reduced by 60%. The bearing is also currently designed with a L/D-ratio of around 0,25 and given the reduction in diameter, Ø500 to Ø410, it is possible to use a much smaller insert and bearing. A problem that was encountered with this draft was the geometry of the actuation channels in the shaft, as seen in appendix 5, the angle needed to get the "in-line" actuation creates sharp edges on the perimeter of the shaft which creates points of high stress. To circumvent this problem, a rounder geometry of the channel could be used – however, because the shaft needs to be forged to be strong enough, this could be very problematic, or almost impossible, to accomplish with a cost-efficient material. Analysis showed in appendix 5 indicates that this solution does not have its highest point of stress in the actuation channel, but rather in the feedback channel.

Property of Wärtsilä

Figure 20: Draft 3 nomenclature

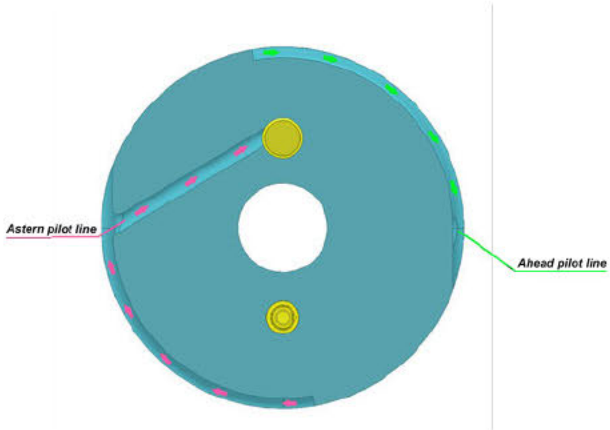

Figure 21: Valve insert nomenclature 1

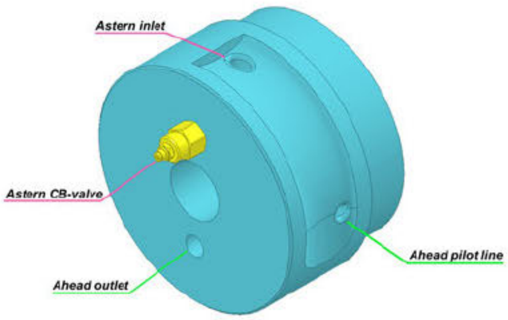

Figure 22: Valve insert nomenclature 2

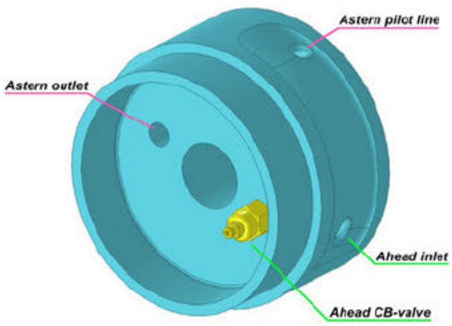

Figure 23: Valve insert nomenclature 3

Because of the valve-insert design, some practical problems were pointed out by Wärtsilä. It could prove hard to accomplish a seal between the actuation channels around the circumfurence, and the actuation lines going to the propeller hub. To get the seal, production and assembly methods like freeze-fitting, high-tolerance production, and other methods that is recommeneded to avoid, would have to be used. To avoid these methods, changes could be made to the design – in particular the valve insert. Wärtsilä suggested that instead of the in-line actuation, that perpendicular radial holes could be used and having two channels around the whole insert, which would mean o-rings could be used to maintain a seal between the actuation channels and lines. This would however increase the length of the insert, which in turn would mean that the geometry of the insert would have to be changed to avoid a long section of contact between the ahead actuation line and valve insert. So to fix certain problems with the design, some of the benefits would no longer be there.

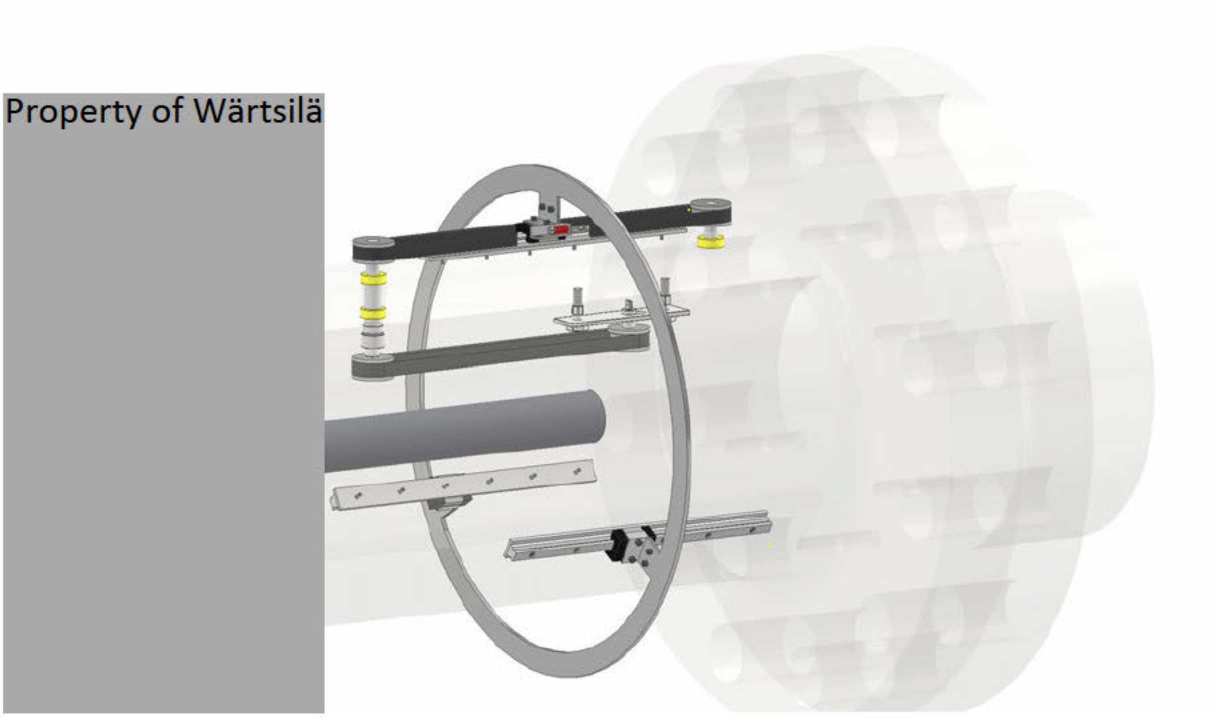

Figure 24: Draft 3 feedback mechanism

Draft 3		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	Complexity in terms of angled holes that causes stress concentrations. Two holes for attaching feedback mechanism. High tolerances in oil distribution.	3
Feedback system	Requires correct instalment to achieve precision. Has two holes for pulleys.	3
Oil distribution	Good on paper but tolerances needed for sealing results in difficult instalment. Missing lubrication line.	3
Design robustness	Valves have safety release, which gives safety if unwanted pressures arise. Feedback mechanism could be a weakness.	3
Assembly	High tolerances needed in oil distribution, makes for difficult instalment. Maintenance of valves is source of difficulties.	2
Sum	Total score of design (out of 25)	**14**

Table 13: Draft 3 evaluation

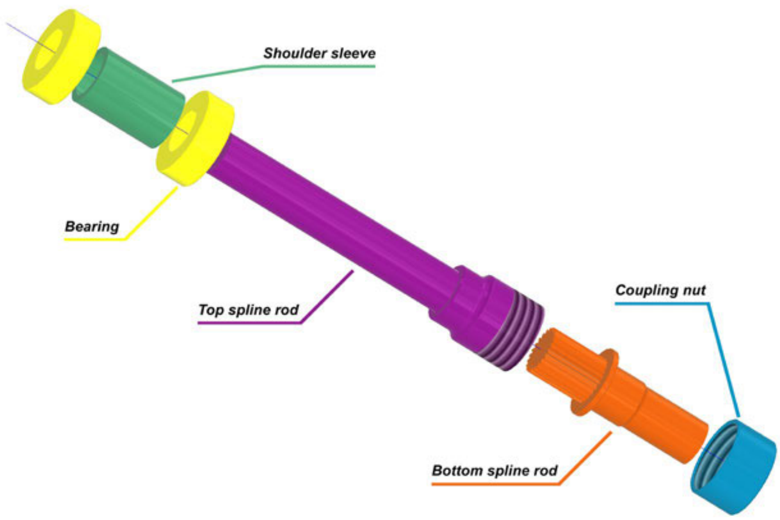

Figure 25: Motor shaft unit nomenclature

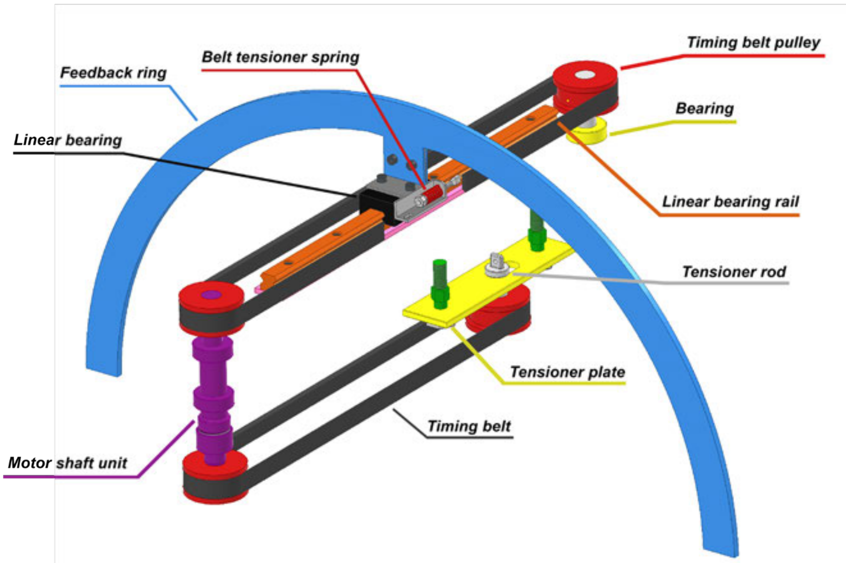

Figure 26: Draft 3 feedback mechanism nomenclature

The feedback mechanism builds on the same principals as draft 2 but implementing ideas that were discussed with Wärtsilä. Changing the driving mechanism of the linear bearing, previously a gear mechanism, to a belt driven rod means that the precision and reliability could potentially be improved. The mechanism connects two timing belts, one on the inside and one outside the shaft. Inside the shaft the belt is attached to the feedback pipe, and as this pipe moves back and forth with change in pitch, the belt rotates. This rotation is transferred outside the shaft via the motor shaft unit. The outside belt is drivven by the motor shaft, the linear bearing is then pulled either direction along the linear bearing rail. Where the belt is connected to the linear bearing is a system for keeping belt tension. To transmitt the feedback, there is a feedback ring attached to the linear bearing. The position of this ring is then read by a stationary sensor unit. The motor shaft unit ensures low friction rotation througt the shaft. It has a spline connection as a measure to get easy installation of the bottom motor wheel, as this wheel is press fitted to the bottom spline rod. The spline carries the torque, and the coupling nut holds the two rods together. This method removes the need for a pressfit done inside the OD-shaft when installing.

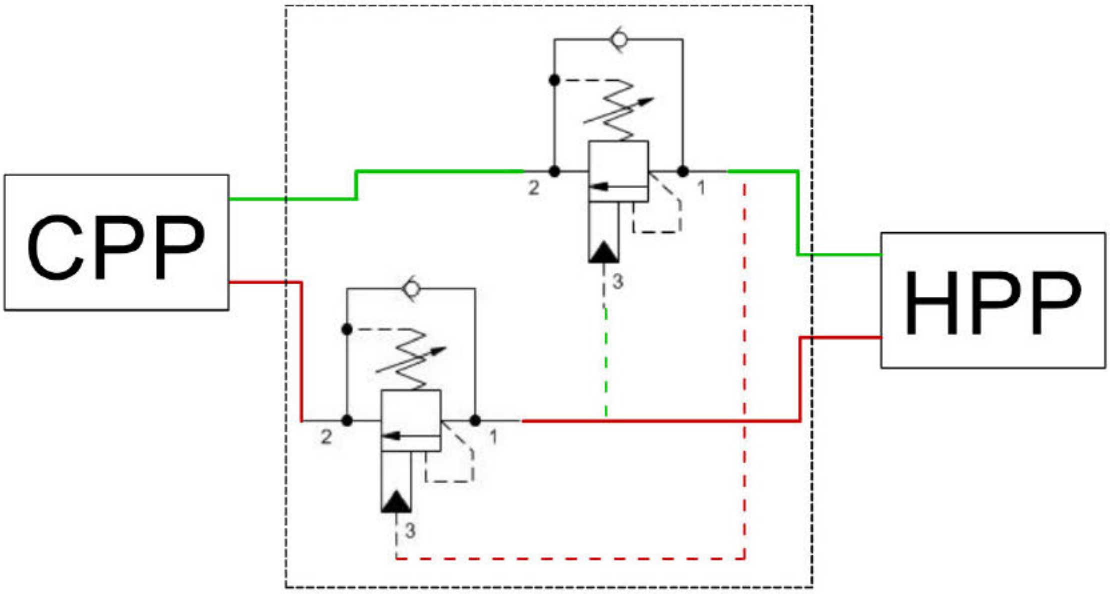

Figure 27: Hydraulic scheme for Draft 3

4.3.4 Final proposal

The final concept is an accumulation of all the research and knowledge gained throughout this thesis; where the concept is very similar to the third draft, however, some functions have been heavily revised. The oil distribution function of the third draft had some practical issues where the solution required would be to create a part which is very similar to something Wärtsilä already has explored. So, it was decided to go another route, where instead of using a valve-box or valve-insert, the valves would be placed in cartridges that preferably made by the same producer as the valves. This would ensure that the design of the cartridges would be optimal in terms of keeping the functionality and reliability of the valve. The valve used in this concept is a 4-port counter-balance valve, explained more in depth in chapter 3.6. This valve requires four channels to function properly over time: venting, piloting, inlet, and outlet. By using separation inserts to create two chambers inside the shaft, it is possible to have a cartridge that would be free-floating between these inserts and all the required channels could be placed in a space- and cost-efficient way.

Property of Wärtsilä

Figure 28: Final proposal nomenclature

This concept also is very easily scaled, meaning if a customer requires certain specification in terms of flow depending on the environment the vessel is operating in, there is opportunities to change to sizing or number of valves to accommodate for different flow specifications. It is also very flexible in terms of the dimensions of the shaft, as the concept does not need major redesign to fit different shaft dimensions. Venting channel also double as the lubrication channel for the propeller hub, as this is one of the requirements of the oil distributions shaft. As the required flow of this RO500-unit is 480 l/min, it needs to have two valves for each actuation line; Sun hydraulics is a producer which Wärtsilä currently uses for other applications, so the proposed drawings includes four CWGALHN 240 l/min 3:1 pilot ratio, vented counterbalance valves to accommodate for the required flow. The cartridges that are included in the drawing are custom-made and is there for illustration purposes as Sun Hydraulics cartridges should be the aim for this unit.

To actuate the flow in the ahead line, there is another chamber which needs to have a length that corresponds to the stroke distance of the ahead line, or feedback rod. This is because a hole in the ahead line is the means of actuating the flow through the ahead line, and this hole needs to always be open so that there is always a complete channel from the HPP to the propeller hub during actuation. For the astern line, the actuation of flow is very simple as seen in Figure 28. The dimensions of the actuation lines are currently for a maximum flow of 480 l/min, as per the requirement from Wärtsilä.

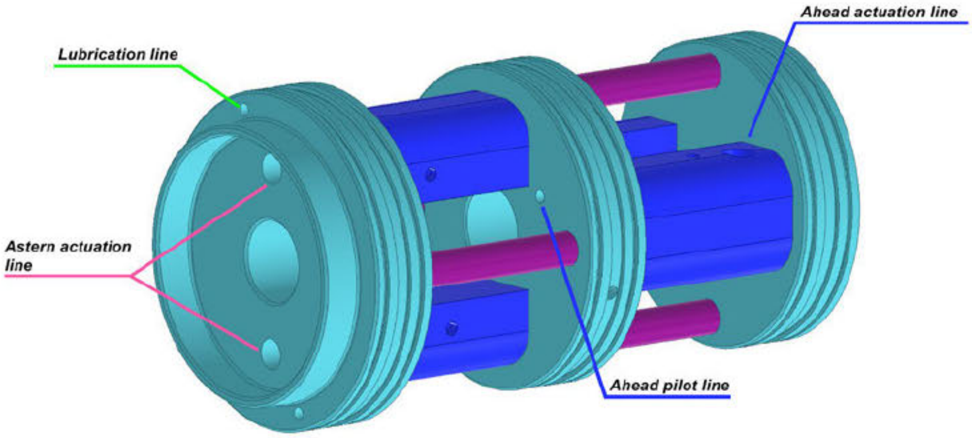

Figure 29: Valve insert nomenclature

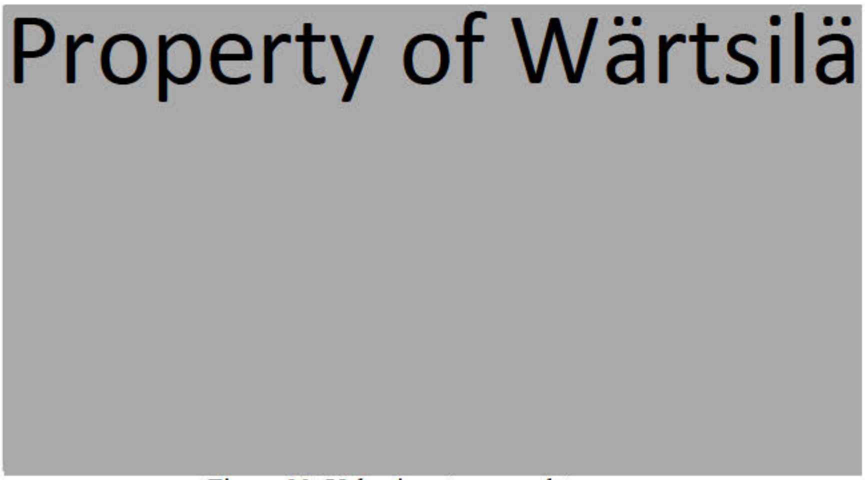

Figure 30: Valve insert nomenclature

One of the requirements of the RO-unit is that in case of blackout or seizing of the bearing, it should still be possible to manually pitch the propeller as a redundancy. To accommodate for this, there are two separate emergency pitch actuation points; the actuation points on the bearing will have a T-connection which enables the use of handheld hydraulic pump to actuate during blackout or problems with the HPP. This T-connection when closed allows hydraulic oil to pass through without any hinderance, but when it is opened blocks off the line leading to the HPP and opens the line to the CPP without any leakage. This way of manually pitching is easily operated, however, if the bearing seizes there needs to be an emergency solution that will work for a short amount of time, for example during take-me-home-mode to get the vessel to shore or out of dangerous situations. This emergency pitch setting will be done through a channel in the shaft directly into the ahead actuation chamber, bypassing the bearing completely.

The flange is somewhat removed from the scope of the report when the emergency pitch setting was moved to the servo-shaft, so the flange in this proposal is an internal standard used by Wärtsilä in similar applications. This is also true for the torque stay, in this proposal it is placed mainly for illustrations purposes.

The feedback mechanism stays mostly the same from draft 3 with minor updates. Geometry of the timings belts and pulleys has been changed to illustrate better. The fastening components between the timing belt and feedback ring is also added, along with the mechanism for transferring the axial movement of the feedback pipe. One conceptual change is that the passive pulley on the outside is now mounted to a plate ratter then a hole in the shaft. This is to prevent any stress concentrations close to the flange. Because of this the bearing is moved to be inside the pulley.

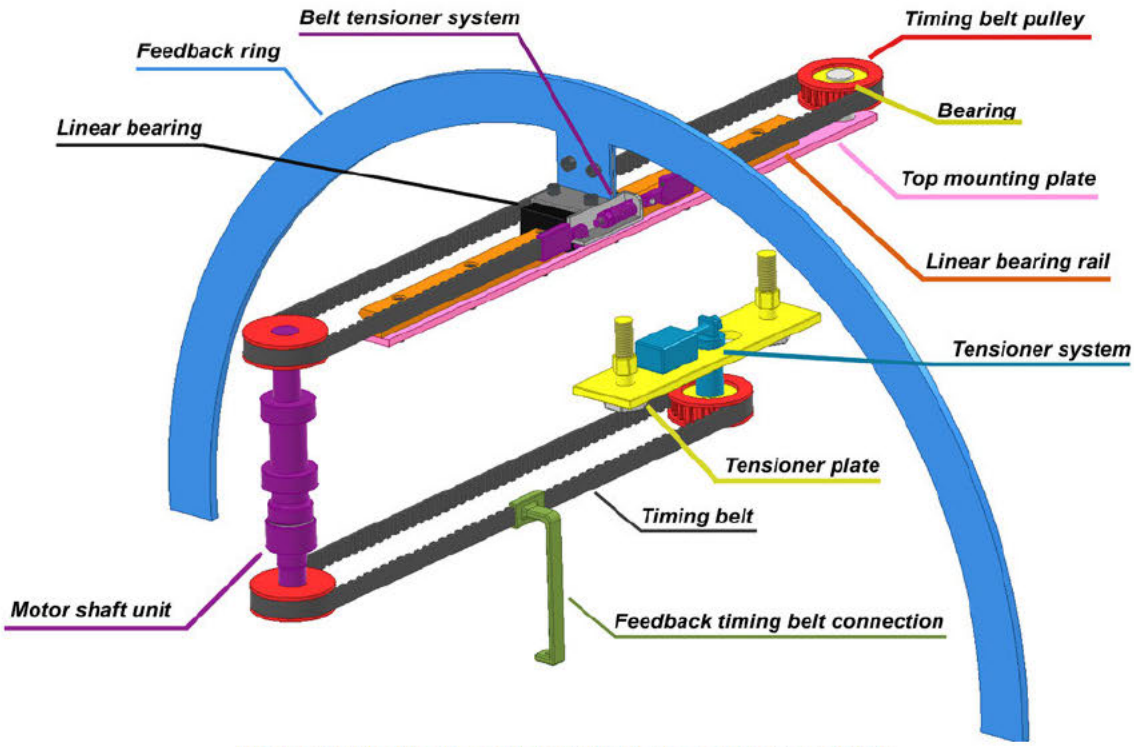

Figure 31: Feedback mechanism final proposal nomenclature

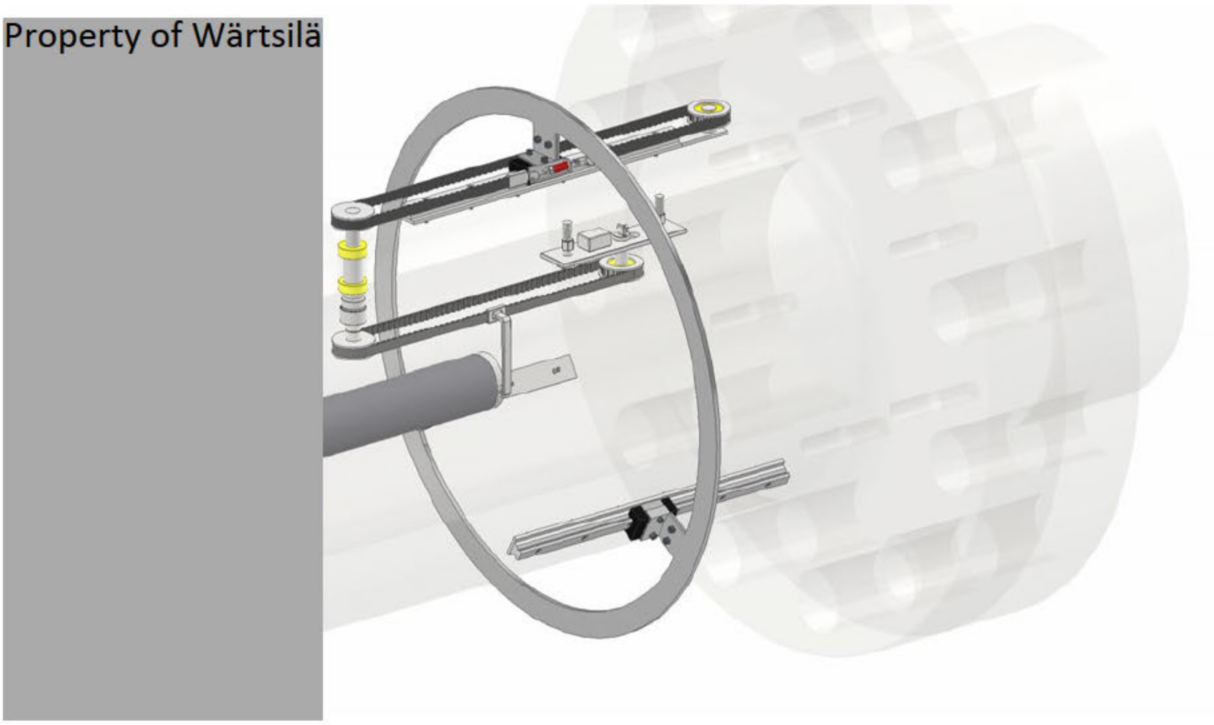

Figure 32: Feedback mechanism final proposal

In Figure 32 one can see that the feedback mechanism has two additional linear bearings and rails. This was done to make the feedback ring more stable, so that unwanted flex would not occur. This could however lead to there being a risk of jamming the system if not installed correctly.

The inspiration behind the idea with timing belts came from high precision machines such as 3D-printers and Computer Numerical Controlled routers (CNC routers). These timing belts drive the build plate and extrusion head in the case of 3D printers, or a router head in the case of CNC. To implement this idea with timing belts the belt had to drive something that could transfer the rotational motion of the motor unit to a linear movement that could be read by a sensor. To solve this the idea with linear bearings came. This would ensure a smooth movement and as a bonus the linear bearings that Ewellix deliver comes with the possibility to attach components on top. The bearing chosen is their size 15 LLTHC carriage and can be seen in appendix 8. It is this bearing that the linear bearing in the feedback system is modelled after, as well as the guiding rail for that specific bearing (size 15) as seen in appendix 8. The belt tensioner system as seen in Figure 31 was inspired by garage door openers which uses a similar system to ensure good belt tension.

Where this bearing would be placed there is a risk of contaminating the bearing. To lower the risk of that happening there is options such as Ewellix seal kit or bellows as shown in appendix 8. There would also have to be some sort of box surrounding the feedback mechanism to protect it against accidental and possibly damaging actions or occurrences. This is outside the thesis scope but that would also help keeping contaminants away from crucial parts. To measure the position of the feedback ring, a stationary sensor would continuously measure the distance to the feedback ring and transmit this signal to the control system of the vessel.

The required geometry of the shaft was studied using a finite element method, using the same approach as the reference study done in chapter 3.4.2. The results are within the reference point, and as expected the strongest stress concentration is in the feedback hole as seen in Figure 35. This indicates that it is possible to use the same material as the current shaft, which is beneficial in terms of cost and preference of producer. Comparatively to the current design, it should be a cost-efficient design to produce because of the simplicity.

Final proposal		
Criteria	*Description*	*Score (1-5)*
Complexity drivers	One diameter across the shaft. Biggest source of complexity is the feedback mechanism.	4
Feedback system	Requires correct instalment to achieve precision. Removes second hole from previous draft.	4
Oil distribution	Multiple valves in each direction gives the system redundancy in case of failure. Not highly space efficient.	4
Design robustness	Has two valves for each direction which gives redundancy, valves has safety release which gives safety if unwanted pressures arise. Feedback mechanism might be prone to damage.	4
Assembly	Correct instalment of feedback mechanism is crucial.	4
Sum	Total score of design (out of 25)	**20**

Table 14: Final proposal evaluation

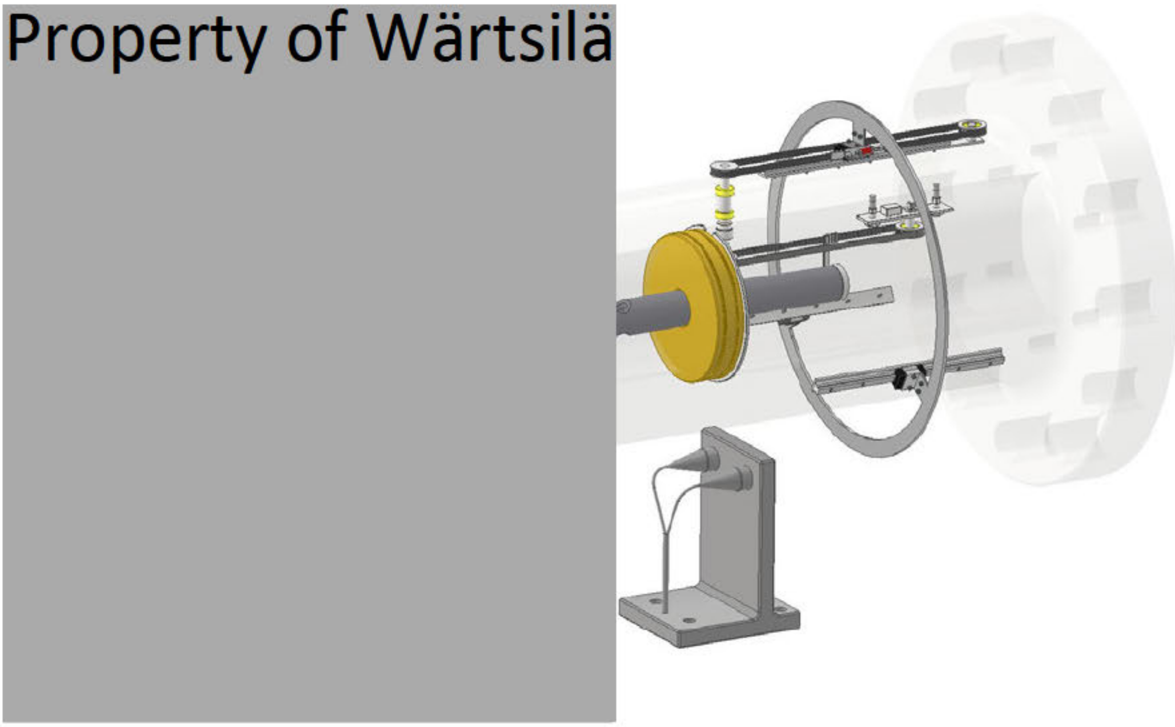

Figure 33: Final proposal transparent

Figure 34: Final proposal

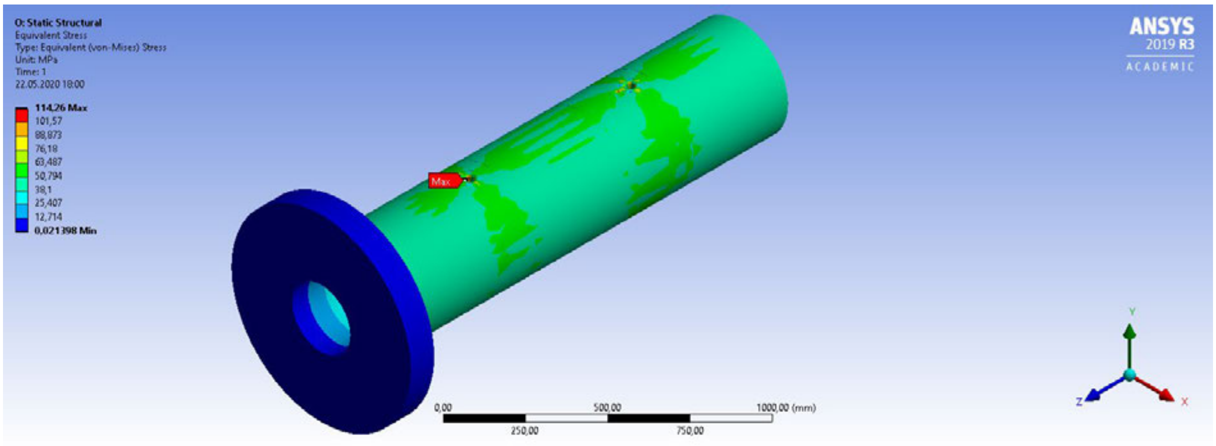

Figure 35: FEM analysis of final proposal

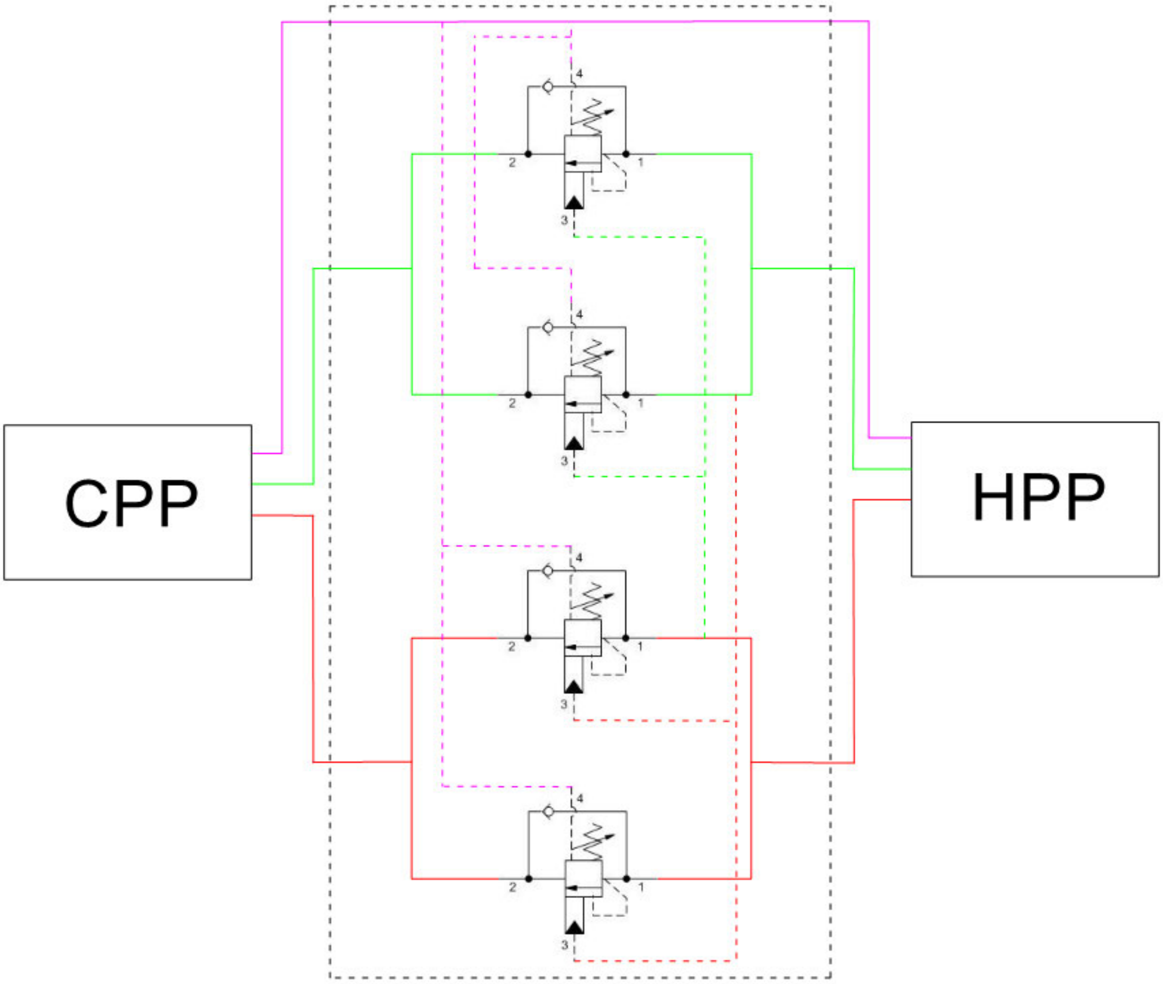

Figure 36: Hydraulic scheme of final proposal

5. Discussion

During the development process, Wärtsilä continuously shared their experience and knowledge on theoretical and practical aspects of this thesis. Information given by Wärtsilä sparked ideas that would be very interesting to pursue, however, because of time limitations some ideas had to be put aside. For example, ideas like the use of sensors to measure the pitch position and Wi-Fi to transmit the signals to an external receiver. The geometry of the shaft can be significantly simplified further reducing the overall cost of the system. One of the main issues with this solution is the delivery of the power needed to run the sensor and the Wi-Fi transmitter, there are technologies on the market that solve this issue, but as of today these technologies are expensive which would result in a similar overall cost as the current design. Another issue is that the customers are sceptical about the sole use and reliability of electrical components used in essential machinery.

Digital hydraulic is also a very interesting concept that should be further researched, as this could allow for the oil distribution to circumvent the need for a journal bearing to actuate the pitch positioning. Digital hydraulics allow for complete hydraulic systems to be placed in a space efficient manner, possibly inside the servo-shaft, and because of the precision of the hydraulic flow – feedback mechanisms could also be removed if functional requirements and class guidelines would permit for it.

Wärtsilä also shared some of their concept development, regarding this subject, as mentioned in chapter 4.3.4 this lead to the thesis taking a different route so the proposal could benefit Wärtsilä more than receiving a version very similar to their own concept.

6. Conclusion

As the thesis developed, some changes were made to the scope of the report. The scope of the study regarding journal bearings were simplified, because of the time frame of the thesis and complexity of the calculation. Originally the scope of the thesis stated that there should be developed a method for full journal bearing analysis done on their journal bearing inserts.

Research done throughout the thesis accumulated in a pragmatic solution to the problem statement, the proposed concept is a simple and cost-efficient variation of the OD-shaft. By separating the oil distribution function and the feedback mechanism, it was possible to create a shaft geometry that does not have high stress concentration factors and should result in a lower production costs for the shaft. The reduction of shaft diameter also reduces the dimensions of the bearing, which during the thesis was deemed as one of the largest cost-drivers. However, because of the reduced scope of the thesis, a finalised bearing proposal was not made.

Further research on this subject should substantiate the reasoning of this thesis further, by using stronger methods of analysis and more accurate circumstances and parameters of the subjects analysed. Also look further in to designing a cost-efficient journal bearing, or maybe even other types of bearings, as the load capacity required in the proposed concept has been significantly reduced.

7. References

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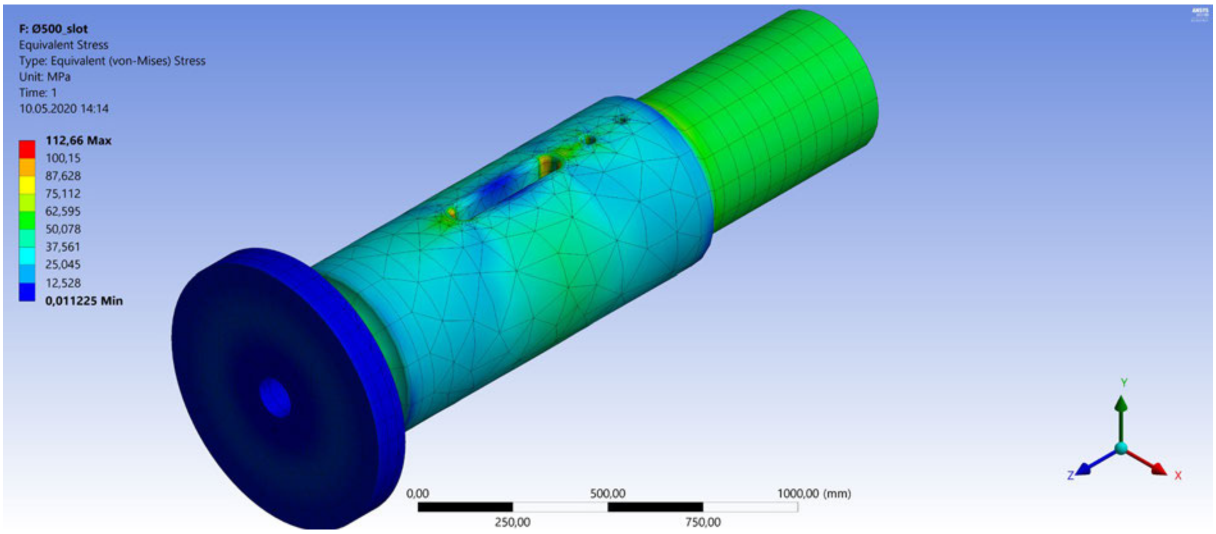
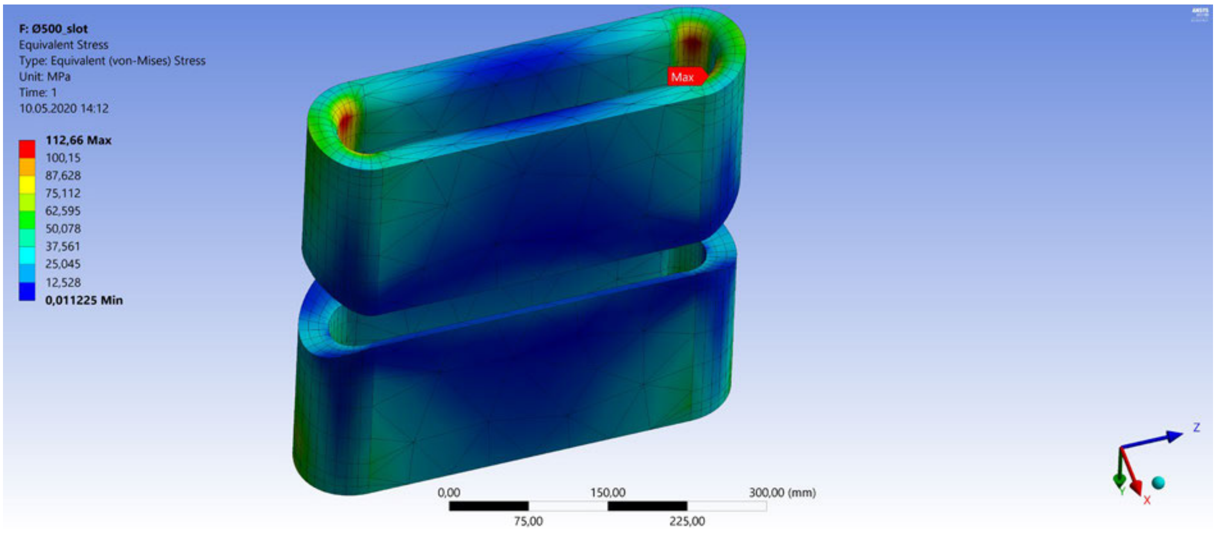
List of figures

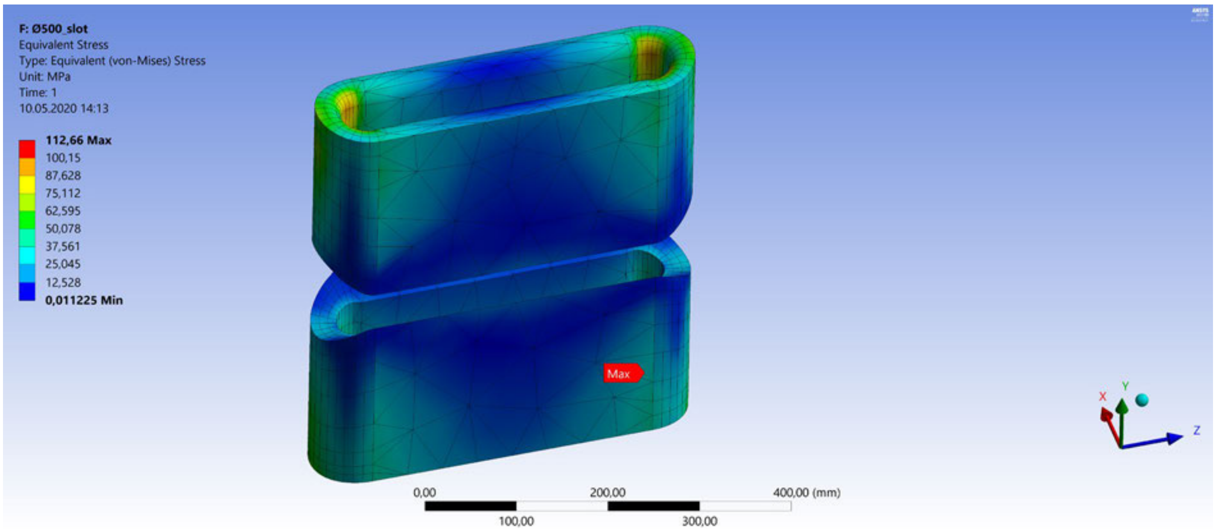
List of tables

1. Appendix (FEM original)

Description: Current OD-shaft

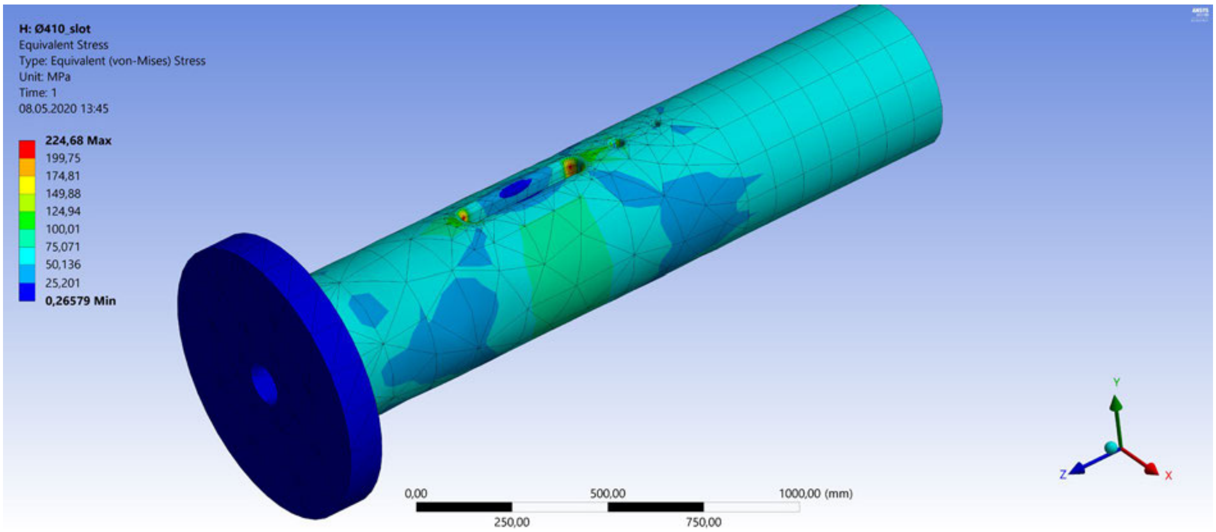






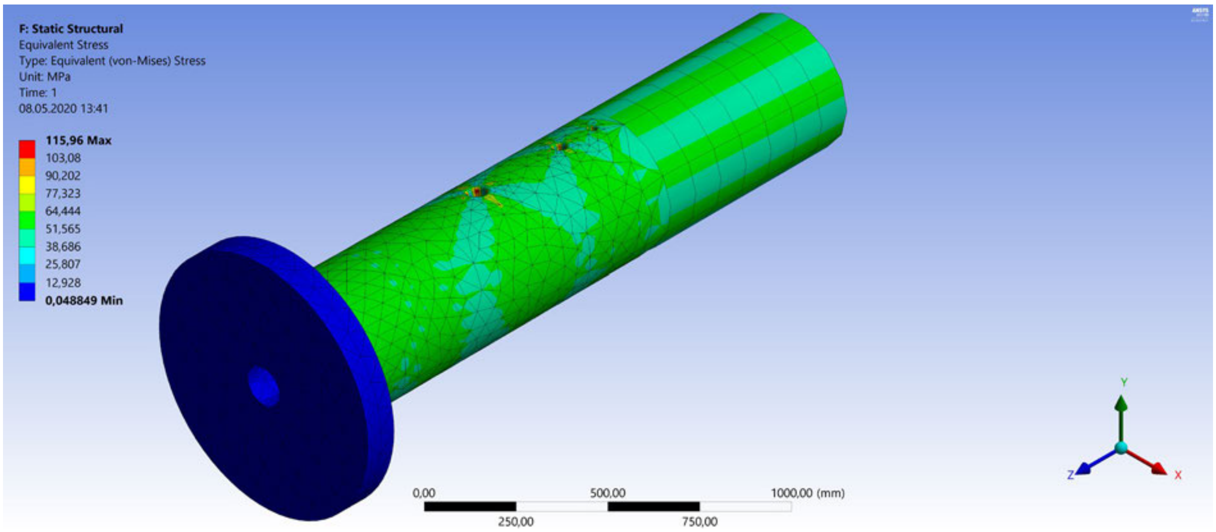
2. Appendix (FEM)

Description: Ø410 with longitudinal slot as means of feedback mechanism



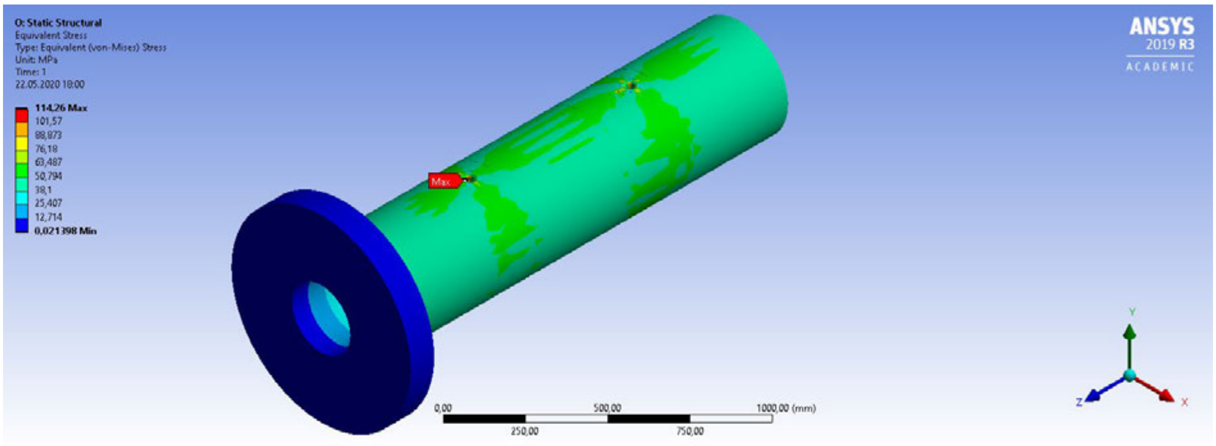
3. Appendix (FEM)

Description: Ø410 with radial hole as means of feedback mechanism

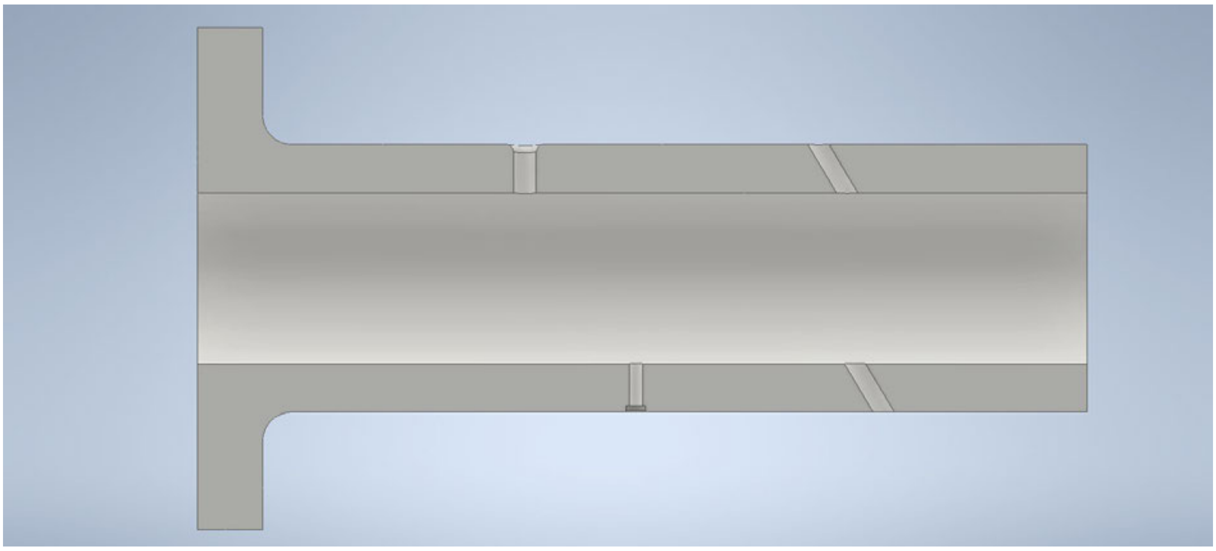


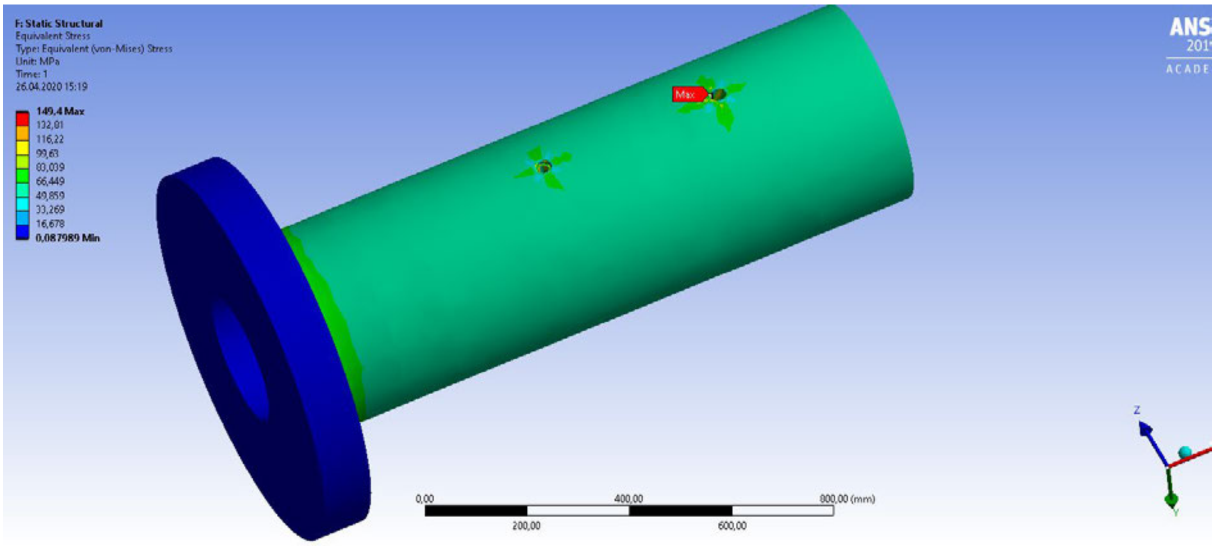
4. Appendix (FEM)

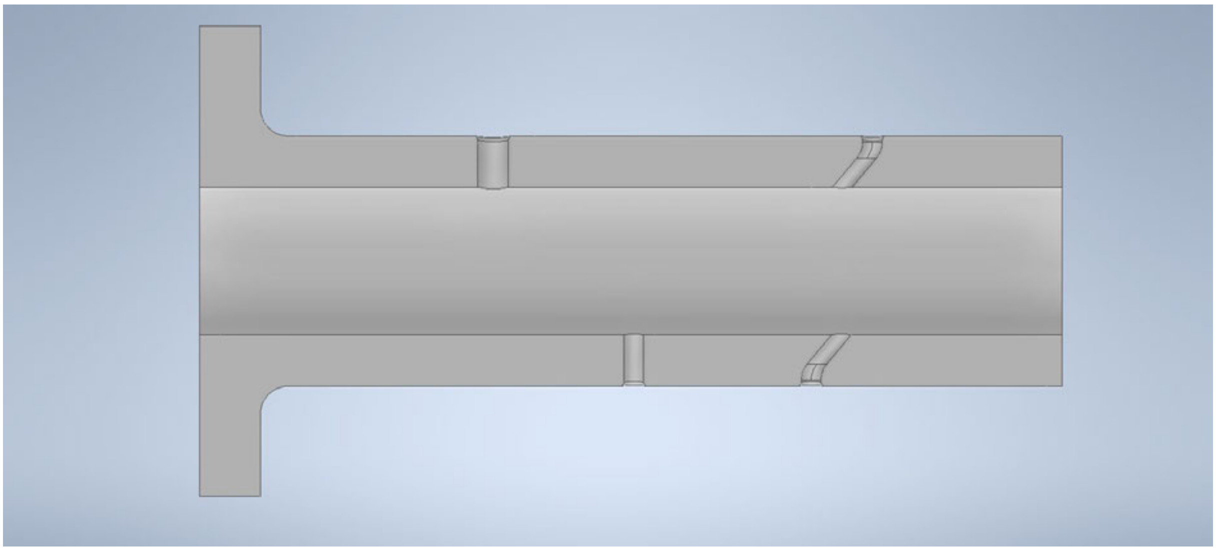
Description: Shaft from final proposal

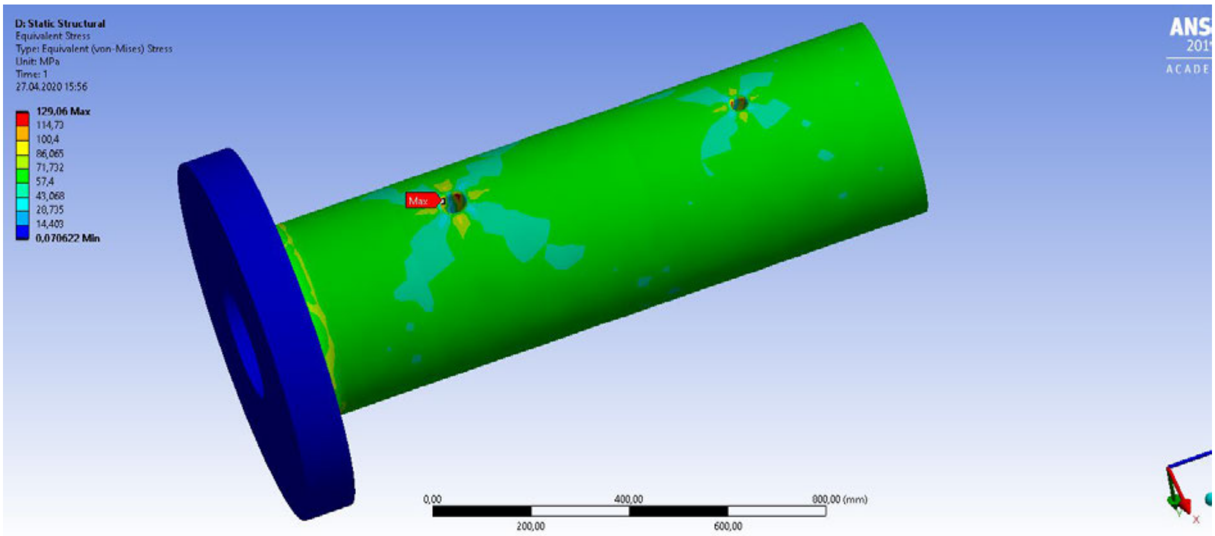


5. Appendix (FEM)



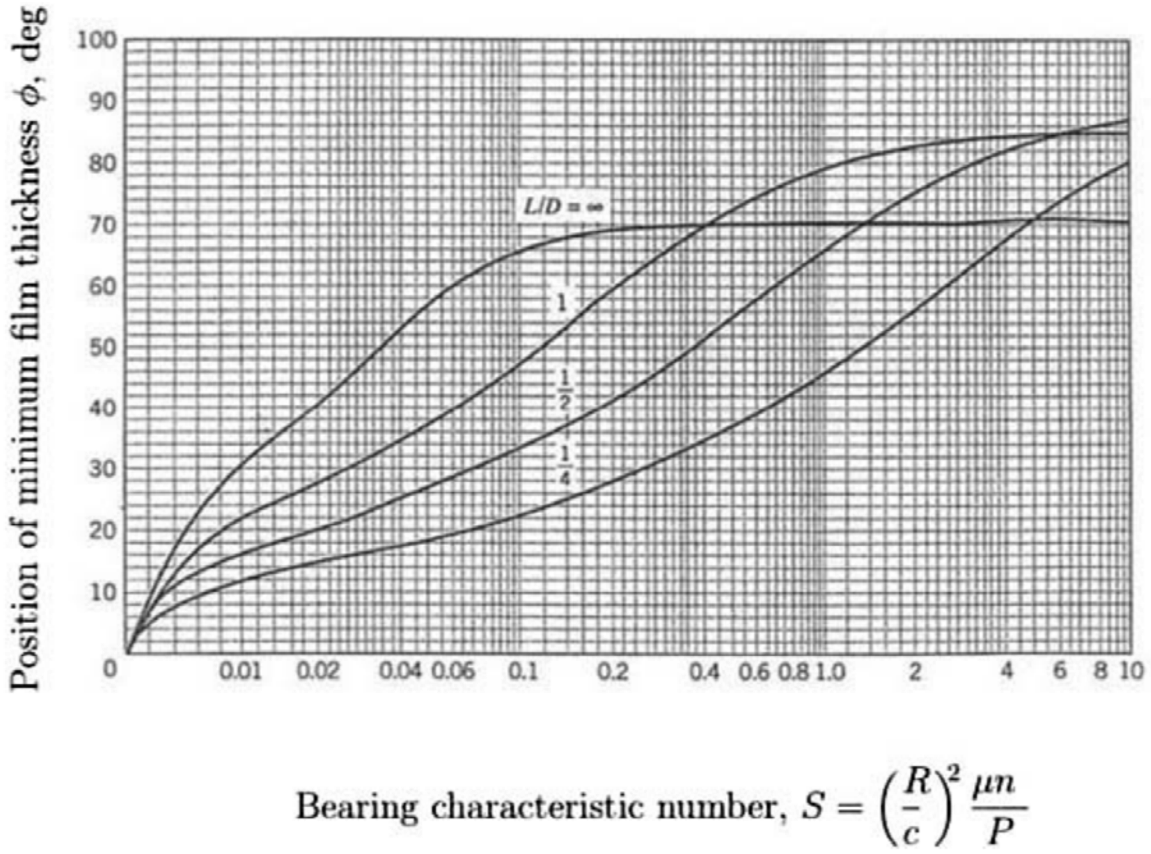


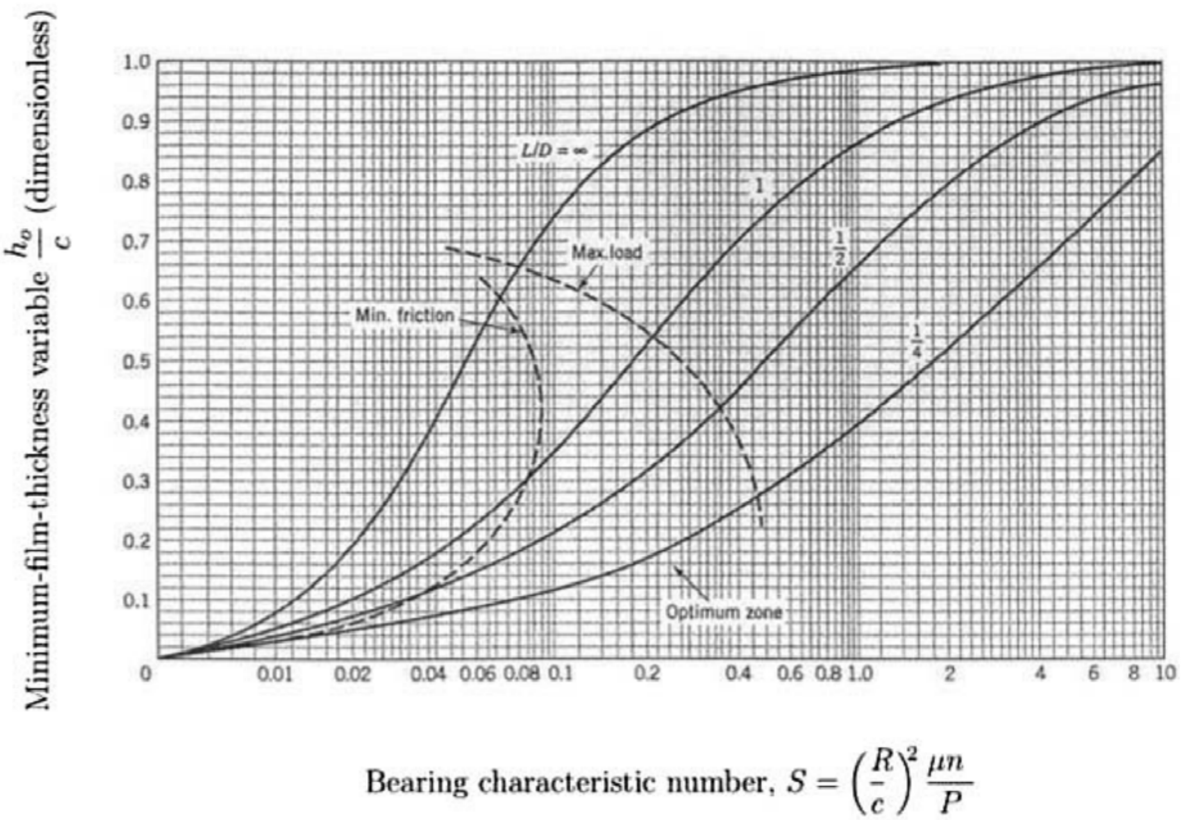


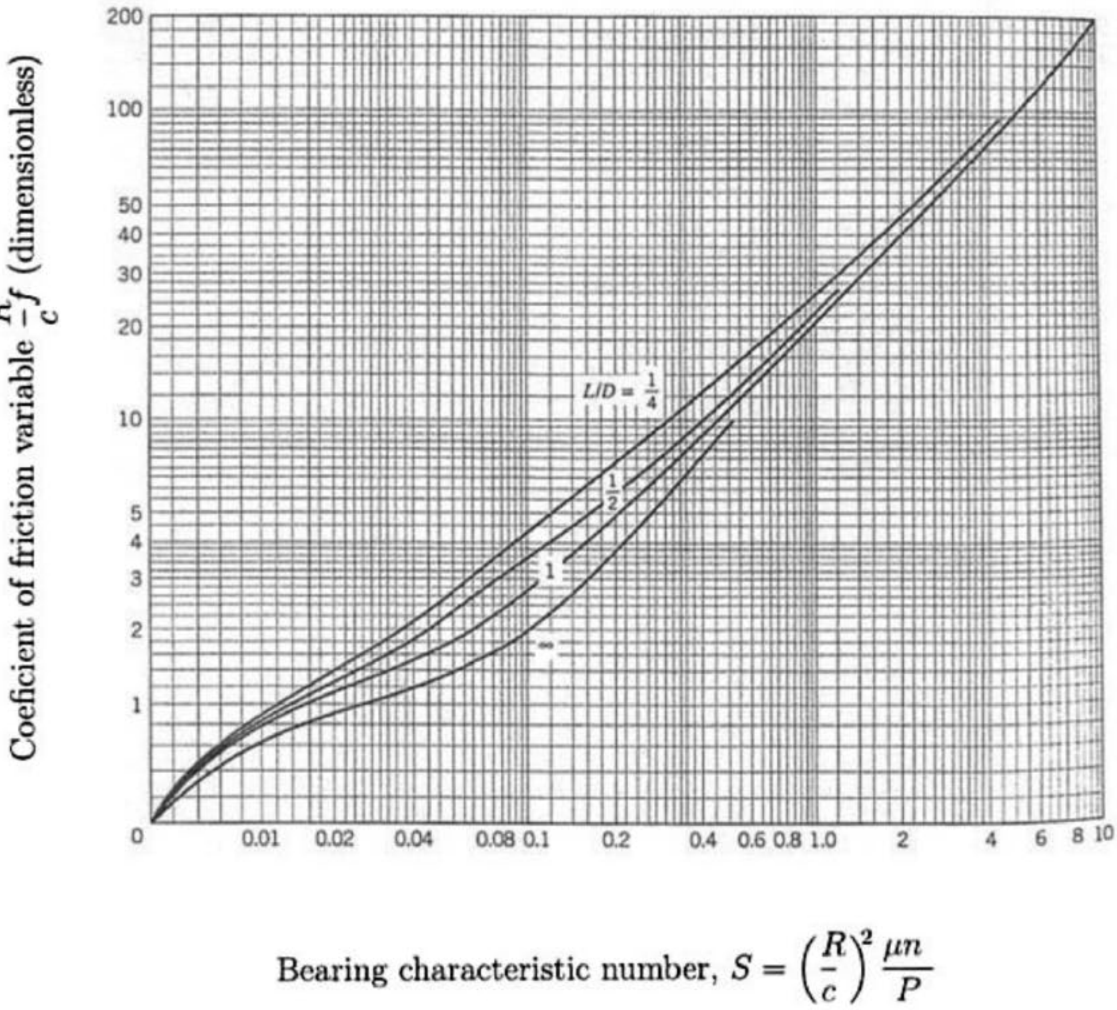

D: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
27.04.2020 15:56
129,06 Max
114,73
100,4
86,065
71,732
57,4
43,068
28,735
14,403
0,070622 Min
Max
0,00
400,00
800,00 (mm)
200,00
600,00

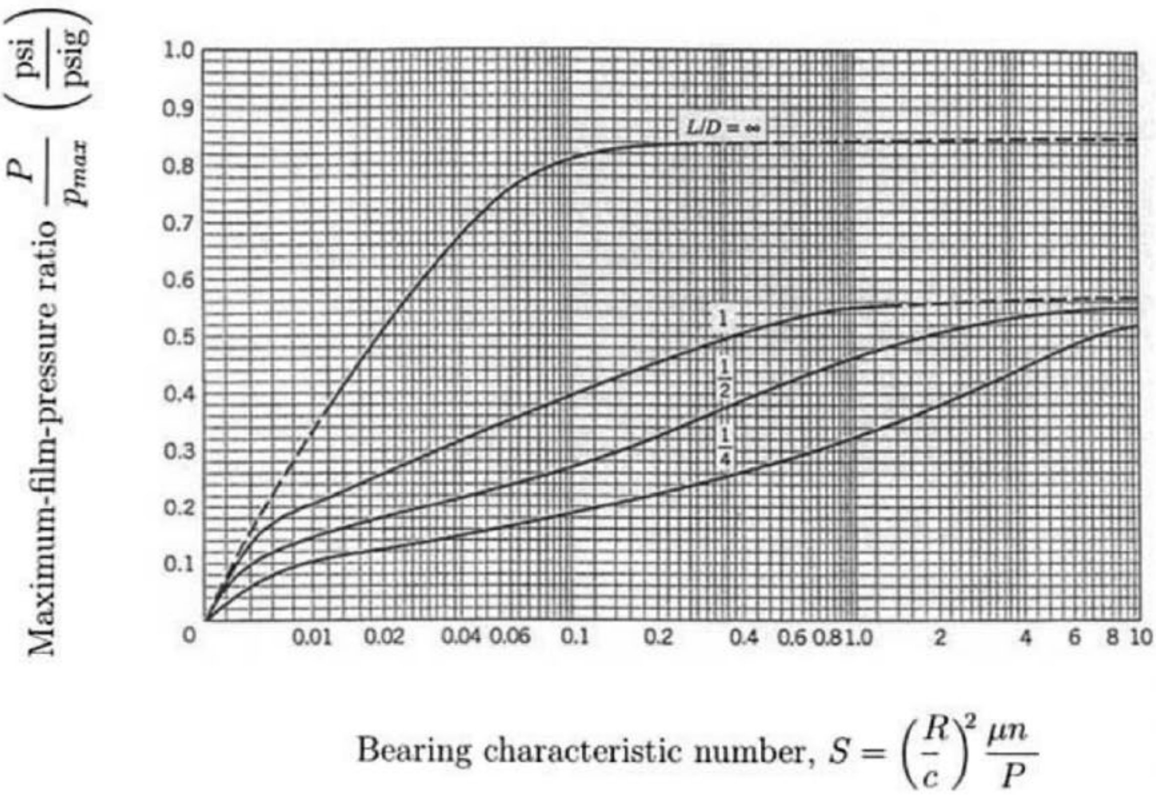
6. Appendix (Journal bearing design graphs)

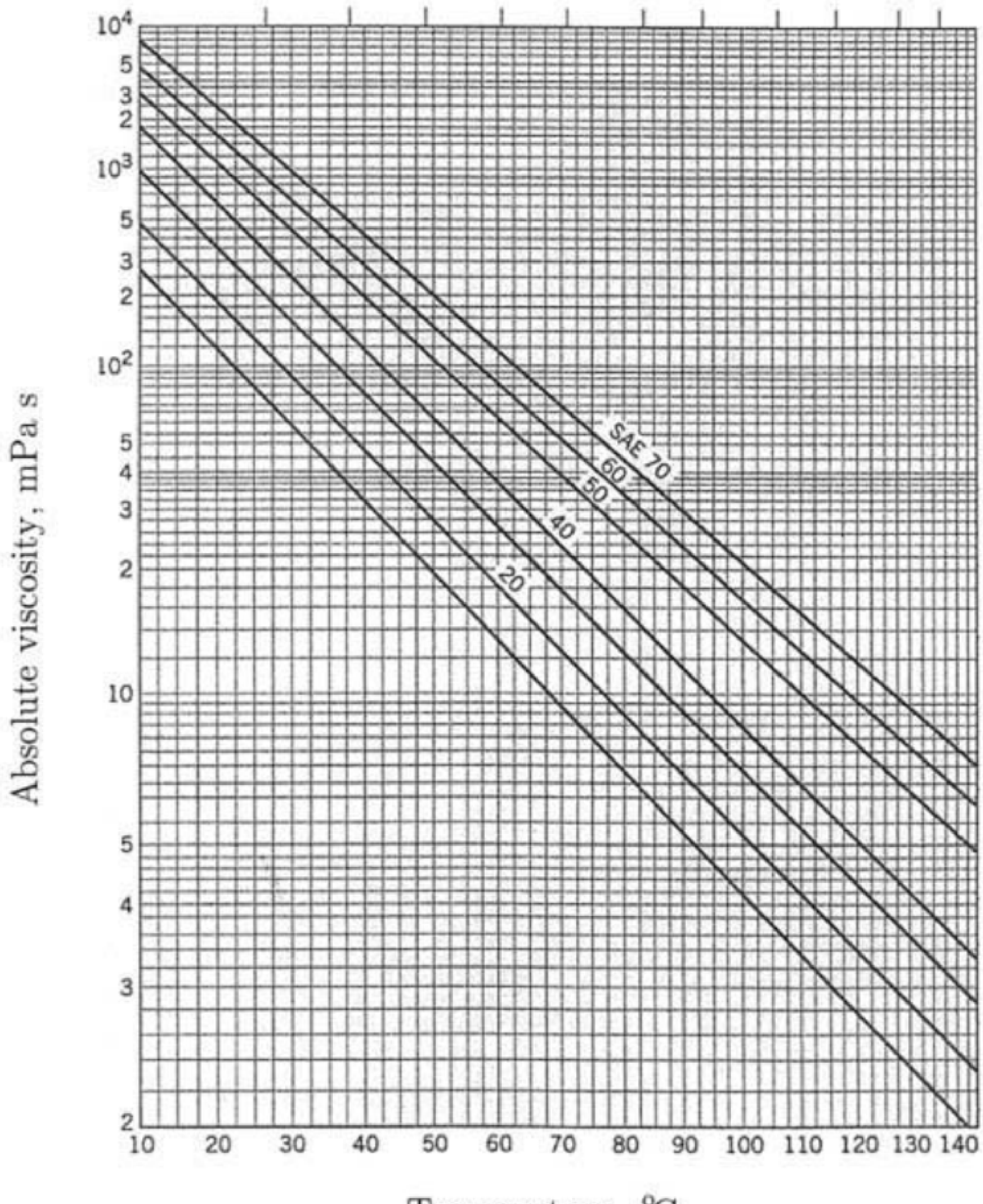
Graphs for bearing design [20].

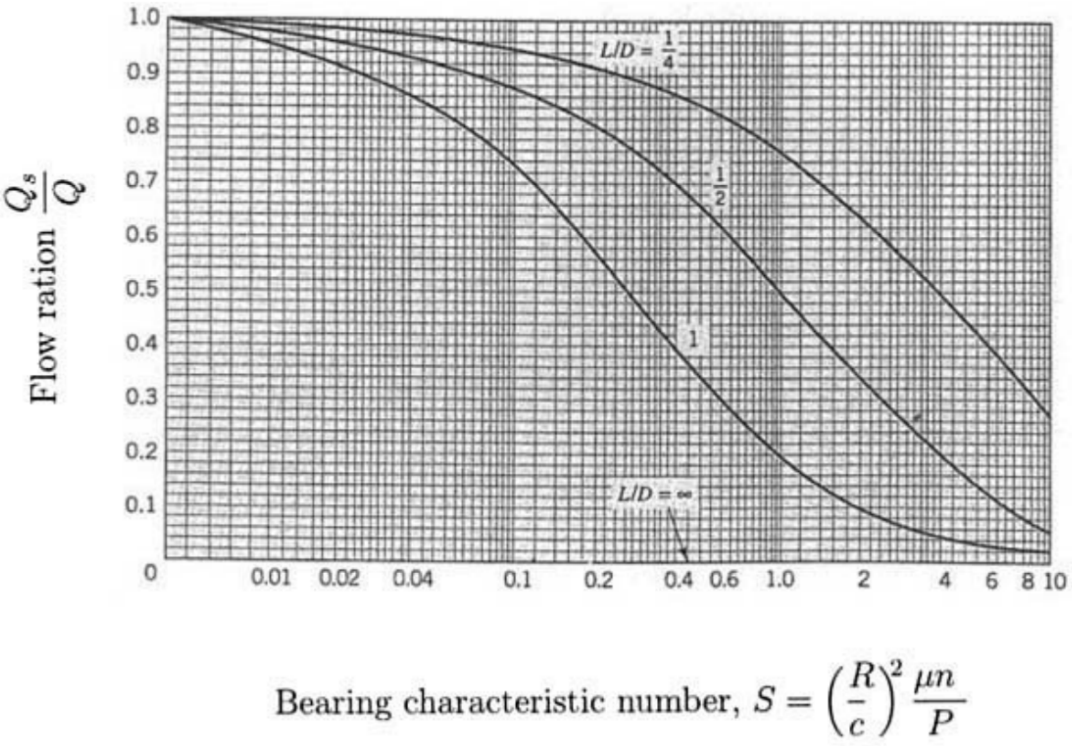


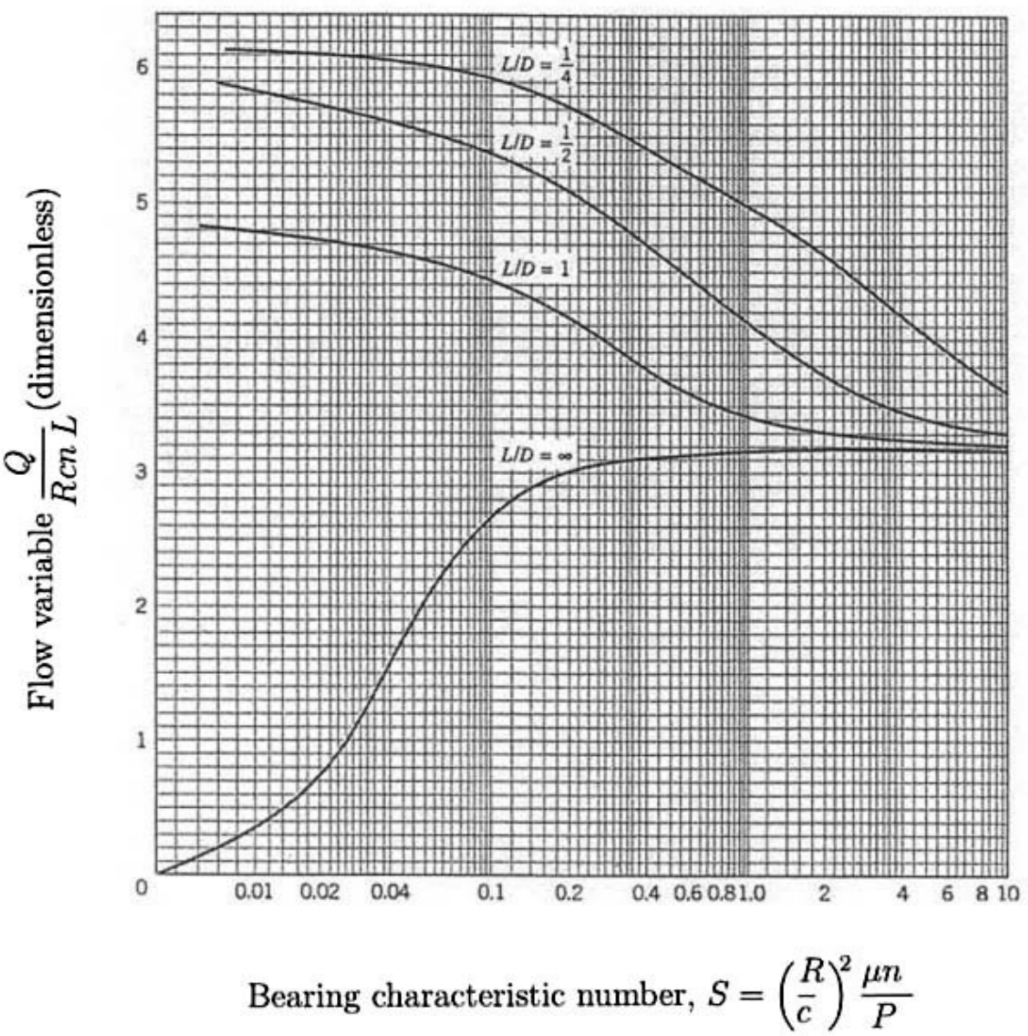



Coeficient of friction variable $\frac{R}{c}f$ (dimensionless)
200
100
50
40
30
20
10
5
4
3
2
1
0
0.01
0.02
0.04
0.08 0.1
0.2
0.4
0.6 0.8 1.0
2
4
6
8 10
$L/D = \frac{1}{4}$
$\frac{1}{2}$
1
∞
Bearing characteristic number, $S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P}$


Maximum-film-pressure ratio $\frac{P}{p_{max}}$ $\left(\frac{\text{psi}}{\text{psig}}\right)$
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.01
0.02
0.04 0.06
0.1
0.2
0.4
0.6 0.8 1.0
2
4
6
8 10
$L/D = \infty$
1
$\frac{1}{2}$
$\frac{1}{4}$
Bearing characteristic number, $S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P}$


Absolute viscosity, mPa s
Temperature, °C
SAE 70
60
50
40
20
10
20
30
40
50
60
70
80
90
100
110
120
130
140


1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
Flow ration $\frac{Q_s}{Q}$
$L/D = \frac{1}{4}$
$\frac{1}{2}$
1
$L/D = \infty$
0.01 0.02 0.04 0.1 0.2 0.4 0.6 1.0 2 4 6 8 10
Bearing characteristic number, $S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P}$


6
5
4
3
2
1
0
Flow variable $\frac{Q}{RcnL}$ (dimensionless)
$L/D = \frac{1}{4}$
$L/D = \frac{1}{2}$
$L/D = 1$
$L/D = \infty$
0.01 0.02 0.04 0.1 0.2 0.4 0.6 0.8 1.0 2 4 6 8 10
Bearing characteristic number, $S = \left(\frac{R}{c}\right)^2 \frac{\mu n}{P}$

7. Appendix (Table of journal bearing values)

T, ˚C	Dynamic viscosity, Pa s	Sommerfeld nr.	h0/C	h0	e	Q/RCnl	Q, m3/s	Qs/Q	Qs, m3/s	P/Pmax	Pmax	(R/C)* f	f
10	0.46	56.59	OG	OG	OG	OG	OG	OG	OG	OG	OG	OG	OG
20	0.18	22.15	OG	OG	OG	OG	OG	OG	OG	OG	OG	OG	OG
30	0.085	10.46	0.84	0.000084	0.16	3.6	0.0000279	0.26	0.000007254	0.52	244230.7692	OG	OG
40	0.045	5.54	0.72	0.000072	0.28	4	0.000031	0.42	0.00001302	0.48	264583.3333	OG	OG
50	0.028	3.44	0.62	0.000062	0.38	4.3	0.000033325	0.52	0.000017329	0.42	302380.9524	75	0.03
60	0.018	2.21	0.54	0.000054	0.46	4.6	0.00003565	0.62	0.000022103	0.38	334210.5263	50	0.02

OG = Outside Graphs

8. Appendix (Ewellix linear bearing)

This appendix is gathered from Ewellix catalogue of linear bearings. [21]

3.1.4 Carriage LLTHC ... SU

Slim-line carriage, short length, standard height.

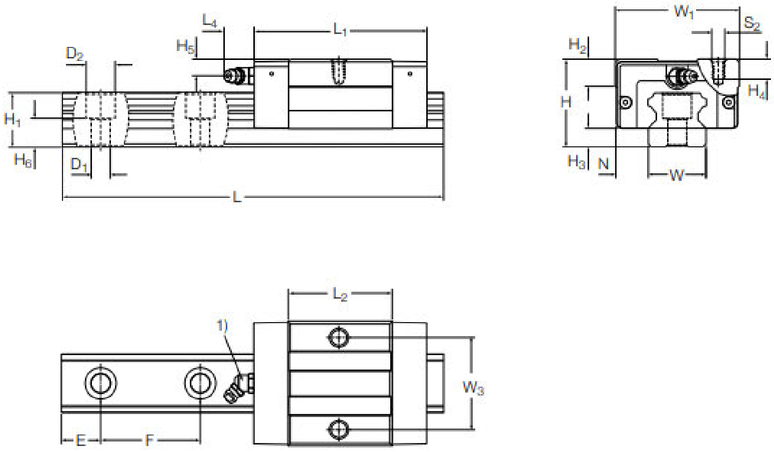
Carriages from size 15 to 30 are also available with low friction S0 shield. Dimensions are the same as standard version. For designation, refer to **Ordering key carriages** (↳ **page 103**).

Technical data

Size	Precision class	Designation [1] Preload class T0	T1
–		–	
15	P5	LLTHC 15 SU T0 P5	LLTHC 15 SU T1 P5
	P3	LLTHC 15 SU T0 P3	LLTHC 15 SU T1 P3
	P1		LLTHC 15 SU T1 P1
20	P5	LLTHC 20 SU T0 P5	LLTHC 20 SU T1 P5
	P3	LLTHC 20 SU T0 P3	LLTHC 20 SU T1 P3
	P1		LLTHC 20 SU T1 P1
25	P5	LLTHC 25 SU T0 P5	LLTHC 25 SU T1 P5
	P3	LLTHC 25 SU T0 P3	LLTHC 25 SU T1 P3
	P1		LLTHC 25 SU T1 P1
30	P5	LLTHC 30 SU T0 P5	LLTHC 30 SU T1 P5
	P3	LLTHC 30 SU T0 P3	LLTHC 30 SU T1 P3
	P1		LLTHC 30 SU T1 P1
35	P5	LLTHC 35 SU T0 P5	LLTHC 35 SU T1 P5
	P3	LLTHC 35 SU T0 P3	LLTHC 35 SU T1 P3
	P1		LLTHC 35 SU T1 P1

[1] • Preferred range.
• Only available as system.
For designation, please refer to designation system.

Dimensional drawing



Size	Assembly dimensions					Carriage dimensions						
	W_1	N	H	H_2	H_3	L_1	L_2	L_4	W_3	H_4	H_5	S_2
–	mm											
15	34	9,5	24	4,8	4,6	48,9	25,6	4,3	26	4	4,3	M4×0,7
20	44	12	30	9,3	5	55,4	32,1	15	32	6,5	5,7	M5×0,8
25	48	12,5	36	9,6	7	66,2	38,8	16,6	35	6,5	6,5	M6×1,0
30	60	16	42	12,6	9	78	45	14,6	40	8,5	8	M8×1,25
35	70	18	48	12,3	9,5	88,8	51,4	14,6	50	10	8	M8×1,25

Size	Rail dimensions									Weight		Load ratings [2]		Moments [2]			
										carriage	rail	dynamic	static	dynamic	static	dynamic	static
	W	H_1	H_6	F	D_1	D_2	E_{min} ±0,75	E_{max} ±0,75	L_{max} ±1,5			C	C_0	M_{xC}	M_{xC_0}	$M_{yC}=M_{zC}$	$M_{yC_0}=M_{zC_0}$
–	mm									kg	kg/m	N		Nm			
15	15	14	60	4,5	7,5	8,5	10	50	3 920	0,1	1,4	5 800	9 000	39	60	21	32
20	20	18	60	6	9,5	9,3	10	50	3 920	0,17	2,3	9 240	14 400	83	130	41	64
25	23	22	60	7	11	12,3	10	50	3 920	0,21	3,3	13 500	19 600	139	202	73	106
30	28	26	80	9	14	13,8	12	70	3 944	0,48	4,8	19 200	26 600	242	335	120	166
35	34	29	80	9	14	17	12	70	3 944	0,8	6,6	25 500	34 800	393	536	182	248

[1] For detailed information on grease nipples, please refer to **page 70**.
[2] Dynamic load capacities and moments are based on a travel life of 100 km. Please refer to **page 15** for further details.

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EWELLIX

Profile rail guides - LLT

3.2.1 LLTHR rails

Rails are supplied with protective plastic caps for mounting from above. For designation, refer to **Ordering key rails** (↳ **page 104**).

NOTE: If a rail length is required that exceeds the maximum length available, jointed rails can be ordered. These rails are manufactured so they match seamlessly to each other.

Technical data

Size	Precision class	Designation [1] One-piece rail	Multi-piece rail	Pitch F mm
–	–	–		
15	P5	LLTHR 15 - ... P5	LLTHR 15 - ... P5 A	60
	P3	LLTHR 15 - ... P3	LLTHR 15 - ... P3 A	
	P1	LLTHR 15 - ... P1	LLTHR 15 - ... P1 A	
20	P5	LLTHR 20 - ... P5	LLTHR 20 - ... P5 A	60
	P3	LLTHR 20 - ... P3	LLTHR 20 - ... P3 A	
	P1	LLTHR 20 - ... P1	LLTHR 20 - ... P1 A	
25	P5	LLTHR 25 - ... P5	LLTHR 25 - ... P5 A	60
	P3	LLTHR 25 - ... P3	LLTHR 25 - ... P3 A	
	P1	LLTHR 25 - ... P1	LLTHR 25 - ... P1 A	
30	P5	LLTHR 30 - ... P5	LLTHR 30 - ... P5 A	80
	P3	LLTHR 30 - ... P3	LLTHR 30 - ... P3 A	
	P1	LLTHR 30 - ... P1	LLTHR 30 - ... P1 A	
35	P5	LLTHR 35 - ... P5	LLTHR 35 - ... P5 A	80
	P3	LLTHR 35 - ... P3	LLTHR 35 - ... P3 A	
	P1	LLTHR 35 - ... P1	LLTHR 35 - ... P1 A	
45	P5	LLTHR 45 - ... P5	LLTHR 45 - ... P5 A	105
	P3	LLTHR 45 - ... P3	LLTHR 45 - ... P3 A	
	P1	LLTHR 45 - ... P1	LLTHR 45 - ... P1 A	

[1] • **Preferred range**,
• Only available as system.
By rail length in mm, e.g. LLTHR 15 -1000 P5

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Product range

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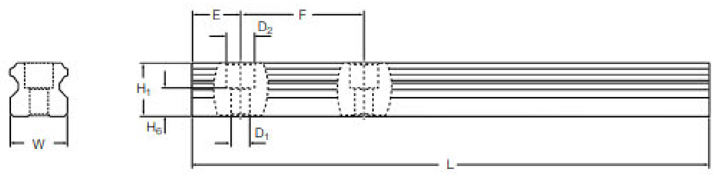
Dimensional drawing



Size	Dimensions									Weight
	W	H_1	H_6	D_1	D_2	E_{min} ±0,75	E_{max} ±0,75	F	L_{max} ±1,5	
–	mm									kg/m
15	15	14	8,5	4,5	7,5	10	50	60	3 920	1,4
20	20	18	9,3	6	9,5	10	50	60	3 920	2,3
25	23	22	12,3	7	11	10	50	60	3 920	3,3
30	28	26	13,8	9	14	12	70	80	3 944	4,8
35	34	29	17	9	14	12	70	80	3 944	6,6
45	45	38	20,8	14	20	16	90	105	3 917	11,3

3

The "E" dimension designates the distance from the rail end to centre of the first attachment hole. If no specific "E" dimension is provided by the customer with the order, the rails are produced according to the following formulae:

Calculation of number of attachment holes in rail guide

(1) $n_{real} = \frac{L}{F}$

(2) Round down of n_{real} to n

(3) n + 1 = z

F = Distance of attachment holes

L = Rail length

n_{real} = Real calculation value number of hole distances

z = Number of attachment holes in rail

Determination of E dimension based on z

(4) $E_{real} = \frac{L - F\,(z - 1)}{2}$

E_{real} = Real calculation value for E-dimension

E_{min} = Minimum E-dimension according to catalogue

Comparison with catalogue value of E_{min}

(4.1) If $E_{real} \geq E_{min}$
Usage of E_{real} from **formula 4**

(4.2) If $E_{real} < E_{min}$
Calculation of E_{real} according to **formula 5**

(5) $E_{real} = \frac{L - F\,(z - 2)}{2}$

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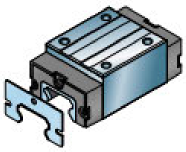
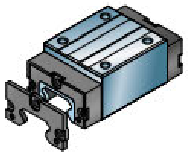
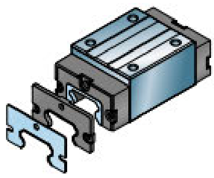
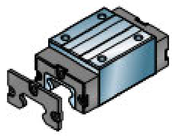
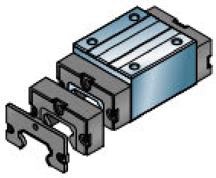
Profile rail guides - LLT

3.3 Accessories

Accessories Item name	Illustration [1]	Purpose
Scraper plate **LLTHZ ... S1**		Scraper plates are spring-steel, non-contact components. They protect the front seal from coarse contaminants or hot metal chips.
Additional front seal **LLTHZ ... S7**		Additional front seals are contact seals that can be attached to the carriage end faces. They are single-lip seals consisting of special heavy-duty material and offer additional protection against liquids and smaller contaminants. An additional front seal, in combination with carriages equipped with a low friction S0 shield, result in a sealed system with lower friction.
Seal kit **LLTHZ ... S3**		The seal kit consists of a metal scraper and an additional front seal. It is intended for applications involving exposure to coarse and fine dirt as well as liquids.
Low friction shield **LLTHZ ... S0**		The low friction shield has no contact with the rail. It replaces the standard front seal and results in a reduced friction force. Since the sealing function is not in place at the front ends of the carriage, the lubrication interval is reduced. The low friction shield is available for carriages from size 15 to 30.
Lube element **LLTHZ ... S6**		The lube element is designed to enhance the service life of a profile rail guide system. It provides an additional lubrication reservoir which consists of foam filled with oil in permanent contact with the raceways at the rail. By capillary attraction of the foam, the oil is constantly dosed in the correct amount. The ball elements roll through the applied oil film, whereby the optimal tribological conditions are maintained. The lube element is mounted on the end plate. The use of the originally delivered front seal keeps the dirt outside and the lubricant inside the carriage.

[1] Appearance can vary slightly depending on the size

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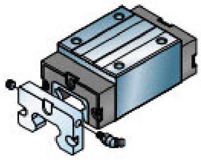
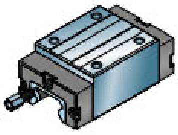
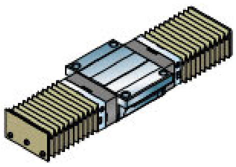
Accessories Item name	Illustration [1]	Purpose
Adapter plate **LLTHZ ... PL**		Adapter plates provide a side lubrication point, either for a grease nipple or for central lubrication systems. The interface of the adapter plate is the same on both sides. The adapter plate can be mounted on both end sides of the carriage. Usually only one adapter plate is used per carriage. Please note that this accessory is also part of the bellow sets.
Lubrication connector **LLTHZ ... VN UA**		The lubrication connector is used to provide an interface for central lubrication systems. The lubrication connector can be mounted on both end sides of the carriage. Usually only one lubrication connector is used per carriage. Please note that the lubrication connector cannot be used in combination with additional seals (scraper plate, additional front seal, seal kit and adapter plate).
Bellows **LLTHZ ... B**		Bellows protect the entire system against solid and liquid contaminants from above. They are suitable for highly contaminated environments like machining centres in the woodworking and metals industries.

[1] Appearance can vary slightly depending on the size

9. Appendix (Properties of SUN hydraulics valve)

From SUN-hydraulics website [16].

Note: Data may vary by configuration. See CONFIGURATION section.

Cavity	T-22A
Series	2
Capacity	120 L/min.
Pilot Ratio	3:1
Maximum Recommended Load Pressure at Maximum Setting	215 bar
Maximum Setting	280 bar
Factory Pressure Settings Established at	30 cc/min.
Maximum Valve Leakage at Reseat	0,3 cc/min.
Check Cracking Pressure	1,7 bar
Adjustment - No. of CCW Turns from Min. to Max. Setting	5
Operating Characteristic	Standard
Reseat	>85% of setting
Valve Hex Size	28,6 mm
Valve Installation Torque	61 - 68 Nm
Adjustment Screw Internal Hex Size	4 mm
Locknut Hex Size	15 mm
Locknut Torque	9 - 10 Nm
Model Weight	0.36 kg.
Seal kit - Cartridge	Buna: 990022007
Seal kit - Cartridge	Polyurethane: 990022002
Seal kit - Cartridge	Viton: 990022006

10. Appendix (Graphs from stress factor calculations)