
Design of a Rolling Bearings Test Rig

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Dissertation submitted to
Faculdade de Engenharia da Universidade do Porto
for the degree of:

Mestre em Engenharia Mecânica

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Porto, 2017

The work presented in this dissertation was performed at the
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To my family...

Acknowledgements

I have started my academic journey in Faculdade de Engenharia da Universidade do Porto (FEUP) in September de 2011 in Civil Engineering course. I went to this course with an open mind and knew that even though it was not my priority, I would learn many concepts that could be important for my life. It was a great pleasure and never a mishap, I made great friendships and these people were my "Porto Family". My first acknowledgements are for Alexandre Pinto and Pedro Vila Pouca and many other guys whom helped me to integrate in faculty.

It was with the bittersweet feeling that in September 2013 I grabbed the opportunity to integrate the course that I always wanted and left behind the one that although not my first choice was my course with my people. Therefore, I would like to say a special thank you to the former Director of the course Professor Francisco Vasques by the opportunity that he gave me.

In Mechanical Engineering everything happened differently, I did not make many friends, but the right ones. Therefore I would like to express my sincere acknowledgements to my friends Marco Bento, Carlos Pereira and Leonardo Moreira. Recently during this dissertation two more great friendship were made and for them, António Vieira e Joana Sousa, go my big thank you.

For the all the support, the teachings and the advices, I would like to give my greatly acknowledgement to my supervisors Doctor Ramiro C. Martins, Doctor Carlos M.C.G. Fernandes and Professor Jorge H.O. Seabra. It is a truly meaningful recognition because, without them, such successful work would not have been done.

I would also like to acknowledge to Professor Paulo T. Castro, Eng. Sérgio Tavares, Eng. José Almacinha, Eng. Fernando G. Almeida and Eng. Paulo Coelho, whom greatly contributed to the development of this work.

I would like to appreciate the entire CETRIB group for the availability, hospitality and help. I really, really enjoyed having done the dissertation in this company, it was a pleasure.

The last but most important thanks go to my family, because of them I always had all the support that I needed at the right moment. I really love them. To my mother, I would like to say that those motivational advices worked at key moment. To my father, I would like to give my best recognition for the work he does every day for our family. To my brother, I would like to thank you very much for your importance, friendship and to be always present in the good and bad moments. Finally, I would like to thank the most recent member of my family, my girlfriend, my soulmate, for all the support and love she has given me in the last 8 years.

Keywords:

Test Rig
Radial Rolling Bearings
Lubrication
Power Loss in Rolling Bearings
Heat Dissipation

Palavras Chave:

Banco de Ensaios
Rolamentos Radiais
Lubrificação
Perda de Potência nos Rolamentos
Dissipação de Calor

Abstract

In order to bridge a gap in the market where a standard test machine for evaluating power losses in radial bearings is not present, the present work aims at the design of a radial bearing test bench capable of evaluating the torque of losses. The losses in a bearing are due to different mechanisms, such as, friction, slipping and churning. These are dependent on the service conditions, such as applied load, lubricant type, immersion height and the lubricant feeding in the active contact zone between the raceways and the rolling bodies. Therefore, it is essential that all bearings tested simultaneously are subject to the same test conditions.

To this end, four rolling bearings are assembled on a shaft, with the two external rolling bearings mounted in the case walls while the load is applied on the two central ones. This assemble configures a load system based on four-point bending applied on the horizontal plane maintains the same load conditions for each bearing, the same immersion height and fundamentally the same lubricant feeding in the active contact zone between the raceways and the rolling bodies. By varying the conditions between each test, the bearing behavior pattern at different operating conditions (power loss) will be measured and thus improve the power loss prediction models.

The test case has an approximate size to one of the FZG gear test rig, has a load application system by means of a hydraulic cylinder which drives a shaft thereby transmitting load to the bearings.

A temperature control system is also designed for either heating or cooling the oil bath thereby enabling testing at temperatures different than the oil stabilization temperature. The whole course from the basic idea to the mechanical drawing of the parts is presented in detail throughout this report.

The final solution proposed for the test rig meets all the objectives and constraints applied to the project and is therefore a viable solution for implementation.

Resumo

O presente trabalho tem como objectivo o dimensionamento de um banco de ensaios de rolamentos radiais capaz de avaliar o binário de perdas, e assim colmatar uma lacuna presente no mercado, que não oferece uma máquina de teste normalizada para avaliação de perdas de potencia em rolamentos radiais. As perdas num rolamento devem-se a diferentes mecanismos, tais como, atrito, escorregamento e chapinagem. Estes são dependentes das condições de serviço, como sejam, a carga aplicada, o tipo de lubrificante, altura de imersão e as condições de alimentação de lubrificante na zona ativa do contacto entre as pistas e os corpos rolantes. Portanto é fundamental que todos os rolamentos ensaiados em simultâneo estejam sujeitos às mesmas condições de teste.

Para tal, quatro rolamentos são montados num veio, onde os dois exteriores são fixados na parede do carter enquanto que a carga é aplicada nos dois rolamentos centrais. Esta montagem dá origem a um sistema de carga baseado em uma flexão a quatro pontos aplicada sobre o plano horizontal mantém as mesmas condições de carga por cada rolamento, a mesma altura de imersão e fundamentalmente a mesma condição de alimentação de lubrificante na zona ativa do contacto entre as pistas do rolamento e os corpos rolantes. Fazendo variar as condições entre cada teste, o padrão de comportamento do rolamento a diferentes condições de operação (perda de potência) será medido, com esses valores os modelos de previsão de perda de potência serão melhorados.

O cárter para ensaio de rolamentos tem uma dimensão aproximada a um cárter do equipamento de teste de engrenagens FZG, possui um sistema de aplicação de carga por meio de um cilindro hidráulico que traciona um veio transmitindo assim carga aos rolamentos.

Um sistema de controlo de temperatura é também dimensionado, quer para o aquecimento, quer para o arrefecimento do banho de óleo permitindo assim efetuar testes a temperaturas diferentes da temperatura de estabilização do óleo. Todo o percurso desde a ideia de base até aos desenhos de fabrico das peças é apresentado com detalhe ao longo deste relatório.

A solução final proposta para o equipamento de teste cumpre com todos os objectivos e restrições aplicadas ao projecto, sendo portanto uma solução viável para implementação.

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List of Symbols

Symbol	Description	Unit
a	Diameter of the clevis pin	mm
A	Material elongation	%
A_{ca}	Surface area of the carter	m ²
$A_{ca,hb}$	Surface area of horizontal plate of the carter	m ²
$A_{ca,v}$	Surface area of vertical wall of the carter	m ²
b	Width of yoke ear	m
B	Rolling bearing width	mm
c	Height of rod end	m
C	Rolling bearing dynamic load capacity	kN
C_0	Rolling bearing static load capacity	kN
d	Internal diameter of bearing	mm
d_e	External diameter of rod thread	m
$d_{e,hub}$	External diameter of hub	m
d_i	Internal diameter of rod thread	m
$d_{i,hub}$	Internal diameter of hub	m
d_m	Mean diameter of rod thread	m
d_r	Diameter of rod end	m
d_{shaft}	Diameter of the main shaft	m
D	External diameter of bearing	mm
e	Semi-width of the rod end	m
F	Radial Force	N
Gr	Grashof number	-
h_{bs}	Height of rolling bearing sleeve cut area	mm
$h_{conv,hb,ext}$	Heat transfer coef. of ext. convection of horizontal plate	W· m ⁻² K ⁻¹
$h_{conv,hb,int}$	Heat transfer coef. of int. convection of horizontal plate	W· m ⁻² K ⁻¹
$h_{conv,v,ext}$	Heat transfer coef. of ext. convection of vertical wall	W· m ⁻² K ⁻¹
$h_{conv,v,int}$	Heat transfer coef. of int. convection of vertical wall	W· m ⁻² K ⁻¹
h_{conv}	Heat transfer coef. of radiation of carter (<i>Höhn et al.</i>)	W· m ⁻² K ⁻¹
h_{rad}	Heat transfer coef. of radiation of carter (<i>Höhn et al.</i>)	W· m ⁻² K ⁻¹

LIST OF SYMBOLS

Symbol	Description	Unit
j_f	Endurance limit factor FKM Standard	-
j_m	Static safety against fracture FKM Standard	-
j_{mt}	Creep strength safety against fracture FKM Standard	-
j_p	Static safety against yield point FKM Standard	-
j_{pt}	Creep strength safety against yielding FKM Standard	-
k_a	Surface modification factor	-
k_b	Size modification factor	-
k_c	Load modification factor	-
k_{carter}	Thermal conductivity of the carter material	$W \cdot m^{-1}K^{-1}$
k_d	Temperature modification factor	-
k_e	Reliability modification factor	-
k_f	Miscellaneous-effects modification factor	-
K	Stress concentration factor (Equivalent stresses)	-
K_f	Stress concentration factor (Normal stresses)	-
l_{bs}	length of rolling bearing sleeve cut area	m
l_f	Load factor	-
l_v	Height of vertical wall	m
l_h	Length of horizontal plate	m
w_h	Width of horizontal plate	m
L_0	Length between the center of outer and inner bearings	m
L_{pipe}	Length of pipe of water exchanger	m
L_{shaft}	Length of main shaft	m
m	Width of the rod end	m
M_a	Alternating bending moment	$N \cdot m$
M_b	Bending moment	$N \cdot m$
M_{drag}	Frictional moment of drag losses	$N \cdot mm$
M_m	Midrange bending moment	$N \cdot m$
M_{rr}	Rolling frictional moment	$N \cdot mm$
M_{seal}	Frictional moment of the seals	$N \cdot mm$
M_{sl}	Sliding frictional moment	$N \cdot mm$
M_t	Total frictional moment	$N \cdot mm$
n	Number of thread helices of the rod	-
n_s	Number of bolts	-
N_p	Axial force in bolt	N
N_{rd}	Axial resistance force	N

Symbol	Description	Unit
$\overline{Nu}_D$	Nusselt number for diameter characteristic length	-
P_{VZP}	Power loss due to the gears	W
P_{VL}	Power loss due to the rolling bearings	W
P_{VD}	Power loss due to the seals	W
Q_a	Ratio between internal and external hub diameter	-
$\dot{Q}_{carter}$	Heat loss due to carter horizontal and vertical walls	W
$\dot{Q}_{cond,hb}$	Conduction heat loss horizontal plate	W
$\dot{Q}_{cond,v}$	Conduction heat loss vertical walls	W
$\dot{Q}_{conv,hb,ext}$	External convection heat loss horizontal plate	W
$\dot{Q}_{conv,hb,int}$	Internal convection heat loss horizontal plate	W
$\dot{Q}_{conv,v,ext}$	External convection heat loss vertical walls	W
$\dot{Q}_{conv,v,int}$	Internal convection heat loss vertical walls	W
$\dot{Q}_{hb}$	Total heat loss by horizontal plate	W
$\dot{Q}_{in,b}$	Heat gained due to rolling bearings	W
$\dot{Q}_{loss,e}$	Heat loss due to water exchanger	W
$\dot{Q}_{rad,hb}$	Radiation heat loss horizontal plate	W
$\dot{Q}_{rad,v}$	Radiation heat loss vertical walls	W
$\dot{Q}_{total,loss}$	Total heat loss by the test rig	W
$\dot{Q}_v$	Total heat loss by vertical walls	W
Re	Reynolds number	-
R_m	Material rupture stress	MPa
R_{pt}	Material yield stress for traction load	MPa
R_{pc}	Material yield stress for compression load	MPa
R_{ps}	Material yield stress for shear load	MPa
$R_{p0.2\%}$	Material yield stress at 0.2% elongation	MPa
S_e	Material modified endurance limit	MPa
S'_e	Material endurance limit	MPa
S_f	Part safety factor against yield	MPa
$S_{f,static}$	Main shaft static safety factor against yield	MPa
$S_{f,ASME}$	Main shaft fatigue safety factor	MPa
$S_{f,fyield}$	Main shaft dynamic safety factor against yield	MPa
T_{room}	Room temperature	K
T_{avg,H_2O}	Average water temperature	K
$T_{H_2O,in}$	Water inlet temperature	K
$T_{H_2O,out}$	Water outlet temperature	K

LIST OF SYMBOLS

Symbol	Description	Unit
$T_{hb,ext}$	Temperature of exterior surface of horizontal plate	K
$T_{hb,int}$	Temperature of interior surface of horizontal plate	K
T_{mean}	Average between Oil and Room temperature	K
T_{oil}	Oil temperature	K
$T_{v,ext}$	Temperature of exterior surface of vertical walls	K
$T_{v,int}$	Temperature of interior surface of vertical walls	K
$T_{wall,pipe}$	Temperature of water exchanger pipe wall	K
v_{air}	Air velocity	m·s ⁻¹
V	shear force	N
V_a	Alternating shear force	N
V_m	Midrange shear force	N
w_h	Width of horizontal plate	m
y	Main shaft deflection	m
ε	Emissivity	-
θ_{room}	Room temperature	°C
θ_{avg,H_2O}	Average water temperature	°C
$\theta_{H_2O,in}$	Water inlet temperature	°C
$\theta_{H_2O,out}$	Water outlet temperature	°C
θ_{oil}	Oil temperature	°C
$\theta_{wall,pipe}$	Temperature of water exchanger pipe wall	°C
σ_a	Alternating normal stress	MPa
σ_b	Normal bending stress	MPa
σ_{bs}	Normal stress in bearing sleeve	MPa
σ_c	Normal compression stress	MPa
σ_i	Normal stress in i direction ($i = x, y$ or z)	MPa
σ_m	Midrange normal stress	MPa
σ_{SB}	Stefan Boltzmann constant for radiation	W·m ⁻² K ⁻⁴
σ^{eq}	Equivalent stress (by von Mises and Tresca criteria)	MPa
$\sigma_{a,vM}^{eq}$	Equivalent alternating stress by von Mises criterion	MPa
τ_a	Alternating shear stress	MPa
τ_j	Shear stress in j direction ($j = xy, yz$ or zx)	MPa
τ_m	Midrange normal stress	MPa
ω_c	Critical speed of the main shaft	rpm

Introduction

1.1 Problem statement

In the 21st century, power sources and the way we consume them are a high priority. Recently the titlepage of the newspaper was “If all countries lived like Portugal lives, the world would start using the environmental credit card on World Environment Day”. Based on Global Footprint Network [1], 2.3 planets would be necessary to respond to Portuguese consumption. The biggest problem is the intrinsic relationship between people and fossil resources to produce energy. This paradigm has to change otherwise these problems will not be solved in time to avoid energy sources shortage.

Given the significance of energy in the present-day society, it is crucial to ensure maximum efficiency in all systems. Motivated by the rolling bearings power loss role in the global efficiency of several machines, it is of overriding importance to set an accurate method to estimate the rolling bearings power loss. In consequence, the main goal of this work is the development of a test rig to try out radial rolling bearings to know their behavior under different load stages. Notice that, rolling bearings are one of the widely used components in the mechanical world. In most machines, rotating parts are supported by many types of bearings which may have different requirements.

It is thought that the first use of rollers appeared about 5000 years ago, in the transition from sliding sledges to rolling, to transport building blocks. Later, with the wheel invention, the first rolling bearing emerged. Names like Aristotle, which made reference to the friction force; Marcus Vitruvius Pollio, a famous Roman engineer; and, later Leonardo da Vinci, the great genius also in the field of tribology, will be forever intrinsically connected to the rolling bearings. More recently, in the beginning of the twentieth century Richard Stribeck and John Goodman dominate the history of scientific studies of ball bearings (1898-1902) [2].

The high accuracy rolling bearings, such as those used nowadays, are product of the advancements in technology of the 20th and 21st Centuries [3]. Their main function is to transmit load at very low friction. However, the rolling bearings friction power loss can reach up to 30% of the power loss occurring inside a gearbox, being a major contribution to the overall energy consumption [4]. Therefore, the efficiency of the bearings is of extreme importance in achieving an efficient machine. The manufacturers are continually trying to improve their bearings design in order to reduce the power loss and operating temperatures, and to improve the lubrication conditions [5–9].

In a gearbox, for example, two types of power losses can be identified, the load and no-load dependent losses. The load losses are all the power losses that result directly from the effect of the applied torque and power transmission. The no-load losses are all the

power loss that can occur even when there is no torque applied [10]. There are, several semi-empiric models, that evaluate each one of the several components of the load and no load losses. For bearings the most recognized is the SKF model [11].

1.2 Dissertation proposal and objectives

For the development of a rolling bearings power loss model it is necessary to feed and validate with experimental values. In the case of radial bearings, there are no standardized machines available in the market. Therefore, CETRIB proposed a dissertation whose objective was the design of a radial rolling bearing test rig based on modified FZG carter.

The main objectives and constrains for the design of this rig are:

- Measuring capability of the torque loss;
- Simple and quick assembly of the rolling bearings;
- The shaft diameter should be in the range of 20mm to 30mm;
- The radial load applied to each bearing should be around 50% of C of the most loaded bearing;
- Possibility of imposing pre-load on tapered rolling bearings;
- The lubrication of the bearings will be performed with oil, and because of that, oil drain must be present;
- Possibility of controlling the oil bath temperature;
- Using the existing drive system (shown in Figure 1.1);
- The manufacturing process available is machining.

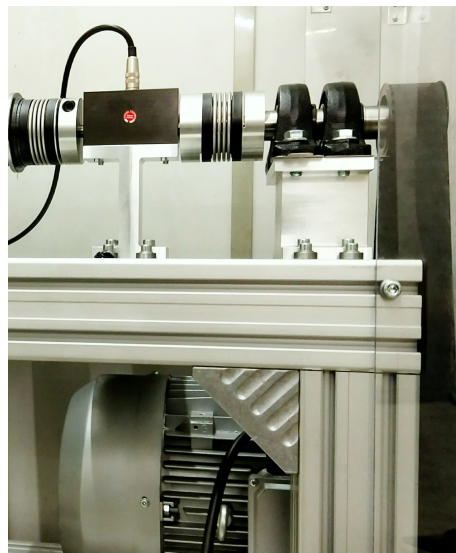


Figure 1.1: Drive system.

The motor has a power of 7.5 kW and torque cell has as application ranges 0-5/0-10/0-20/0-50 Nm.

The selection of possible rolling bearings to be tested was narrowed down to the list presented in Table 1.1 for a shaft diameter of 25 mm, and in Table 1.2 for the shaft diameter of 30 mm. The selection was performed, with SKF catalogue, taking into consideration the objectives proposed, the availability of rolling bearings, their price and also their internal and external diameter for which several types of rolling bearings types are available.

The shaft diameter of 20 mm was discarded because the relation of the load capacity of the bearings and the selection of the shaft was not very favorable. Comparing these with the rolling bearings with 25 mm of inner diameter, the last ones presents similar load capacity and the shaft is much more resistant.

Table 1.1: Rolling bearings types and their main characteristics for a 25 mm shaft [11].

Bearing designation	d mm	D mm	B mm	C kN	C_0 kN	Misalignment arcmin (°)	Type
6205*	25	52	15	14.8	7.8	10	ball bearing
7205 BEP	25	52	15	14.8	9.3	4	angular contact
1205 ETN9	25	52	15	14.3	4	150	self-aligning
NJ 205 ECP	25	52	15	32.5	27	4	cylindrical roller
30205 J2/Q	25	52	15	30.8	33.5	4	tapered roller

Table 1.2: Rolling bearings types and their main characteristics for a 30 mm shaft [11].

Bearing designation	d mm	D mm	B mm	C kN	C_0 kN	Misalignment arcmin (°)	Type
6206*	30	62	16	20.3	11.2	10	ball bearing
7206 BEP	30	62	16	22.5	14.3	4	angular contact
1206 ETN9	30	62	16	15.6	4.65	150	self-aligning
NJ 206 ECJ*	30	62	16	44	36.5	4	cylindrical roller
30206 J2/Q	30	62	16	40.2	44	4	tapered roller

As expected, the rolling bearings with more load capacity are the cylindrical roller bearings, which are written in bold in the Tables 1.1 and 1.2. However, the tapered roller bearings present a load capacity similar to the previous ones. The test rig should be then designed in order to test the tapered and cylindrical roller bearings.

The most common rolling bearings which will be tested are present in the Tables 1.1 and 1.2, however, provided that they maintain the same nominal dimensions their variants can also be tested such as the rolling bearings lubricated for life and others.

The rolling bearing types chosen, such as ball bearing, angular contact, self-aligning, cylindrical roller and tapered roller bearing respectively, are represented in Figure 1.2.

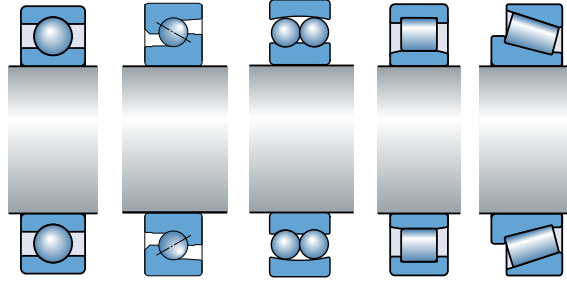


Figure 1.2: Rolling bearing types.

1.3 Dissertation outline

In the following topics, an overview of the various chapters will be presented:

Chapter 2

In chapter 2, a review of a state of the art of rolling bearings and test rigs is made. The many types of the rolling bearings available in the market and the criteria used to choose them are object of study. After this, the radial bearings test rigs found in literature with detailed information are presented as well as their importance to the development of a power loss model.

Chapter 3

Chapter 3 is dedicated to the definition and design of all tests rig parts. The main shaft behavior is analyzed in detail, from the static resistance to the dynamic behavior. This chapter also includes the design of the load application system as well as the selection of its components.

To complete the design, a thermal analysis of the test rig is made aiming to perform an oil bath temperature control system. The production drawing are presented in the final of the chapter.

Chapter 4

In chapter 4, the final solution of this test rig is presented. An exploded view and an overall drawing is shown. The test rig assembly and the test preparation assembly are shown step by step, being also given the appropriate assembly advice.

Chapter 5

The last chapter is dedicated to the conclusions based on the final test rig evaluating the objectives and requirements that it meets. Some future works are also suggested.

State of the art

2.1 Rolling Bearing

A rolling bearing is a precision-made component that allows relative movement between two bodies, usually applied to connect a fixed part to a moving part with low friction. Rolling and plain bearings hold rotating components, such as shafts. They transfer axial and radial loads from the source to the structure that support it. The bearings have been continuously studied because they must present maximum efficiency (the lowest possible power loss), reliability, durability, high precision and high performance, even at high speed.

2.1.1 Rolling bearing types and designs

There is a great variety of bearing types that can be found in the market. Two big groups can be formed, the radial bearings that support most of the load in the radial direction and the thrust bearings that support the load in the axial direction. Table 2.1 displays the main radial and thrust bearing types available in the market.

Table 2.1: Main radial and thrust bearing types.

Radial	Deep groove ball
	Y
	Angular contact ball
	Self-aligning ball
	Cylindrical roller
	Needle roller
	Tapered roller
	Spherical roller
	CARB toroidal roller
Thrust	Ball
	Angular contact ball
	Cylindrical roller
	Needle roller
	Spherical roller
	Tapered roller

Boundary dimensions

The boundary dimensions are the main dimensions of the bearing, presented in the Figure 2.1, which can easily be consulted in every catalogue.

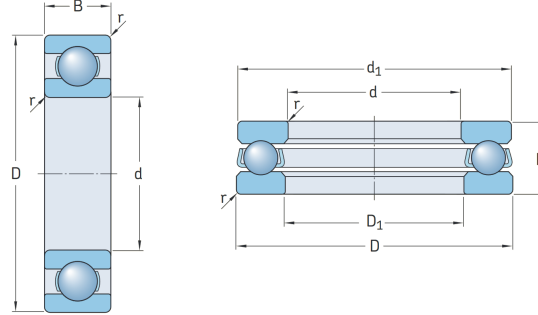


Figure 2.1: Main bearing dimensions[11].

According to the Figure, D is the external diameter, d is the internal diameter, B and H are the width and r is the fillet radius.

Basic selection criteria

Each bearing type has different characteristics, based on its design, which makes it more, or less, suitable for a given application.

Deep groove ball bearings can accommodate normal radial loads as well as axial loads of lower magnitude. These type of bearings are also available in the efficient performance class for high precision, high efficiency and quiet class [11].

Spherical and toroidal roller bearings can accommodate very high loads and have self-aligning properties. These properties make them popular for applications with severe requirements as heavy loads, shaft deflections and misalignment.

For a detailed and full bearing selection, some important topics need to be verified, such as load carrying capacity and direction, life, friction moment that intrinsically connected with the power loss and efficiency, permissible speeds, bearing internal clearance and pre-load, lubrication and sealing solutions.

Magnitude and direction of load

The magnitude of the load is one of the factors that usually determines the size and type of the bearing. Generally, roller bearings are able to support heavier loads than similar-sized ball bearings. Comparing cage and cageless (“full complement” of rolling elements) rolling bearings, the “full complement” can support heavier loads however they can only be used at low speed.

About direction of load, most of the bearings have a preferential load carrying direction, however, they can support relatively small loads in the direction normal to the preferential, as can be observed in Figure 2.2. There are also bearings that support high loads in radial and axial directions.

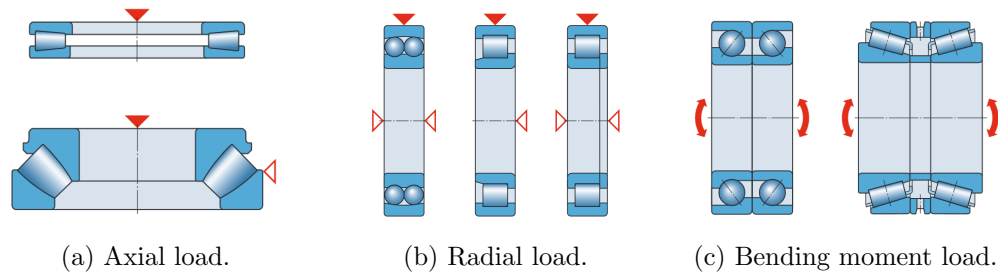


Figure 2.2: Examples of load direction in bearings[11].

The filled triangles represents direction with high capacity and the empty triangle represents that the bearing can support a low load in this direction.

Misalignment

Angular misalignment between the shaft and the housing occurs when the shaft deflects under operating loads. Rigid bearings as deep groove ball and cylindrical roller bearings can accommodate only a few minutes of arc of misalignment without damaging the bearing. Self-aligning ball, spherical roller and toroidal roller bearings can accommodate shaft deflections as well as initial misalignment.

Precision

The precision is given by the tolerance classes for service precision and the dimensional precision. The manufacturers provide some bearings with super-precision for careful applications.

Speed and Friction

The permissible operating temperature determines the limits on the speed at which rolling bearings can be lead on service. Bearing types that operate with low friction generate low frictional heat and therefore are most suitable for high-speed operation. Deep groove ball and self-aligning bearings are therefore used at higher service speeds, when the loads are purely radial. Angular contact ball bearings are usually used when there are combined loads.

Although the bearings are called “anti-friction” they have some frictional losses due to the elastic deformation of the rolling elements and raceways under load (sliding friction). They also generate sliding friction between the rolling elements and the cage, flanges and guide rings, and between seals, if any, and their counterface. Friction in the lubricant (rolling friction) also contributes to the total frictional moment.

2.2 Rolling bearing test rigs

2.2.1 Rolling bearing test rig and power loss model

It is of utmost importance to know the power losses generated inside a transmission. For this propose a power loss model is applied which consists in evaluating the power loss sources. One of the most important power loss sources are the rolling bearings, for which experimental measurements of total friction torque loss are needed to be applied in the

SKF model, in order to improve it, although it already gives a very good approximation of the total friction torque. Then it is possible to know the sliding friction torque by subtracting all friction torques to the total, as can be understood with the equation (2.1), and finally reach the coefficient of sliding [11].

$$M_t = M_{rr} + M_{sl} + M_{seal} + M_{drag} \quad (2.1)$$

Where M_t is the total frictional moment, M_{rr} is the rolling frictional moment, M_{sl} is the sliding frictional moment, M_{seal} is the frictional moment of the seals and M_{drag} is the frictional moment of drag losses, like churning, splashing and others.

The rolling bearings test rig allows to study the M_{rr} , M_{sl} and M_{drag} parameters for each test conditions. Changing the oil and varying the immersion height, different parameter values will be obtained.

2.2.2 Existing rolling bearings test rigs

After some literature search about radial bearings test rigs that are capable to evaluate the rolling bearing torque loss some were found that corresponds to the requirements. Since none are normalized, the access to the detailed information was more difficult.

The first test rig found, presented with more detail in the literature and the one that gave frame to this work was developed in Sweden by Harström, Jonsson, Lind, Magnerius and Strand, which is presented in the Minghui Tu Master Thesis [12]. This rig was designed to measure the friction torque from radial rolling bearings under no load and loaded condition. As previously explained in Chapter 1, to measure the torque loss the same test condition must be verified on each bearing. For this, *Strand et al.* proposed a test rig based on FZG gear test rig carter, modifying it so that four rolling bearings can be placed, two of them in the housing and other two in the center of the shaft involved by a load application sleeve. The main idea is based on the four point bending, where the two inner rolling bearings receives the force, half for each, and the two outer rolling bearings which are positioned in the housing are subjected to the same force due to the reactionary load. The bearing location is shown in the Figures 2.3 and 2.4a. The concept chosen to apply the force was the traverse crane push, Figure 2.4b, which is simple but usually a complex system and implies to have exact weights to apply force, which implies to have a large variety of weights to apply different loads.

According to Tu's master thesis [12], based on the analysis in the report of *Strand et al.*, the test rig can be loaded from 0.6 to 9 kN, rotation speed between 50 to 3500 rpm, oil temperatures between 30 to 120 °C and test both dip and spray lubrication.

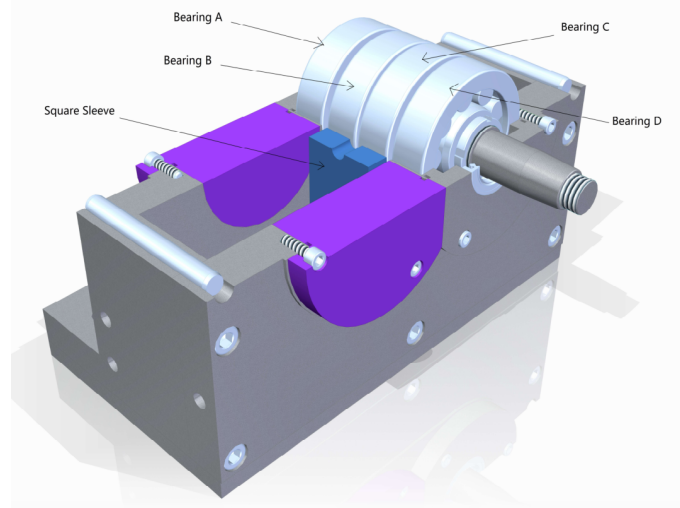
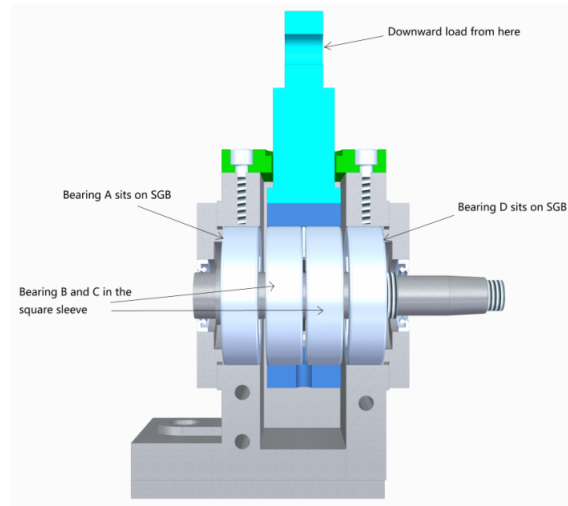
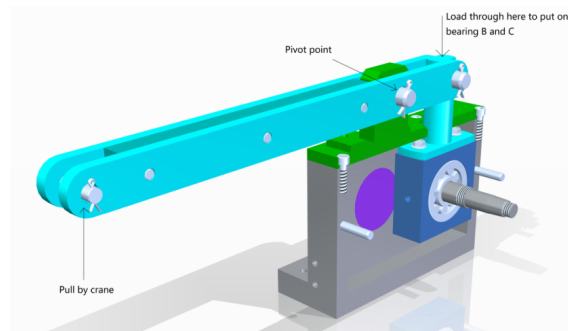


Figure 2.3: Bearing location in test rig proposed by *Strand et al.* [12].



(a) Cut view of the test rig.



(b) Load application type.

Figure 2.4: Test rig proposed by *Strand et al.* [12].

The second test rig found in literature was presented in the paper “Rolling-Element Bearing Heat Transfer—Part III: Experimental Validation” [13]. In the “Experimental

Setups” chapter the radial rolling bearings test rig that was used was found. As the previous one, this test rig is based on four-point bending, as the Figure 2.5 shows. It allows to test four different types of rolling bearings, ball, cylindrical, spherical, and tapered bearings. This test rig was designed for each bearing dimensions of 60 mm bore diameter, 100 mm outside diameter, and an average width of 23 mm. As explained for the previous test rig, each test must be performed with four identical bearings (e.g., four double-row spherical bearings). The applied radial load, induced by a Honeywell 060-0743-06TJL pressure transducer, acted upward on the two center rolling bearings. The reactionary downward load was supported by the left and right flanking rolling bearings. The motor rested on a Lebow Model 2528 with 11.3 Nm strain gage nominal. The difference between the motor input torque and nominal torque is the torque loss. Thus, the difference was only the cumulative torque of the four symmetrically loaded rolling bearings and so the torque loss for each rolling bearing is achieved by dividing the torque loss by four rolling bearings.

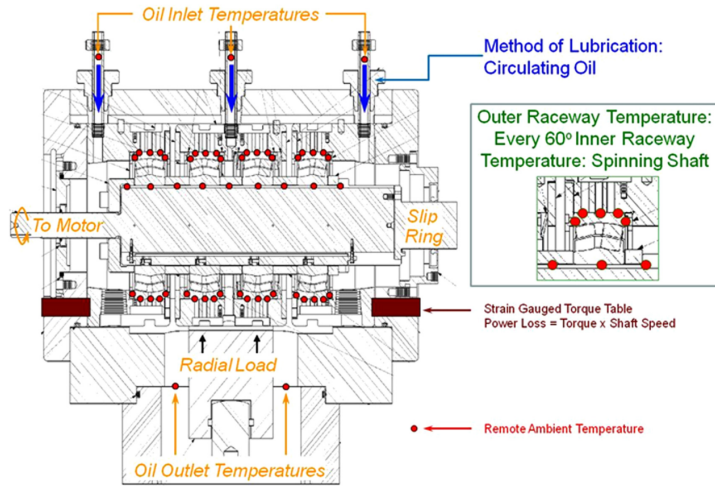


Figure 2.5: Test rig scheme proposed by *Hannon et al.* [13].

Test rig design

The initial idea was to build a carter based on FZG test rig carter, to test radial bearings. Since CETRIB does not have easy access to precision welding manufacturing, neither casting, it was decided to build a carter by bolted construction. This kind of construction allows the high quality manufacturing with low costs, however, it implies very strict assembly fits in order to assure the oil sealing and the alignment requirements for the rolling bearings operation.

3.1 First draft and part list of the test rig

As aforesaid, the objectives presented in the Section 1.2 are now presented adapted to the present test rig proposal. The main requirement of this test rig is to promote equal power loss on each rolling bearing keeping in mind that the simple and quick mounting of rolling bearings are also required. Therefore, to promote the same power loss in each rolling bearing the same operating conditions must be observed on all the four rolling bearings, i.e. the same radial force applied and same lubrication conditions. This radial rolling bearings test rig is based on the four-point bending in a symmetric shaft, where the force is directly applied to each of the two inner rolling bearings. In order to maintain the static equilibrium, the two outer rolling bearings are loaded with an identical force in opposite direction. This concept is similar to the one outlined in Tu's master thesis [12], based on FZG test rig carter. This idealization needs to be designed to maximize the range of applicability with the most accurate results possible.

As seen previously in Chapter 2, in the literature there are many test rigs capable of testing radial rolling bearings. These test rigs are also based on four point bending, and using this concept, the load can be applied in two different directions. The most common direction is the vertical, which is generally applied using the dead load principle. This option was discarded because, looking at the Figure 3.1a considering the oil level by the middle of the bearing and with the load applied downward on two inner rolling bearings, the active contact zone between the raceways and the rolling bodies occurs in the top (2) of the rolling bearing, leading to a contact area outside the oil bath, while the two outer rolling bearings have the active contact zone in the bottom (1) of the rolling bearing, dipped in the oil bath. This occurrence leads to different lubrication conditions in the four bearings, thus making it difficult to identify the weight of each rolling bearing in total power loss.

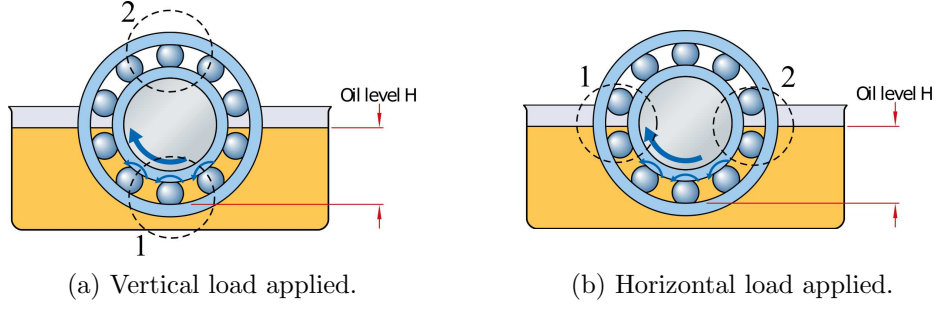


Figure 3.1: Bearing oil feed in contact zone.

In order to obtain similar power loss in all bearings, it was decided to apply the load horizontally. This option seems more sustainable. By the analysis of the Figure 3.1b considering the oil level by the middle of the bearing and with the load applied laterally on two inner rolling bearings, the active contact zone between the raceways and the rolling bodies occurs in one side (1) of the rolling bearing, while the two outer rolling bearings have the active contact zone in the other side (2) of the rolling bearing. Even so, the lubrication conditions are not the same in the four bearings because in contact zone (1) the rolling element are dipped in the oil bath before the contact, while in the contact zone (2) the rolling element going to bath after the contact zone. However, the lubrication conditions are more similar in the four rolling bearings with horizontal load than with vertical load.

In order to obtain a wider range of test conditions, an oil bath temperature control should be integrated in this test rig. The temperature control allows to perform tests at a different temperature of the stabilization temperature of the oil.

The first draft of the test rig is presented in the Figure 3.2, with the respective part list shown in Table 3.1. This kind of arrangement was chosen according to the simplicity of assembly and the machining manufacturing process, which is the process available.

Table 3.1: Part list of test rig first draft.

Item Number	Item Name
1	Front Wall
2	Base Plate
3	Right Wall
4	Hydraulic Cylinder
5	Rod
6	Back Wall
7	Rod End
8	Pin
9	Left Wall
10	Yoke
11	Rolling Bearings Sleeve
12	Rolling Bearings
13	Spacer
14	Main Shaft
15	Hub

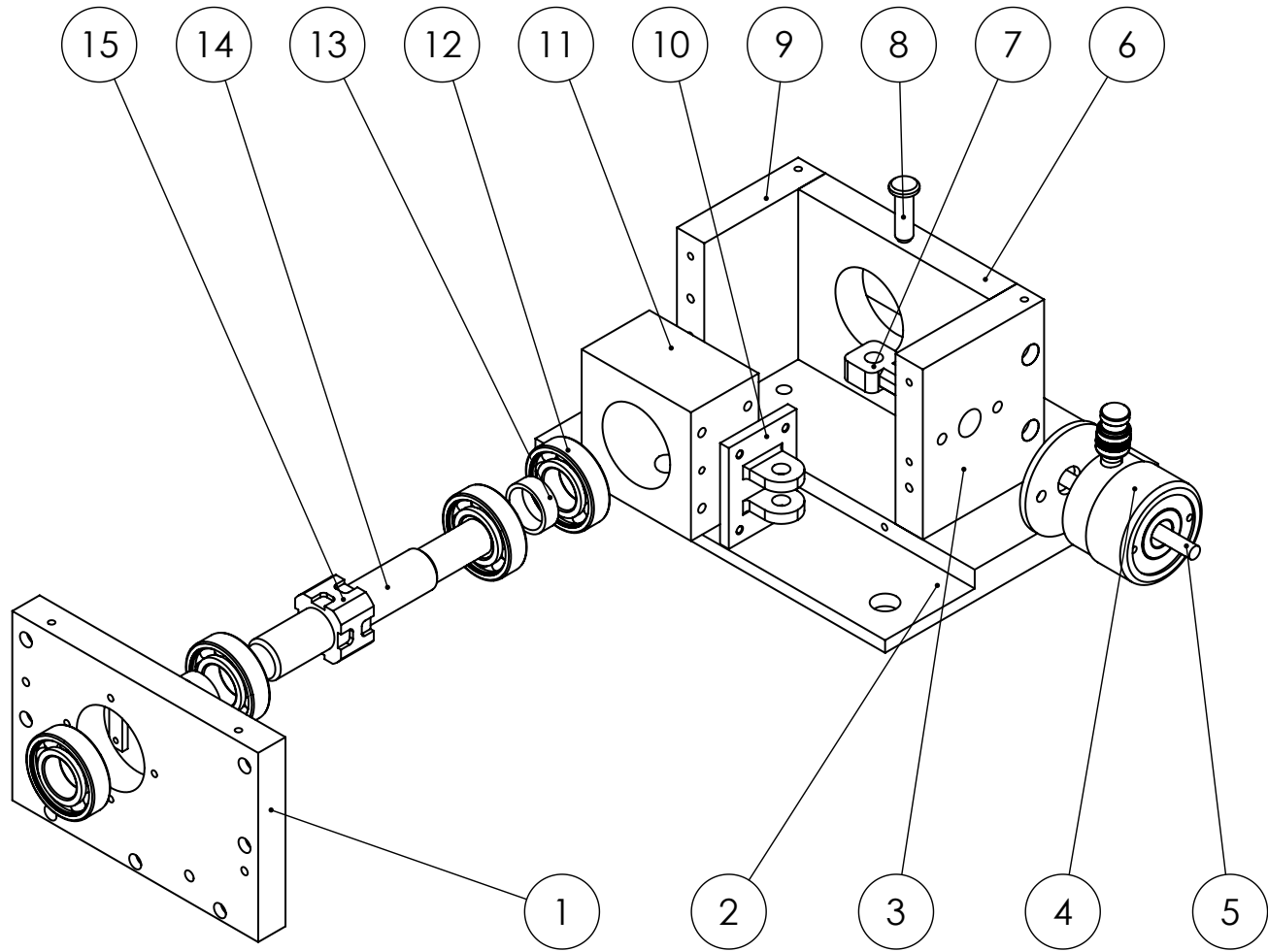


Figure 3.2: Test rig first draft.

3.2 Parts design

3.2.1 Main shaft and hub - Parts 14 and 15

The main shaft, represented by the part number 14 in Figure 3.2, is the most important part of this test rig. It supports the high loads applied on the rolling bearings and must support them without damage. The shaft design was performed for infinite life cycle under maximum load conditions while the diameter was maintained in the range of 20 to 30 mm, as was shown in the problem objectives and constraints in Section 1.2. This design was based on the book "Shigley's Mechanical Engineering Design" [14] and MATLAB® was used to implement the calculation routines. Then KISSsoft® Software, that is a powerful tool for the design of mechanical components was used, in addition with the FKM Guideline "Analytical Strength Assessment" [15] to verify the solution.

The geometry of main shaft with the press-fit hub is represented in Figure 3.3. The rolling bearings, represented by the part number 12 in Figure 3.2, are mounted two by each side and are separated therebetween by a spacer which is represented by the part number 13 in Figure 3.2.

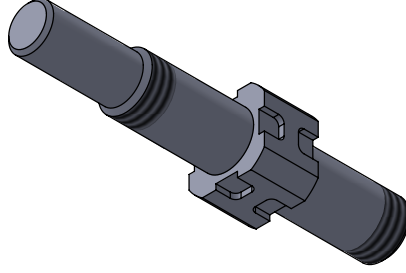


Figure 3.3: Isometric view of main shaft.

Considering that the shaft is supported by the center of the outer rolling bearings positioned in A and A', the load is applied in the center of the inner rolling bearings B and B', the shaft loading is presented in the Figure 3.4.

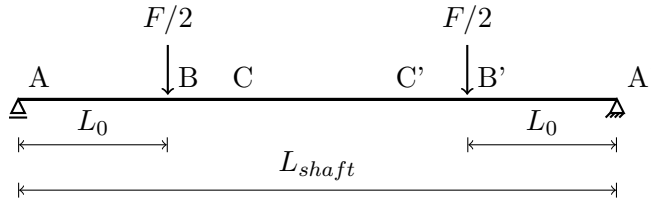


Figure 3.4: Shaft loading scheme.

Where L_{shaft} is the length of the shaft between the center of outside rolling bearings and L_0 is the length between the center of the outer and inner rolling bearings.

As this is a typical problem of four-point bending in a symmetric shaft, the L_0 distance must be chosen carefully because with the increase of L_0 , the maximum stress as well as deflection and rotation will increase too. The detailed analysis of four-point bending problem is made in the Appendix D.

Between the C-C' there is an interference fit hub, which gives some important characteristics to the shaft, as will be explained later in this chapter. The hub was chosen instead of a shoulder in order to reduce the machining cost and the waste of material.

The most severe case, presented in the Table 3.2, was defined according to the rolling bearing with higher load capacity which is cylindrical roller bearing.

Table 3.2: Design conditions for the main shaft.

d_{shaft} mm	F kN	L_{shaft} mm	L_0 mm	B mm
25	24	102	28	15
30	40	102	28	16

It was required to be able to test radial rolling bearings without having to destroy in the removal, this shaft must have space between the rolling bearings to use a extractor tool to remove them. Even so, it is necessary to manufacture a extractor tool that reaches the inner raceway in order to remove the rolling bearing safely. Considering the SKF extractor tool as guide model, the clamp has 9 mm length which means that the free space between the rolling bearings should be at least 12 mm, which is the length of the spacer. Adding to this length half the thickness of two rolling bearing, the L_0 should be at least 28 mm.

In this report the calculation method for the shaft with 30 mm of diameter is presented. The following process can also be applied to the 25 mm shaft.

An overview of the shaft with hub is presented in the Figure 3.5.

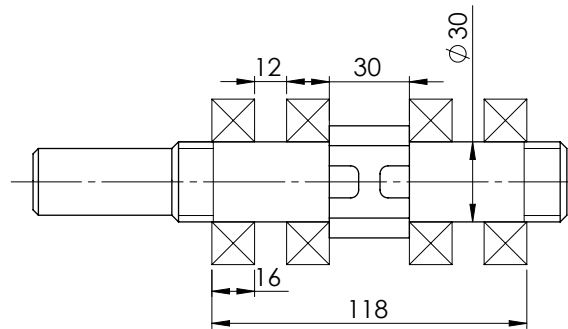


Figure 3.5: Dimensions of shaft with hub.

Load Diagrams

Shear load diagram

The shear diagram of four-point bending problem in a symmetric shaft with equal force applied at two points with equal distance to the rolling bearings that support it, presents two steps, one positive and another negative in the first and third zone. In the center, the shear force cancels. This behavior is shown in the Figure 3.6.

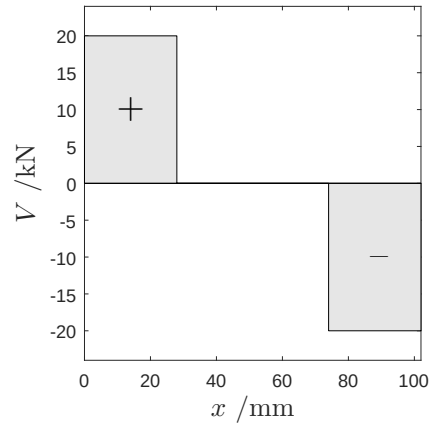


Figure 3.6: Shear load diagram for $F=40$ kN.

Bending moment diagram

The bending moment diagram is obtained by the integration of the shear force. Therefore, it is represented in the Figure 3.7.

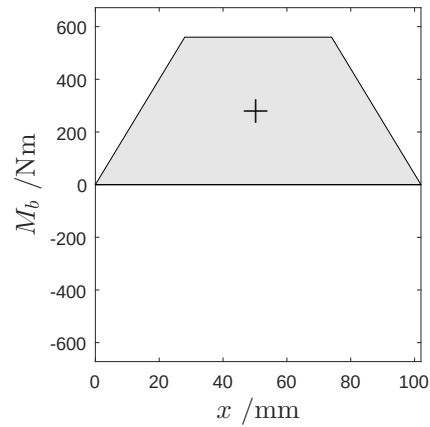


Figure 3.7: Bending moment diagram for $F=40$ kN.

The maximum bending moment occurs in the central zone, between the applied forces and its value is 560 N·m.

Deflection and Rotation Diagrams

The shaft rotation is obtained by the integration of the bending moment, being its evolution along the shaft represented by Figure 3.8a. The shaft deflection is obtained by the second integration of the bending moment, as shown in Figure 3.8b.

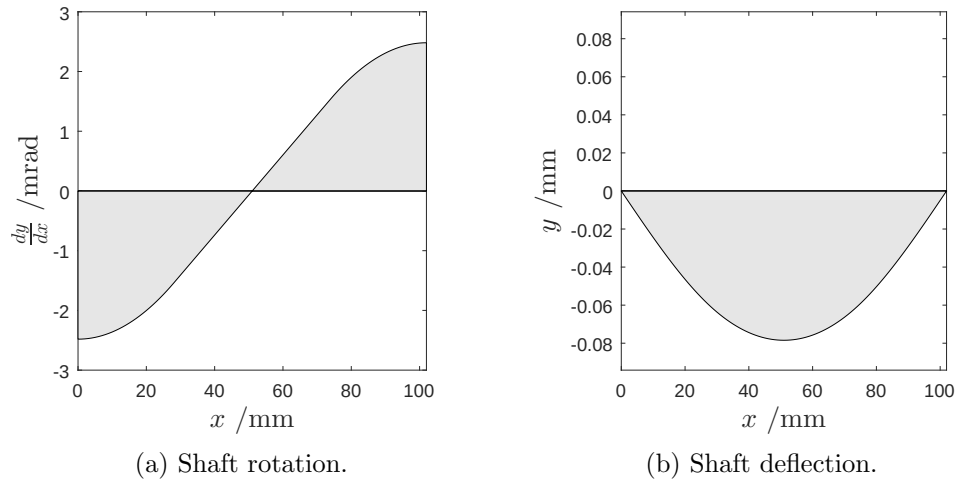


Figure 3.8: Shaft behavior for 40 kN.

The maximum rotation of the shaft is about 2.5 mrad (8.59 arcminute). This shows that the maximum rolling bearing misalignment tolerance, shown in Table 1.2, is not respected for the most severe case. However, according to the manufacturer it is possible to use larger misalignment between the raceways with reduced rolling bearing service life. Since the main objective of the test rig is not to test the rolling bearings fatigue life, the misalignment for maximum load is not crucial factor.

The maximum deflection obtained for 40 kN applied is close to 0.08 mm.

Static Calculation

The static calculation process begin with the determination of the stress state and possible critical sections. Then the equivalent stress is calculated by Tresca or von Mises criteria. In the end, the static safety factor is achieved by the ratio between yield stress and the applied stress.

Equivalent static stress

Considering a ductile material, the equivalent stress can be calculated by von Mises's equivalent stress. Also, the equivalent shear stress can be calculated by the Tresca's equivalent shear stress.

The installed normal stress is given by the equation (3.1). The equation (3.2) give the shear stress in the section.

$$\sigma_y = \frac{32 \cdot M_b}{\pi \cdot (d_{shaft})^3} \quad (3.1)$$

$$\tau_{yz} = \frac{16 \cdot V}{3 \cdot \pi \cdot (d_{shaft})^2} \quad (3.2)$$

von Mises's failure criterion

The von Mises's failure criterion was based on the distortion-energy theory, which predicts that yielding occurs when the distortion strain energy per unit volume reaches

or exceeds the distortion strain energy, per unit volume, for yield in simple tension or compression of the same material.

The von Mises equivalent stress for the elemental volume is given by equation (3.3).

$$\sigma_{vM}^{eq} = \sqrt{\frac{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6 \cdot (\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)}{2}} \quad (3.3)$$

If $\sigma_x = \sigma_z = \tau_{xy} = \tau_{zx} = 0$, the equation (3.3) comes simplified, von Mises's equivalent stress is now given by equation (3.4).

$$\sigma_{vM}^{eq} = \sqrt{\sigma_y^2 + 3 \cdot \tau_{yz}^2} \quad (3.4)$$

Tresca's failure criterion

Tresca's criterion is based on the maximum shear stress, which can be found with Mohr's circle on plane YZ. The principal shear stresses can be obtained after some geometric manipulations, equation (3.5).

$$\tau_{Tr}^{eq} = \tau_1 = \sqrt{\left(\frac{\sigma_y - \sigma_z}{2}\right)^2 + \tau_{yz}^2} \quad (3.5)$$

If $\sigma_z = 0$, the equation (3.5) comes simplified, Tresca's equivalent stress is now given by equation (3.6).

$$\tau_{Tr}^{eq} = \sqrt{\left(\frac{\sigma_y}{2}\right)^2 + \tau_{yz}^2} \quad (3.6)$$

It is possible to manipulate the equation (3.6) to present it in terms of maximum normal stress as shown in equation (3.7).

$$\sigma_{Tr}^{eq} = \sqrt{\sigma_y^2 + 4 \cdot \tau_{yz}^2} \quad (3.7)$$

The difference between the permissibility of the von Mises's and Tresca's criteria is represented in Figure 3.9, where it is easily understood that Tresca's criterion is less permissible, since its safety area is smaller and therefore the design will be performed according with it.

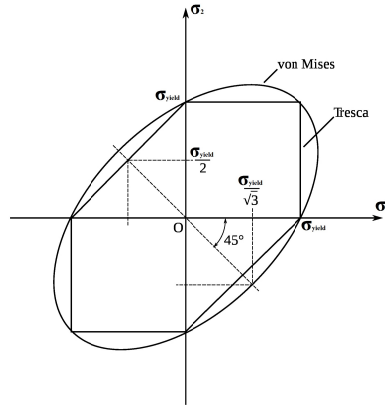


Figure 3.9: Comparison of failure criteria.

Note that the shear stress due to torsion was neglected because its contribution is very small. In order to prove that, the Table 3.4 shows the torque loss by the bearings, calculated with SKF bearing calculator tool, for the most severe test conditions, shown in Table 3.3.

Table 3.3: Rolling bearing most severe test conditions.

Velocity rpm	Oil type	Oil level position	Oil temperature °C	Lubrication type
3000	ISO VG 32	Center of the bearing	100	Oil bath

Table 3.4: Torque loss per rolling bearings at test conditions presented in Table 3.3.

Bearing designation	Type	C kN	Applied Load kN	Torque loss Nm
6206*	ball bearing	20.3	10	0.54
7206 BEP	angular contact	22.5	10	0.74
1206 ETN9	self-aligning	15.6	7	0.07
NJ 206 ECJ*	cylindrical roller	44	20	0.18
30206 J2/Q	tapered roller	40.2	20	0.61

The torque distribution along the shaft, considering the rolling bearing with more torque loss which is the angular contact rolling bearing, is represented in Figure 3.10.

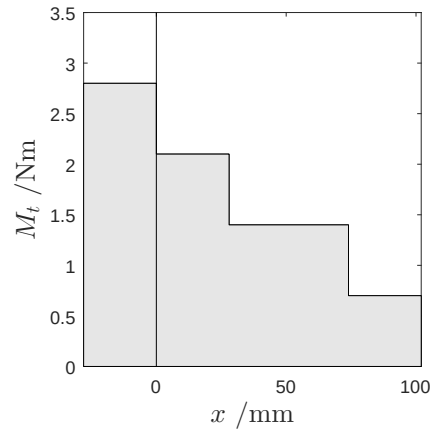


Figure 3.10: Torque loss along the shaft.

Where the negative part of x represents the shaft connection with the drive system, being $x = 0$ the center of the first rolling bearing.

The shear stress due to torsion can be calculated by equation (3.8).

$$\tau = \frac{16 \cdot M_t}{\pi \cdot d_{shaft}^3} \quad (3.8)$$

The shear stress due to torsion along the shaft is represented in Figure 3.11.

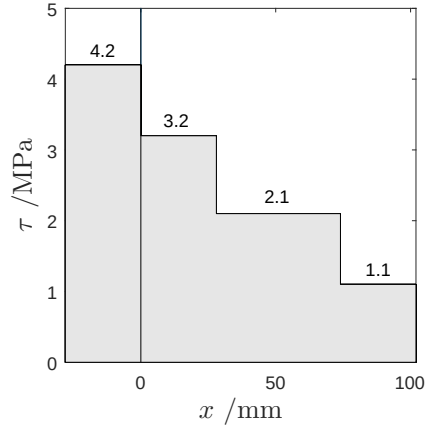


Figure 3.11: Torque loss along the shaft.

Thus, it is proved that the torsion has negligible influence in the shaft stress state.

Static safety factor

The static safety is the first point that usually is verified, however the fatigue safety is usually the determining factor for the component life. Considering the static safety factor in the critical section as the ratio between the material yield stress and installed equivalent stress, as in equation (3.9). Considering a static load factor l_f which is a factor recorded in the DIN 743 standard that modifies the applied load usually between 1.0 and 2.0 times higher than the nominal load, it can be applied to the installed stress due to the direct proportionality between the load and the stress.

$$S_{f,static} = \frac{R_{pt}}{l_f \cdot \sigma^{eq}} \quad (3.9)$$

Assuming $l_f = 1.5$, which is the mean value of the range indicated above. Using the 34 CrNiMo 6 steel to build the shaft, for which the properties are shown in the Table C.2, presented in the Appendix C. The safety factor, using von Mises's and Tresca's criteria is displayed in table 3.4.

Table 3.5: Static equivalent stress by two failure criteria for critical sections.

Section	V kN	M_b Nm	σ_y MPa	τ_{yz} MPa	σ_{vM}^{eq} MPa	σ_{Tr}^{eq} MPa	R_{pt} MPa	$S_{f,static,vM}$	$S_{f,static,TR}$
B and B'	20	560	211.3	37.7	221.2	224.3	1000	3.01	2.97
C and C'	0	560	211.3	0	211.3	211.3		3.15	3.15

The stress distribution along the shaft, calculated by von Mises criterion, is represented in Figure 3.12.

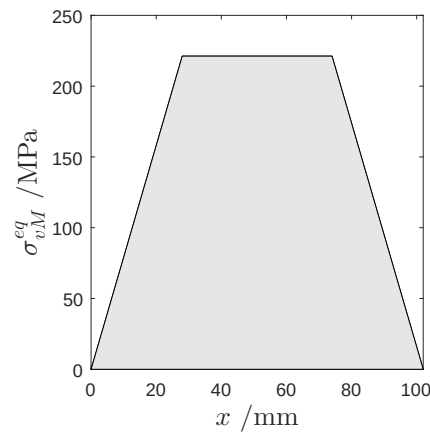


Figure 3.12: Equivalent stress distribution along the shaft.

Fatigue calculation

The fatigue calculation process begins with the determination of the stress state and possible critical sections. Then, the location of possible stress concentrations is analyzed and its concentration factors are calculated. With this it is possible to reach the equivalent state by von Mises's or Tresca's criteria as well as the critical section is determined. The material properties must also be modified, applying fatigue related coefficients, to calculate a more realistic endurance limit. The process ends with comparison of the equivalent stress installed in the critical section with the material endurance limit, which gives the fatigue safety factor.

The press-fit hub and the stress concentration factor

The shaft has a hub, mounted with interference fit, in the center. This hub has three fundamental roles, being them, the horizontal positioning of the rolling bearings, gives resistance to the center of the shaft, decreasing the deflection. Fundamentally, it has a geometry that was designed to allow the use of a spanner to hold the shaft while the locknuts are tightened. The surface grooves allow the extractor to reach to the inner raceway of the rolling bearings. The hub is displayed in the Figure 3.13.

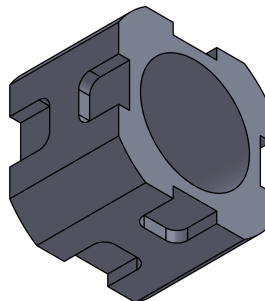


Figure 3.13: Hub

The hub design was made in KISSsoft®, the report can be consulted in the Appendix E.3, considering that the material of manufacture is a 34 CrNiMo 6 steel.

This press fit creates a stress concentration in the edge of the hub, which is difficult to account accurately. *Croccolo et al.* [16] proposed a parametric model, obtained with FEM analysis, that gives a stress concentration factor in compressed-fit couplings.

According to Croccolo's model, for a solid shaft with a hub mounted with interference fit, the stress concentration factor, K , is independent of the interference level.

$$K = \frac{\sigma_{VM,max}^{eq}}{\sigma_{VM}^{eq}} \quad (3.10)$$

K is given by the equation (3.10), where $\sigma_{VM,max}^{eq}$ represents the maximum equivalent von Mises stress in the critical zone and the σ_{VM}^{eq} represents the equivalent von Mises stress out of the critical zone. The expected ratio was $\frac{\sigma_{y,max}}{\sigma_y}$, where the stress is expressed in terms of y direction, but K can give a very good approximation.

Being Q_A the ratio between the hub internal diameter (equal to the shaft diameter) and the hub external diameter, as shown in equation (3.11), the stress concentration factor can be interpolated by the third order polynomial equation (3.12), with errors always lower than 5%.

$$Q_A = \frac{d_{i,hub}}{d_{e,hub}} \quad (3.11)$$

$$K = 0.982 \cdot (Q_A)^3 - 2.063 \cdot (Q_A)^2 + 1.048 \cdot Q_A + 1.023 \quad (3.12)$$

The Table 3.6 shows K values for the two different simple annular sections:

Table 3.6: Stress concentration factor in the hub sections.

$d_{ext,hub}$ mm	Q_A	K
36	0.83	1.03
42	0.71	1.08

Assuming that the hub has a simple annular section with 42 mm of external diameter, which corresponds to the bigger K . However it is not true, the hub has three annular sections, two of them have 36 mm and the central has 42 mm which gives less concentration factor than considered.

Dynamic stress state

The fatigue failure is expected to be the critical case and for which the shaft should be designed. So the dynamic analysis must be performed.

As in this problem the load applied is the same during all the test, the solicitation is pure alternated around the neutral axis, as the Figures 3.14a and 3.14b shows, the point P assumes successive positions that request it under traction and compression in each cycle. Therefore, as $\sigma_{max} = -\sigma_{min}$, it means that the $\sigma_m = 0$. As the shear, V , is of constant value, this leads to $\tau_m \neq 0$ and $\tau_a = 0$.

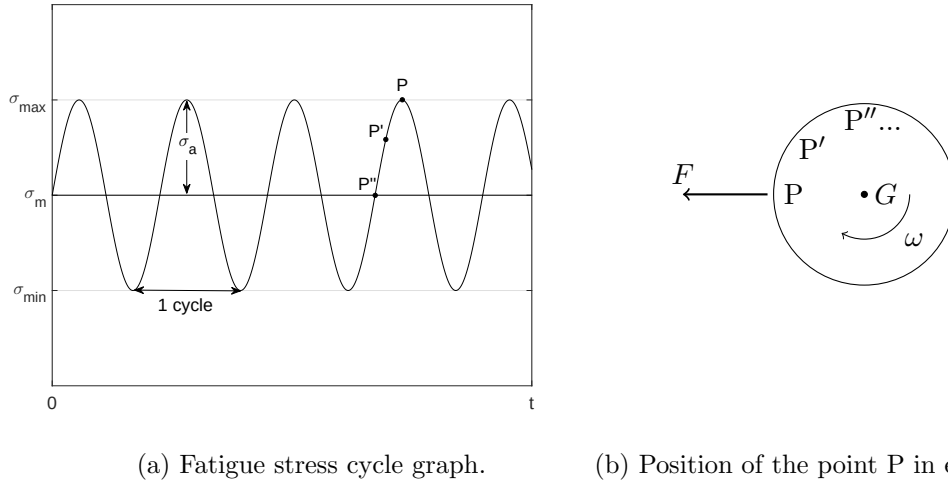


Figure 3.14: Fatigue stress cycle.

Being σ_a the alternating stress, σ_m the midrange stress, τ_a the alternating shear stress and τ_m the midrange shear stress, respectively represented in equations (3.13)-(3.16).

$$\sigma_a = \frac{32 \cdot M_a}{\pi \cdot (d_{shaft})^3} \quad (3.13)$$

$$\sigma_m = \frac{32 \cdot M_m}{\pi \cdot (d_{shaft})^3} \quad (3.14)$$

$$\tau_a = \frac{16 \cdot V_a}{3 \cdot \pi \cdot (d_{shaft})^2} \quad (3.15)$$

$$\tau_m = \frac{16 \cdot V_m}{3 \cdot \pi \cdot (d_{shaft})^2} \quad (3.16)$$

Where M_a and M_m represent respectively the maximum and the midrange bending moment, and where τ_a and τ_m represent the maximum and midrange shear force respectively.

Equivalent fatigue stress

Combining these stresses in accordance with the distortion energy failure theory, the base of von Mises criterion, which is suggested in the book “Shigley’s Mechanical Engineering Design” [14]. The von Mises equivalent stress for shafts in rotation is given by the equation (3.17).

$$\sigma_{a,vM}^{eq} = \sqrt{(K_f \cdot \sigma_a)^2 + 3 \cdot \tau_m^2} \quad (3.17)$$

Table 3.7: Fatigue equivalent stress for critical sections.

Section	V kN	M_a Nm	K_f	σ_a MPa	τ_m MPa	$\sigma_{a,vM}^{eq}$ MPa
B	20	560	1.00	211.3	37.7	221.2
C	0	560	1.08	211.3	0	228.2

Being $K_f = K$ according to the results obtained in Table 3.6.

The fatigue stress distribution along the shaft is represented in Figure 3.15.

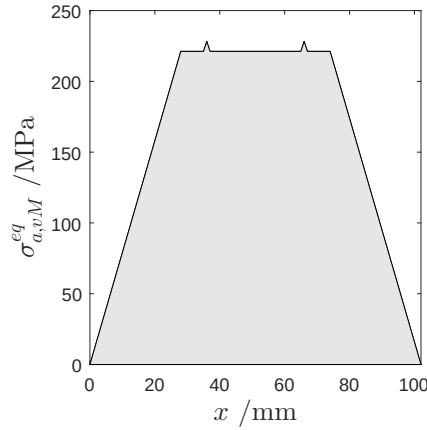


Figure 3.15: Fatigue equivalent stress distribution along the shaft.

Endurance limit

The endurance limit is a fatigue strength, S'_e , that in steel is usually equal to 50% of R_m of the material. However, the rotating-beam specimen used in the laboratory to determine endurance limits is prepared very carefully and tested under closely controlled conditions. So according to the book “Shigley’s Mechanical Engineering Design” [14], it is unrealistic to expect the endurance limit of a mechanical part to match the values obtained in the laboratory. Some differences affecting the life of the beam include:

- Material: composition, basis of failure, variability
- Manufacturing: method, heat treatment, fretting corrosion, surface condition, stress concentration
- Environment: corrosion, temperature, stress state, relaxation times
- Design: size, shape, life, stress state, speed, fretting

Marin [14] identified factors that quantify the effects of surface condition, size, loading, temperature, and miscellaneous items. Thus, the value of the endurance limit is modified by this factors which give a better approximation to the real value. Marin equation is written as the equation (3.18) shows.

$$S_e = k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e \quad (3.18)$$

Where the modification factors are k_a for surface condition, k_b for the size, k_c for the load, k_d for the temperature, k_e for the reliability and k_f for miscellaneous-effects.

k_a surface modification factor

The surface of a rotating-beam specimen is highly polished, with a final polishing in the axial direction to smooth out any circumferential scratches. The surface modification factor depends on the quality of the finish of the actual part surface and on the tensile strength of the part material. To find quantitative expressions for common finishes of machine parts (ground, machined, or cold-drawn, hot-rolled, and as-forged), the data points captured from a plot of endurance limit versus ultimate tensile strength by Lipson and Noll were reproduced by Horger in the form of an mathematical expression, presented in “Shigley’s Mechanical Engineering Design” [14], represented by the equation(3.19).

$$k_a = a \cdot (R_m)^b \quad (3.19)$$

Where for ground surface finish, a and b are given by the equations (3.20) and (3.21).

$$a = 1.58 \quad (3.20)$$

$$b = -0.085 \quad (3.21)$$

k_b size modification factor

The size factor has been evaluated using 133 data points presented in “Shigley’s Mechanical Engineering Design” [14]. The results for bending on the shaft until 51 mm diameter, may be expressed as equation (3.22).

$$k_b = 1.24 \cdot (d_{shaft})^{-0.107} \quad (3.22)$$

k_c load modification factor

When fatigue tests are carried out with rotating bending, axial (push-pull), and torsional loading, the endurance limits differ with R_m . This is discussed in section 6–17 of the “Shigley’s Mechanical Engineering Design” [14]. Here, it is specified average value of the load factor as equation (3.23).

$$k_c = 1, \text{ for bending} \quad (3.23)$$

k_d temperature modification factor

When operating temperatures are below room temperature, brittle fracture is a strong possibility and should be investigated first. When the operating temperatures are higher than room temperature, yielding should be investigated first because the yield strength drops off quickly with temperature. Any stress will induce creep in a material operating at high temperatures, so this factor must be considered too. For service temperature around 100°C, the k_d is shown in equation (3.24).

$$k_d = 1.020 \quad (3.24)$$

k_e reliability modification factor

Most endurance strength data are reported as mean values. The data presented by Haugen and Wirching in “Shigley’s Mechanical Engineering Design” [14] show standard deviations of endurance strengths of less than 8%. Thus the reliability modification factor to account for this can be written as in the equation (3.25).

$$k_e = 1 - 0.08 \cdot z_a \quad (3.25)$$

For reliability at least 99.9%, the z_a value is shown in (3.26).

$$z_a = 3.091 \quad (3.26)$$

k_f miscellaneous-effects modification factor

This modification factor is referent to the other conditions as corrosive atmosphere, cyclic frequency for high temperatures and corrosives and fretage corrosion that is a phenomenon that is the result of microscopic motions of tightly fitting parts or structures like rolling bearing races. To take into account the fretage corrosion, the k_f should be from 0.24 up to 0.90. So, the value used is $k_f=0.90$ because the test ambience is not very aggressive and the other factors have already modified a lot the endurance limit for the safety side.

Modified endurance limit

Finally, with the modification factors presented from the equation (3.19) to (3.25), the modified endurance limit value, S_e , was calculated and displayed in Table 3.8.

Table 3.8: Endurance limit calculation.

R_m MPa	S'_e MPa	k_a	k_b	k_c	k_d	k_e	k_f	S_e MPa
1200	600	0.864	0.862	1.000	1.020	0.752	0.9	308.5

Comparing the S_e with S'_e , it may be noted that the modified value, presented in the Table 3.8 is almost 50% smaller, which gives more safety to our design.

Fatigue safety factor

First of all, equivalent von Mises stress is compared with the yield stress of the material, equation (3.27), because the ASME fatigue criterion does not guards against any yield failure, but only against fatigue failure.

$$S_{f,yield} = \frac{R_{pt}}{\sigma_{a,vM}^{eq}} \quad (3.27)$$

Verified the yield, the fatigue safety factor is calculated by ASME Elliptic criterion presented in equation (3.28).

$$\frac{1}{S_{f,ASME}} = \frac{16}{\pi \cdot (d_{shaft})^2} \cdot \sqrt{4 \left(\frac{K \cdot M_a}{d_{shaft} \cdot S_e} \right)^2 + \frac{1}{3} \left(\frac{V_m}{R_{pt}} \right)^2} \quad (3.28)$$

The results of the safety factors for yield stress and ASME criteria are exhibit in the Table 3.9:

Table 3.9: Fatigue safety factors.

$S_{f,yield}$	$S_{f,ASME}$
4.38	1.35

As expected the fatigue limit is the most conservative criterion. The ASME Elliptic presents a fatigue safety factor of 1.35 against the fatigue failure which shows that the infinite life cycle was reached.

Critical speed

For a rotating shaft it is important to know what is the critical speed. When a shaft is turning, eccentricity causes a centrifugal force deflection, which is resisted by the flexural rigidity of shaft, EI . As long as deflections are small, there are no damage. However, potentially may occur a problem called critical speed. At certain speeds the shaft is unstable, with deflections increasing without upper bound. Although the dynamic deflection shape is unknown, the use of a static deflection curve gives an excellent estimate of the lowest critical speed. Such a curve meets the boundary condition of the differential equation (zero moment and deflection at both bearings) and the shaft energy is not particularly sensitive to the exact shape of the deflection curve. When geometry is simple, as in a shaft of uniform diameter, simply supported, the task of calculate it is easy. The exact critical speed can be expressed as in the equation (3.29) [14; 17].

$$\omega_c = \pi^2 \cdot \sqrt{\frac{E \cdot I}{\rho \cdot A \cdot l^4}} \quad (3.29)$$

However, the critical speed can be approximated by the Rayleigh's energy method as shown in equation (3.30).

$$\omega_c = \sqrt{\frac{g \cdot \sum w_i \cdot \delta_i}{\sum w_i \cdot (\delta_i)^2}} \quad (3.30)$$

The model discretization was made as the Figure 3.16 shows.

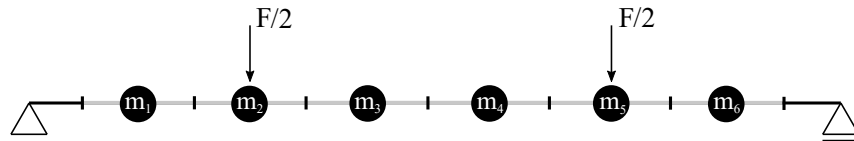


Figure 3.16: Shaft dynamic model discretization

The weight recognized of each parcel of the shaft is the half past and the half before of the node. The force is applied on the node 2 and 5 which is added to the weight. equation (3.31).

$$F = \begin{bmatrix} 0.757 \\ 20001 \\ 0.828 \\ 0.828 \\ 20001 \\ 0.757 \end{bmatrix} \quad [\text{N}] \quad (3.31)$$

Comparing the external applied forces with weight of the shaft, it is seen that the weight may be neglected, as expected.

Beginning to determine the static deflection, given by equation (3.32), at node i caused by a unit force applied at node j .

$$\delta_{ij} = \begin{cases} \frac{b_j \cdot x_i}{6 \cdot EI \cdot l} \cdot [l^2 - (b_j)^2 - (x_i)^2], & x_i \leq a_i \\ \frac{a_j \cdot (l - x_i)}{6 \cdot EI \cdot l} \cdot [2l \cdot x_i - (a_j)^2 - (x_i)^2], & x_i > a_i \end{cases} \quad (3.32)$$

With the computer assistance, MATLAB[®] software, the deflection is determined at each section caused by each force. Then, by effect superposition, multiplying the force applied by the deflection at each location, as the equation (3.33) exhibits, the total deflection at each section is reached.

$$y_i = \sum_{j=1}^n \sum_{i=1}^n F_j \cdot \delta_{ij} \quad (3.33)$$

The results of the deflection caused by a force of 40 kN, half for each rolling bearing, are stored in the matrix (3.34).

$$y = \begin{bmatrix} 34 \\ 61 \\ 76 \\ 76 \\ 61 \\ 34 \end{bmatrix} \quad [\mu\text{m}] \quad (3.34)$$

With some insight, quickly realize that these results are very close to those obtained earlier with more care and presented in the Section 3.2.1.

Now, the much-sought critical speed can be easily determined. Its value is shown in (3.35).

$$\omega_c = 3838 \text{ rpm} \quad (3.35)$$

This is a very important subject, therefore, it should be parameterized, as the Figure 3.17 shows, by the variable F for the range of 0 up to 40 kN. Since in most cases the force applied may be less than the maximum force, this study allows to know the maximum speed that the test rig can be used at each force condition.

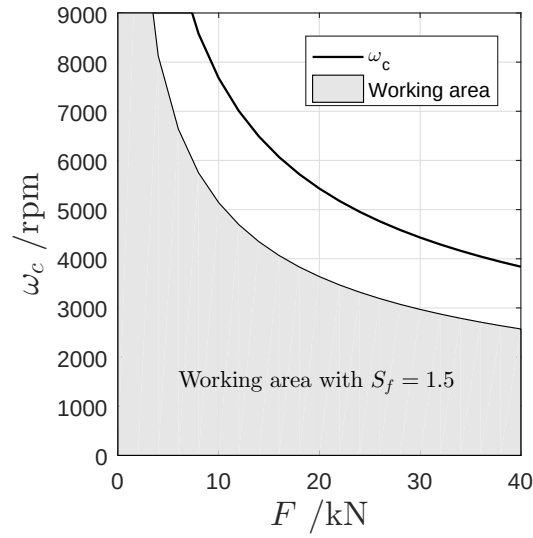


Figure 3.17: Critical speed and the working area.

As the ω_c is the critical speed, this is the speed that leads the system to resonance of displacement, it was defined a working area by applying a safety factor of 1.5, that must be respected in service.

Verification with KISSsoft® according to FKM standards

KISSsoft® software was also used to calculate the shaft in accordance with the FKM standards. This standards was choose because the FKM guideline for analytical strength assessment of mechanical components has gained an increasing interest for the last decade in the industry, once it describes a general procedure directly applicable in an industrial design office and since 2004 it is also available in English, which make its use more attractive [18].

The Figure 3.18 is the representation of the model studied on the KISSsoft®. The safety coefficients shown in bold the Tables 3.10 and 3.11 were chosen according to the failure consequences, present in the FKM standards. The calculation was made for infinite life cycle.

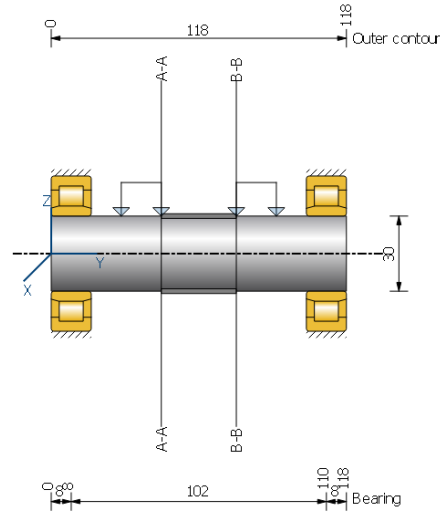


Figure 3.18: Drawing of the shaft in KISSsoft.

Table 3.10: FKM static safety coefficients for stress or stress combination in ductile materials [15].

		Probability of stress or stress combination							
		high				low			
		j_m	j_p	j_{mt}	j_{pt}	j_m	j_p	j_{mt}	j_{pt}
Failure consequences	severe	2.0	1.5	1.5	1.0	1.8	1.35	1.35	1.0
	mean	1.85	1.4	1.4	1.0	1.7	1.25	1.25	1.0
	moderate	1.75	1.3	1.3	1.0	1.6	1.2	1.2	1.0

Where the j_m is the static safety against fracture, j_p is the static safety against yield point, j_{mt} is the creep strength safety against fracture and j_{pt} is the creep strength safety against yielding [15].

Table 3.11: FKM endurance limit factor.

j_f		regular inspections	
		yes	no
Failure consequences	severe	1.35	1.5
	mean	1.25	1.4
	moderate	1.2	1.3

After drawing the shaft with an interference fit on KISSsoft[®] and applying the security coefficients of the FKM standards, the results obtained are presented in the Table 3.12.

Table 3.12: KISSsoft® results for the shaft design.

Max. deflection μm	Max. rotation mrad	σ_a^{eq} MPa	Static Safety	Fatigue Safety *	Fatigue Safety **	Fatigue Safety ***
125.12	2.88	211.24	2.81	1.24	1.35	1.56

* Interference fit hub case

** Slight interference fit hub case

*** Interference fit hub with end relief case

The safety factors are calculated for three types of interference fit because it could not be directly specified in the KISSsoft® in the shaft calculation. Therefore, the shaft was calculated for those three options available. The interference fit hub case that corresponds to the interference of H8/u8 according to FKM Guideline “Analytical Strength Assessment” [15], promotes the maximum tightness of 66 μm . The slight interference fit case, corresponds to a non demountable connection with slight cold tightening, probably H7/p6 without any reference to this in “Analytical Strength Assessment” and finally the interference fit hub with end relief case, which promoted the largest safety factor once this hub has a stress relief in the critical sections.

In comparison with the results obtained according to Shigley’s procedure [14], the case with the slight interference gives similar result. However, if we look at either case, they are on the safety side against the endurance limit, evidencing that the infinity life has been achieved. The detailed report of KISSsoft® calculation can be consulted in the Appendix E.1.

Alternative shaft

As previously explained, the shaft with the hub does not meet the misalignment requirements theoretically, but in practise it can be more approximate to the required value because the influence of the hub in the stiffness was not taken into account, neither the rolling bearing inner rings. So now will be presented an alternative, Figure 3.19, constructed with a shoulder, instead of a hub, that gives higher stiffness and therefore lower deflection and rotation.

This configuration is more expensive than the previous one because the raw diameter should be higher than 42 mm due to the shoulder, hence should be chosen when the rolling bearing life needs to be evaluated.

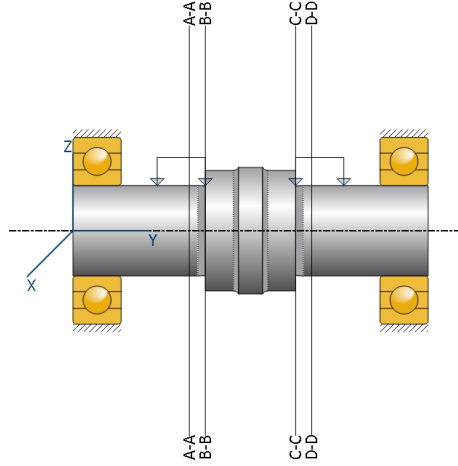


Figure 3.19: Alternative Shaft.

The dimensions of the shaft are the same of the shaft with hub, since the shoulder have the same function as the hub. The shoulder should has surface grooves for the extractor tool as in the shaft with hub, but in the KISSsoft[®] model it is difficult to consider, hence it was considered that the shoulder is composed by three circular sections. The basic dimensions of the real shaft are the same as those shown in the Figure 3.5.

The relief groove applied in the shoulder to reduce the stress concentration factor, was chosen according to the International Standard ISO 18388, being it the relief grove type E serie 1. This relief is presented in the Figure 3.20.

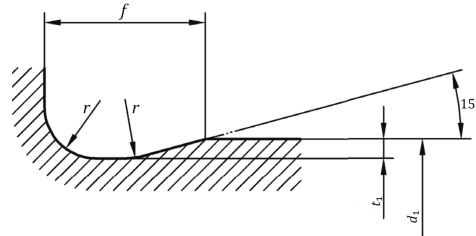


Figure 3.20: Relief groove type E for cylindrical surfaces.

Where d_1 is the workpiece diameter 30 mm, f is the width of relief groove 2.5 mm, t_1 is the deep of recess 0.2 and r is the radius of relief groove 1.2 mm.

The design of this shaft follows the same process of the previous one, therefore the process will not be shown again. The KISSsoft[®] results, calculated with 20 kN applied per rolling bearing and with a static load factor of 1.5, are shown in Table 3.13. The full report, obtained in KISSsoft[®] can be consulted in the appendix E.1.

Table 3.13: Results for the alternative shaft loaded with 20 kN per rolling bearing.

Max. deflection μm	Max. rotation mrad	σ_a^{eq} MPa	Static Safety	Fatigue Safety
74.854	1.78	211.24	3.00	1.67

3.2.2 Rod - Part 5

The load is transmitted to the rolling bearings by the rod that connects the hydraulic cylinder to the bearings sleeve. This rod could be all made of current steel, like Ck 45 K, however, the test rig needs to be small, therefore it is advisable to use a steel with better properties like 34 CrNiMo 6 to the parts with high load. An isometric view of the rod is shown in Figure 3.21.

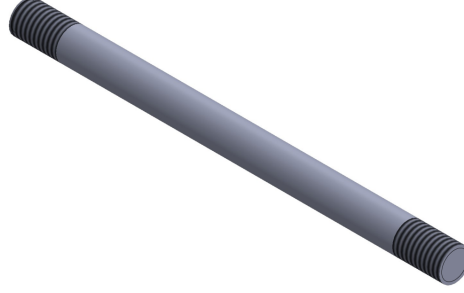


Figure 3.21: Isometric view of rod.

Rod diameter

Once a 40 kN traction force is applied on the rod, it can be designed like a bolt with $R_{pt} = 1000$ MPa. Therefore, the REApE standard for bolts is applied, which is presented in “Parafusos de Transmissão de Potência e de Ligação” [19]. The strength force is given by the equation (3.36). The shear force was neglected because it comes from the bearings loss torque which is quite low.

$$N_{rd} = N_p \cdot n_s \quad (3.36)$$

In this case, the number of “bolts” n_s is 1, hence, the resistant stress is equal to the stress in the rod. Then, it is important to verify if the core of the rod is in safe. According to REApE, the core of the bolt can be stressed up to 80% of the yield stress without damage, equation (3.37).

$$N_p = 0.8 \cdot R_{pt} \cdot \left(\frac{\pi \cdot (d_e)^2}{4} \right) \quad (3.37)$$

Solving the equations (3.36) and (3.37), the results are displayed in Table 3.14, and a rod diameter of at least 8 mm is needed.

Table 3.14: Results for the rod according REApE.

n_s	N_{rd}	N_p	d_e
	kN	kN	mm
1	40	40	≥ 8

Reducing the allowable stress to $0.6 \cdot R_{pt}$ in the core, which increases the safety factor a $S_f = \frac{1}{0.6} = 1.6$, the equation (3.37) is now represented by the equation (3.38).

$$N_p = 0.6 \cdot R_{pt} \cdot \left(\frac{\pi \cdot (d_e)^2}{4} \right) \quad (3.38)$$

Solving equation (3.38) for a $N_p=40$ kN in order to d_e , it results in a minimum diameter for the rod of $d_e=10$ mm. So, a 10 mm rod will be used for load application.

Thread analysis

The force applied by the cylinder on the rod is transmitted by the thread to the rod end. Therefore, it is important to verify the stress in thread.

For the chosen diameter, $d_e = 10$ mm, it is necessary to find the minimum number of thread helices, n , that support and transmit this load without damage. In Figure 3.28 are shown the load types that are applied in the thread.

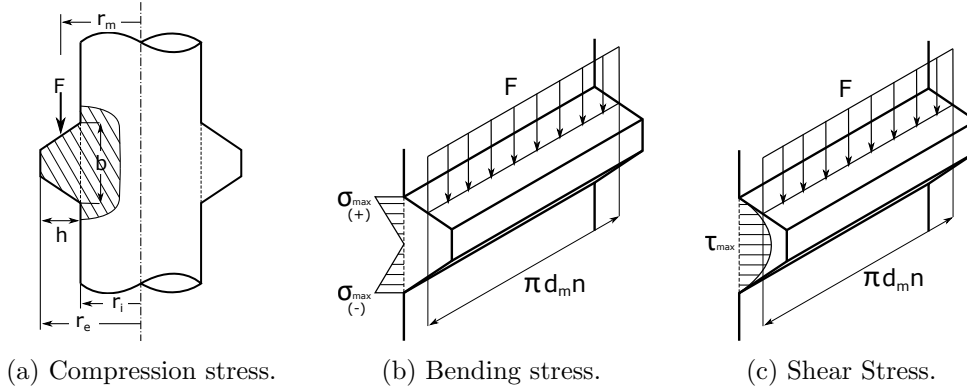


Figure 3.22: Types of load on the thread [19]

Compression Stress

The compression occurs in the face of the thread, as shown in Figure 3.22a, because the force applied tends to axially disconnect the parts and the thread have to support it. The yield stress verification is given by the equation (3.39).

$$\frac{R_{pt}}{S_f} = \frac{F}{\pi \cdot \left[\frac{(d_e)^2 - (d_i)^2}{4} \right] \cdot n} \quad (3.39)$$

Bending Stress

The force applied to the thread is a distributed load along the helice that can be simplified in a equivalent distributed load over one plan in the mean diameter, as represented in Figure 3.22b. This load causes a bending moment that generates bending stress. The static safety is verified by equation (3.40).

$$\frac{R_{pt}}{S_f} = \frac{F \cdot \left(\frac{h}{2}\right) \cdot \left(\frac{b}{2}\right)}{\pi \cdot d_i \cdot \left(\frac{n \cdot h^3}{12}\right)} \quad (3.40)$$

Shear Stress

The last criterion that needs to be checked is the shear stress. This occurs by shear force, that tends to "cut" the thread by the core of the rod, as shown in Figure 3.22c.

equation (3.41) represents the shear stress criterion.

$$\frac{R_{ps}}{S_f} = \frac{F \cdot 1.5}{\pi \cdot d_i \cdot n \cdot b} \quad (3.41)$$

For the three criteria a safety factor of $S_f = 2$ is used, because these expressions assume that the planned thread is a circle instead of a helix, which implies a deviation from the reality. The metric thread M10 dimensions are displayed in Table 3.15.

Table 3.15: Dimensions of the metric thread.

d_e	d_i	d_m	b	h	pitch
mm	mm	mm	mm	mm	mm
10.000	8.376	9.188	1.125	0.812	1.5

The application of the three criteria presented above, considering that the material is 34 CrNiMo 6, material properties are in the Appendix C, result in a number of thread pitches in contact, n , that are represented in Table 3.16.

Table 3.16: Minimum number of threads, n .

	Compression	Bending	Shear
n	3.4	5.9	7.0

The minimum thread length is at least $n \times \text{pitch} = 7 \times 1.5 = 10.5$ mm. According to the ISO standard guidelines, the thread length should be at least, for a M10 screw, $1.5 \times d_e = 15$ mm.

Note that for nitrided steel, this choice will be in the security side because for hard steel the thread length is only equal to d_e .

Connection with rod end

The connection between the rod, part number 5, and rod end, part number 7, is shown in Figure 3.23. This connection was chosen because it was not granted access to the welding process.

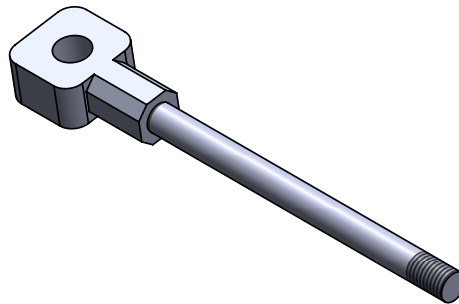


Figure 3.23: Connection between the rod and rod end.

After the design of rod core and the thread, the minimum external diameter of the rod end that supports 40 kN also need to be designed. The resistant area is the annular area between rod end external and internal diameter, as the Figure 3.24 shows.

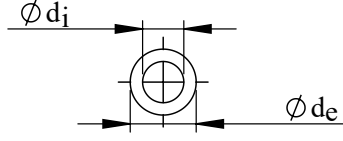


Figure 3.24: Rod end annular section.

Knowing that the d_i is imposed by the rod thread M10, and the material used in the rod end is 34 CrNiMo 6 due to having to withstand high loads while maintaining a small dimension, d_e is known by calculating the minimum external diameter that meets the 60% of the yield, equation (3.42). So the same safety factor of the rod is obtained.

$$d_e = \sqrt{\frac{F \cdot 4}{\pi \cdot 0.6 \cdot R_{pt}} + (d_i)^2} \quad (3.42)$$

The d_e minimum is 13.59, hence it was assumed that $d_e=16$ mm.

3.2.3 Knuckle Joint - Parts 7,8 and 10

This kind of connection, Figure 3.25, was chosen due to the advantages it brings to the test machine. The knuckle joint has the potential to allow horizontal displacement between the pieces as soon as the pin is withdrawn, which allows disassembly of the shaft and rolling bearings without dismantling the load system.

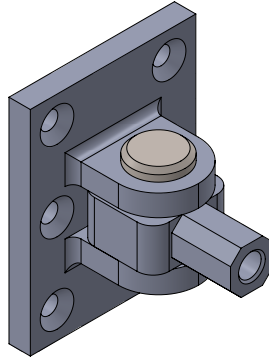


Figure 3.25: Knuckle joint isometric view.

Following the calculation process presented by Moring Faires in the book “Design of machine elements” [20] for the design of the knuckle joint, it was easy to reach a solution. However, KISSsoft® also does the same work, and so, it was possible to verify the solution.

A knuckle joint scheme is shown by the Figure 3.26.

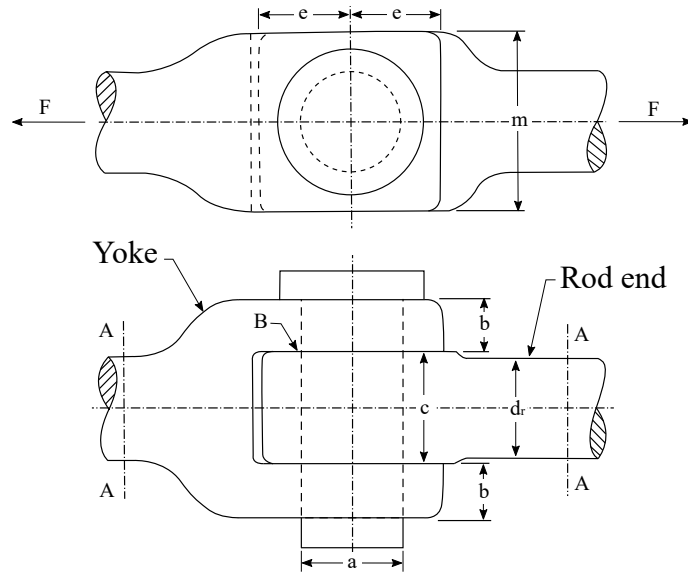


Figure 3.26: Knuckle joint scheme. [20]

The design must be done for the maximum load that can be applied in service, therefore, F value was assumed 40 kN. About the material, it was used a 34 CrNiMo 6 because it was made a previous design with Ck 45 K and the size reached was too big comparing with the dimensions of the carter.

A safety factor $S_f = 1.6$ was used. Dividing the yield strength of the material, presented in the Table C.1, by the safety factor, the tensile yield strength and the shear strength used for the design are given by equation (3.43) and (3.44), respectively.

$$R_{pt} = \frac{1000}{1.6} = 625 \text{ MPa} \quad (3.43)$$

$$R_{ps} = \frac{577}{1.6} = 360 \text{ MPa} \quad (3.44)$$

Diameter of the rod end (d_r)

In the rod end, the value of the diameter (d_r) can be determined in order to assure that the tensile yield strength is not achieved, as given by equation (3.45).

$$d_r \geq \left(\frac{F \cdot 4}{R_{pt} \cdot \pi} \right) \quad (3.45)$$

The minimum diameter of the circular section is 9 mm. The connection with the rod needs 16 mm. So the final value for the rod end diameter is $d_r=16$ mm.

In knuckle joints, the pin is not only subjected to shear stress, this is an idealization, but also to bending. Therefore, the distribution of load on the ears of the yoke is always distributed instead of punctual. There are some studies about this problem, the study made by Edward Kwon “Pin Failure in Shear vs. Bending for a Double Shear Joint” [21] and their assumptions give a good approximation to the real load application. The first approach proposed by Edward, considering the pin requested to bending and shear stress, assumed constant load, as the Figure 3.27a shows. The approximation with better results proposed by Edward assumes that the load is triangular, as shown in Figure 3.27b.

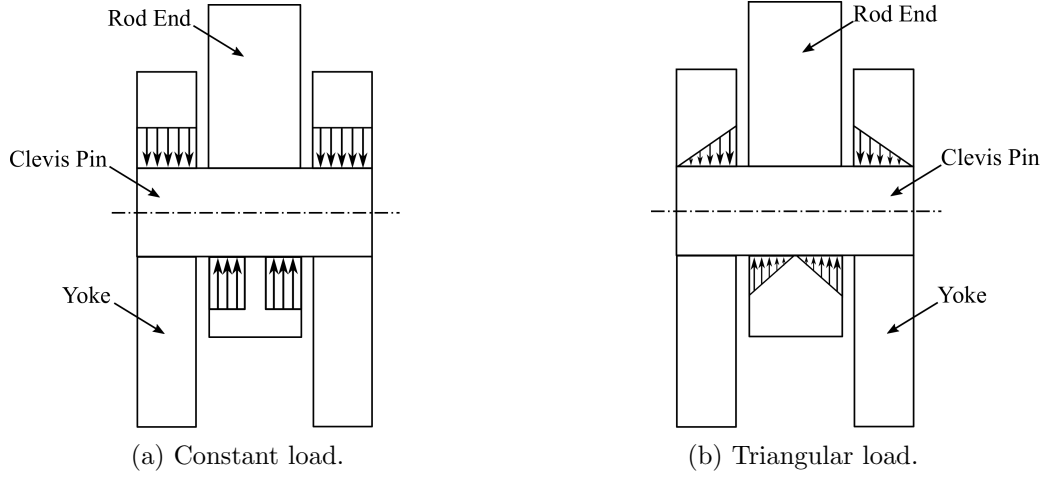


Figure 3.27: Types of load in knuckle joint[21].

According to Eduard Kwon, for a triangular load distribution and neglecting the gaps between the pin and the ears, the bending moment and the normal stress are given by (3.46) and (3.47), respectively.

$$M = \frac{F}{2} \cdot \left(\frac{b}{3} + \frac{c}{6} \right) \quad (3.46)$$

$$\sigma_b = \frac{8 \cdot F \cdot (2b + c)}{3\pi \cdot a^3} \quad (3.47)$$

The equation (3.48) is obtained by rearranging the equation (3.47) in order of the F variable.

$$F = \frac{\sigma_b \cdot (3\pi \cdot a^3)}{8 \cdot (2b + c)} \quad (3.48)$$

The equation (3.49) represents the compression force applied on the projected face of the pin.

$$F = \sigma_c \cdot a \cdot c \quad (3.49)$$

The optimal solution is that which makes the pin equally strong in compression and bending [20; 21]. Combining the equations (3.48) and (3.49) and the following safety assumptions were made:

- $c=2b$ – The yoke will be safe if the rod is safe.
- $\sigma_c=1.6 \cdot \sigma_b$ – Assuming that this is a fixed connection, since it has very small relative movement.

The relation between the a and c values is given by the equation (3.50).

$$a = \sqrt{\frac{\sigma_c}{\sigma_b} \cdot \frac{16}{3\pi}} \cdot c \quad (3.50)$$

Where, by replacing the values, comes $a = 1.644 \cdot c$

Height of the rod end (c)

Using the equation (3.49) with above relation, the c value is reached by the equation (3.51).

$$c \geq \sqrt{\frac{F}{1.644 \cdot R_{pt}}} \quad (3.51)$$

Where it results in $c \geq 6.97$ mm

Diameter of the pin (a)

Using the relation $a = 1.644 \cdot c$ that is the result from the equation (3.50) and assuming that $c = 7$ mm, the diameter of the pin should be $a \geq 12$ mm.

Yield strength verification on the pin under double shear

The equation (3.52) gives the minimum value for the a that verifies the safety of the pin in double shear against the yield stress.

$$a \geq \sqrt{\frac{2F}{R_{ps} \cdot \pi}} \quad (3.52)$$

The minimum diameter of the pin (a) value that verifies the security is 8.4 mm. Since it is less than that required for bending, means that the pin is safe in shear.

Width of the rod (m)

Theoretically, the m dimension can be known by calculating the minimum value that verifies the strength against the yield across the hole, given by the equation (3.53).

$$m \geq \frac{F}{R_{pt} \cdot c} + a \quad (3.53)$$

That makes $m \geq 33$ mm. However, the author suggests that to be safe against yielding due to local stresses, m should be at least $2.5 \cdot a$, because the dimension between the center of the hole and the end should be at least $1.25 \cdot d_{hole}$, which is 30 mm.

Semi-width of the rod (e)

$$e \geq \frac{F}{2R_{ps} \cdot c} \quad (3.54)$$

As previously explained, e can be calculated by the equation (3.54). Assuming that e is $\frac{m}{2}$, which is 15 mm.

All knuckle joint dimensions

As c is less than d_r , and from the point of view of using the capacity of the material, the optimal utilization occurs for $c \geq d_r$. It was assumed that $c = 16$ mm.

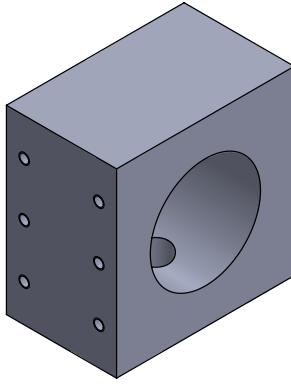
The knuckle connection dimensions, according to the values above obtained, are presented in the Table 3.17.

Table 3.17: Knuckle joint final dimensions.

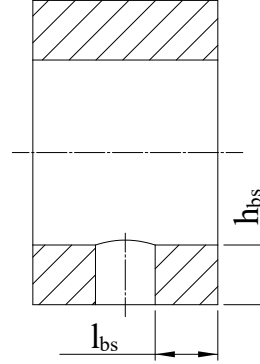
a	b	c	d_r	e	m
mm	mm	mm	mm	mm	mm
12	8	16	16	15	30

3.2.4 Rolling Bearings Sleeve - Part 11

The rolling bearings sleeve, shown in Figure 3.28a, is that applies the force of the cylinder on the rolling bearings, therefore, it is required by 40 kN force in horizontal direction. Hence, the critical section is the vertical section that passes through the center of the hole and it gives two rectangle areas, as the Figure 3.28b shows.



(a) Isometric view.



(b) Cross section.

Figure 3.28: Rolling bearings sleeve.

If it is assumed that the force is distributed half for the upper cut area and half for the two small lower cut areas, the most severe case is in the two small areas. The design of the sleeve should be done for the most severe case.

The candidate length for this sleeve is the sum of the length of two rolling bearings plus central hub. The size of the internal hole is fixed by the external diameter of the rolling bearings, that is 30 mm.

The stress in this two areas is given by the equation (3.55).

$$\begin{aligned}
 l_{bs} &= 21 \text{ mm} \\
 h_{bs} &= 20 \text{ mm} \\
 \sigma_{bs} &= \frac{F}{2 \cdot (l_{bs} \cdot h_{bs})} = 47.62 \text{ MPa}
 \end{aligned} \tag{3.55}$$

Which gives a safety factor against the yield, using the static verification as shown in equation (3.56), for Ck 45 K steel.

$$S_f = \frac{R_{pt}}{\sigma_{bs}} = 6.8 \tag{3.56}$$

3.2.5 Carter - Parts 1,2,3,6,9

As aforesaid the carter must meet some requirements, such as, be built in bolted connection i.e. the only manufacturing process involved is machining and be thick enough to accommodate the bearings. Therefore, the carter is constituted by four walls, parts 1,3,6 and 9, one base, part 2, and one top, which was not represented in the first draft. The volume of carter should be as small as possible because this is directly connected to the oil volume used in the test, a small oil volume also allows a higher concentration of wear particles per oil volume, which assures that each sample is more representative. The walls must be positioned with strict tolerances to ensure sealing.

An exploded view of the carter is shown in Figure 3.29.

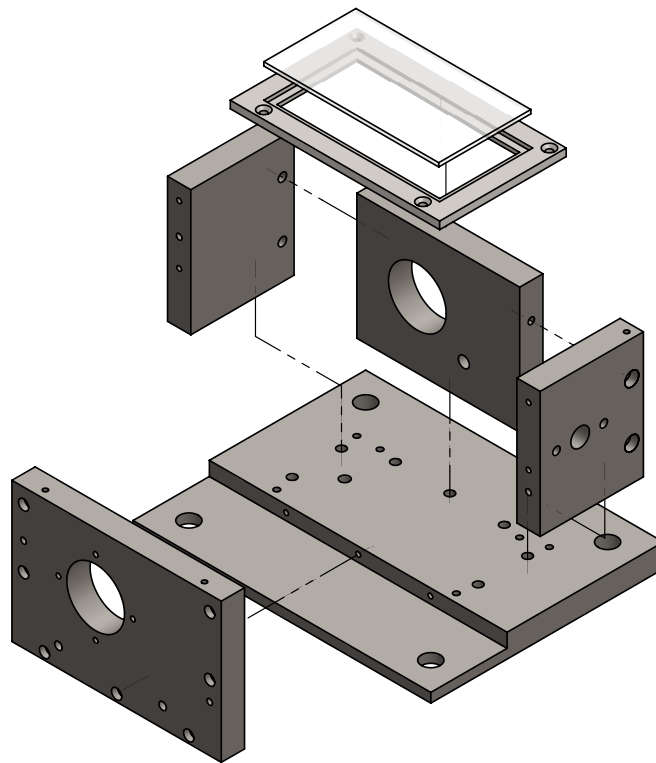


Figure 3.29: Exploded view of carter.

3.3 Oil bath temperature control

3.3.1 Cooling system

The cooling system is an important upgrade to this test rig because it allows us to make some tests with a stabilized target temperature of the oil bath.

For design this system, it is required to analyze the heat transfer of the oil, as a control volume, and then design a control system.

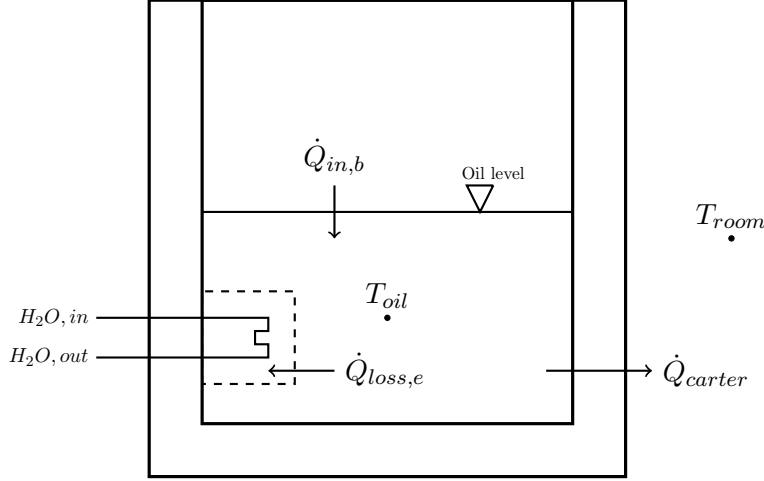


Figure 3.30: Carter heat transfer modes.

Figure 3.30 displays the carter (control volume) heat transfer modes, where $\dot{Q}_{in,b}$ is the heat gained due the rolling bearings, $\dot{Q}_{loss,e}$ is the heat loss due the water exchanger and $\dot{Q}_{carter}$ is the heat loss by the walls.

1st Law of thermodynamics - Oil

The first law is often formulated by stating that the change in the internal energy of a closed system is equal to the amount of heat supplied, minus the amount of work done by the system on its surroundings. This law can be applied, in order of the time, to the oil, which is presented in the equation (3.57), where the control volume is the oil. In this control volume, we do not have work because there is no load applied to the fluid.

$$\dot{Q}_{in,b} - \dot{Q}_{loss,e} - \dot{Q}_{carter} = \frac{dE_{int}}{dt} \quad (3.57)$$

The term $\frac{dE_{int}}{dt}$ represents the variation of the internal energy of the oil, which in this case is null because it is assumed permanent regime.

Power loss due the rolling bearings

Using the SKF model[11] to predict the power loss due the rolling bearings at the most severe test conditions, it is possible choose a reliable value for $\dot{Q}_{in,b}$. The simulated bearing conditions inputed to the SKF model are displayed in Table 3.3. With these conditions, the power loss dissipation for each of the selected bearing to test on the test rig was determined and displayed in Table 3.18.

Table 3.18: Power loss per rolling bearings at test conditions presented in Table 3.3.

Bearing designation	Type	C kN	Applied Load kN	Power loss W
6206*	ball bearing	20.3	10	170
7206 BEP	angular contact	22.5	10	230
1206 ETN9	self-aligning	15.6	7	24
NJ 206 ECJ*	cylindrical roller	44	20	61
30206 J2/Q	tapered roller	40.2	20	190

Assuming that the heat imputed to the oil bath, $\dot{Q}_{in,b}$, is equal to four times the power dissipated by the least efficient bearing, ie. $\dot{Q}_{in,b} = 230 \times 4 = 920$ W.

Model A. Heat loss by the walls, based on general correlations

Vertical walls

Using the “Thermal Resistance Concept” presented in “Fundamentals of Heat Mass Transfer”[22], the detailed heat loss mechanism can be represented as in Figure 3.31.

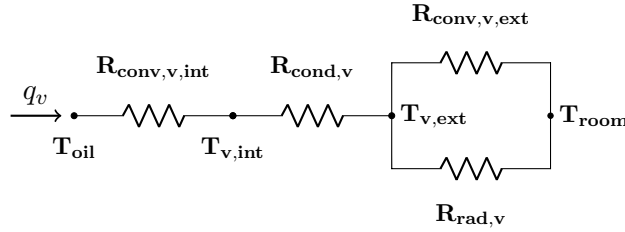


Figure 3.31: Vertical wall heat transfer.

Below, a detailed explanation of each mode of heat transfer is presented.

Using the Newton’s law of convection and applying it to the oil, as in equation (3.58), the heat loss by convection can be predict. For this propose, it is necessary to calculate the heat transfer coefficient for the vertical wall case, as will be done in EES.

$$\dot{Q}_{conv,v,int} = A_{ca,v} \cdot h_{conv,v,int} \cdot (T_{oil} - T_{v,int}) \quad (3.58)$$

For the analysis of the heat loss in the wall, it is used the Fourier’s law of conduction applied to the wall, as in equation (3.59). Knowing the material properties and its geometry, the heat loss by this mechanism can be predicted.

$$\dot{Q}_{cond,v} = \frac{A_{ca,v} \cdot k_{carter}}{t_{ca}} \cdot (T_{v,int} - T_{v,ext}) \quad (3.59)$$

Passing now to the contact on the outer side of the carter walls with the surrounding, again the mechanism of heat loss is the convection, which is evaluated by Newton’s law of convection in the air. The heat loss can be predicted by equation (3.60) with the corresponding heat transfer coefficient calculated for vertical wall.

$$\dot{Q}_{conv,v,ext} = A_{ca,v} \cdot h_{conv,v,ext} \cdot (T_{v,ext} - T_{room}) \quad (3.60)$$

Finally, the Stefan-Boltzmann's law of radiation is used for evaluate the radiation heat loss in vertical walls, as presented in equation (3.61).

$$\dot{Q}_{rad,v} = \varepsilon \cdot A_{ca,v} \cdot \sigma_{SB} \cdot (T_{v,ext}^4 - T_{room}^4) \quad (3.61)$$

As in the outer side of the carter the heat is lost by two different mechanisms, the sum of the two heats, equation (3.62), gives the heat lost to the surrounding.

$$\dot{Q}_{conv,v,ext} + \dot{Q}_{rad,v} = \dot{Q}_{conv-rad,v,ext} \quad (3.62)$$

Matching the heats along the transfer path, it is obtained the heat loss due the vertical walls $\dot{Q}_v$, as shown in equation (3.69).

$$\dot{Q}_{conv,v,int} = \dot{Q}_{cond,v} = \dot{Q}_{conv-rad,v,ext} = \dot{Q}_v \quad (3.63)$$

Base plate

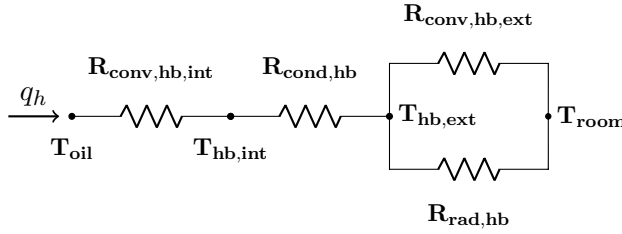


Figure 3.32: Horizontal plate heat transfer.

By the same reason above for vertical walls, the following equations (3.64)-(3.70) are used. The heat loss is predicted and added to the previous.

Newton's law of convection in the oil, presented in equation (3.64).

$$\dot{Q}_{conv,hb,int} = A_{ca,hb} \cdot h_{conv,hb,int} \cdot (T_{oil} - T_{hb,int}) \quad (3.64)$$

Fourier's law of conduction in the plate, presented in equation (3.65).

$$\dot{Q}_{cond,hb} = \frac{A_{ca,hb} \cdot k_{carter}}{t_{ca}} \cdot (T_{hb,int} - T_{hb,ext}) \quad (3.65)$$

Newton's law of convection in the oil, presented in equation (3.66).

$$\dot{Q}_{conv,hb,ext} = A_{ca,hb} \cdot h_{conv,hb,ext} \cdot (T_{hb,ext} - T_{room}) \quad (3.66)$$

Stefan-Boltzmann's law of radiation of the plate surface, presented in equation (3.67).

$$\dot{Q}_{rad,hb} = \varepsilon \cdot A_{ca,hb} \cdot \sigma_{SB} \cdot (T_{hb,ext}^4 - T_{room}^4) \quad (3.67)$$

Sum of the exterior heats, radiation and convection, is given by the equation (3.70).

$$\dot{Q}_{conv,hb,ext} + \dot{Q}_{rad} = \dot{Q}_{conv-rad,hb,ext} \quad (3.68)$$

Matching the heats along the transfer path, it is obtained the heat loss due the vertical walls $\dot{Q}_v$, as shown in equation (3.69).

$$\dot{Q}_{conv,hb,int} = \dot{Q}_{cond,hb} = \dot{Q}_{conv-rad,hb,ext} = \dot{Q}_{hb} \quad (3.69)$$

Total heat loss by the vertical walls and base plate

Finally, by the determination of the $\dot{Q}_v$ and $\dot{Q}_{hb}$, it is possible to know the total heat loss by the carter, by the sum of the heats.

$$\dot{Q}_v + \dot{Q}_{hb} = \dot{Q}_{carter} \quad (3.70)$$

The equations presented in this process were implemented in EES, which is a powerful software with thermodynamic properties, are shown in the appendix F. The results of the calculation are shown in Table 3.19.

Table 3.19: Heat losses calculated with Model A.

$\dot{Q}_{hb}$	$\dot{Q}_v$	$\dot{Q}_{carter}$
W	W	W
11.94	235.3	247.3

Model B. Heat loss by the walls, based on correlations for FZG carter

Höhn et al.[10] proposed a thermal model based on the equilibrium between the power loss and heat dissipation. This model was calibrated for the experimental values of oil temperature and power loss in a FZG test rig. As this rolling bearing test rig has similar carter dimensions, it is assumed that this model can be applied in this case with reliable results.

The equation (3.71) governs the thermal model:

$$\dot{Q}_{total,loss} = P_{VZP} + P_{VL} + P_{VD} \quad (3.71)$$

$$\dot{Q}_{total,loss} = \dot{Q}_{carter} + \dot{Q}_{loss,e} \quad (3.72)$$

Where $\dot{Q}_{carter}$ is the heat loss in carter walls, $\dot{Q}_{loss,e}$ is the heat loss due to the water exchanger, P_{VZP} is the power loss due to the gears, P_{VL} is the power loss due to the rolling bearings and P_{VD} is the power loss due to the seals. The test rig has no gears and the seal power loss is negligible, so the $P_{VZP} = P_{VD} = 0$. The power loss in the rolling bearings is converted into heat and $P_{VL} = \dot{Q}_{in,b}$.

Combining the equations (3.71) and (3.72):

$$\dot{Q}_{rad} + \dot{Q}_{conv} + \dot{Q}_{cond} + \dot{Q}_{loss,e} = \dot{Q}_{in,b} \quad (3.73)$$

Heat transfer coefficients correlations

The heat transfer coefficient for radiation is given by the equation (3.74):

$$h_{rad} = 0.23 \cdot 10^{-6} \cdot \varepsilon \cdot \left(\frac{T_{oil} + T_{room}}{2} \right)^3 \quad (3.74)$$

Considering forced convection on vertical walls, the heat transfer coefficient is given by the equation (3.75):

$$h_{conv,v} = 8.6 \cdot l_v^{-0.34} \cdot v_{air}^{0.64} \quad (3.75)$$

Where l_v is wall height, on which the boundary layer is developed and v_{air} is the air velocity.

If free convection on horizontal bottom plate is considered, the heat transfer coefficients is given by the equation (3.76), developed by *Funk* and presented in the paper[10].

$$h_{conv,hb} = 18 \cdot (l_h + w_h)^{-0.04} \cdot \left(\frac{T_{oil} - T_{room}}{T_{room}} \right)^{0.32} \quad (3.76)$$

Where l_h and w_h are the plate length and width, respectively. The boundary layer develops over the higher length.

Analogously and considering that the top plate are at mean temperature between the oil and the room, the heat transfer coefficients is given by the equation (3.77):

$$h_{conv,ht} = 18 \cdot (l_h + w_h)^{-0.04} \cdot \left(\frac{T_{mean} - T_{room}}{T_{room}} \right)^{0.32} \quad (3.77)$$

Heat Loss

Considering the radiation, the heat loss is calculated by the equation (3.78):

$$\dot{Q}_{rad} = h_{rad} \cdot A_{ca,rad} \cdot (T_{oil} - T_{room}) \quad (3.78)$$

With $A_{ca,rad}$ representing the carter surface area without considering the top plate because it loses a very reduced amount of the heat.

For forced convection, the heat loss is calculated with the equation (3.79):

$$\dot{Q}_{conv} = (h_{conv,v} \cdot A_{ca,v} + h_{conv,hb} \cdot A_{ca,hb}) \cdot (T_{oil} - T_{room}) + h_{conv,ht} \cdot A_{ca,ht} \cdot (T_{mean} - T_{room}) \quad (3.79)$$

Where $A_{ca,v}$, $A_{ca,hb}$ and $A_{ca,ht}$ are the vertical, bottom and top and surface area, respectively.

Now, it is needed to know the heat loss due the conduction and *Funk* proposed an approximation, represented by the equation (3.80), which is function of radiation and convection.

$$\dot{Q}_{cond} = (\dot{Q}_{rad} + \dot{Q}_{conv}) \cdot c_f \cdot \frac{A_{base}}{A_{ca}} \quad (3.80)$$

c_f represents a conduction factor, generally $c_f = 1.5$.

A_{base} represents the conduction area of the base.

Using the heat transfer coefficients and equation (3.73), the heat evacuation from the test rig is quantified.

Results

Table 3.20: Data for the heat loss calculation for Model B.

l_v m	l_h m	w_h m	v_{air} m·s ⁻¹	$A_{ca,ht}$ m ²	$A_{ca,hb}$ m ²	$A_{ca,v}$ m ²	A_{base} m ²	ε	T_{oil} K	T_{room} K
0.135	0.225	0.136	2	0.0306	0.0306	0.0975	0.0014	0.4	373.15	293.15

With the equations (3.74), (3.75), (3.76), (3.77) and the data presented in the Table 3.20 it is possible to calculate the heat transfer coefficients. The values are shown in the Table 3.21:

Table 3.21: Heat transfer coefficients calculated with Model B.

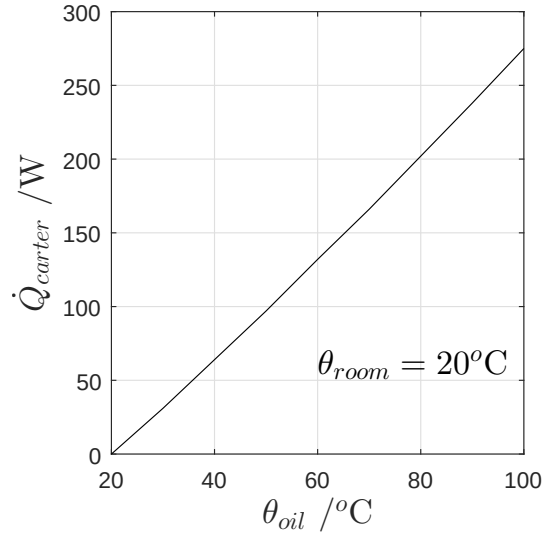
h_{rad} $W \cdot m^2 K^{-1}$	$h_{conv,v}$ $W \cdot m^2 K^{-1}$	$h_{conv,hb}$ $W \cdot m^2 K^{-1}$	$h_{conv,ht}$ $W \cdot m^2 K^{-1}$
3.40	26.46	8.85	7.09

Using the equations (3.78), (3.79), (3.80) and the data presented in the Table 3.20 it is possible to calculate heat losses due to different heat transfer modes . The values are shown in the Table 3.22.

Table 3.22: Heat losses calculated with Model B.

$\dot{Q}_{rad}$ W	$\dot{Q}_{conv}$ W	$\dot{Q}_{cond}$ W	$\dot{Q}_{carter}$ W
34.85	236.78	3.49	275

Varying the θ_{oil} in the service range, the heat loss can be predicted.


Figure 3.33: Heat loss by the carter for variable θ_{oil} .

Model C. Total heat loss by the water exchanger

The main objective of this thermal study is the water exchanger design, so this Model C was developed. The results obtained with Model A, based on correlations learned in Heat transfer curricular unit presented in “Fundamentals of Heat and Mass Transfer” [22], are close to the values obtained by Model B based on the correlations developed by Höhn *et al.* for the FZG carter, which support the results. So, it was considered the Model B proposed by Höhn *et al.* [10] to design.

Considering the equation (3.73) and Table 3.22, it is determined the heat loss for which the water exchanger should be dimensioned, as shown in equation (3.81).

$$\dot{Q}_{loss,e} = \dot{Q}_{in,b} - \dot{Q}_{carter} = 645 \text{ W} \quad (3.81)$$

Using the "Thermal Resistance Concept", the heat exchange modes between the water exchanger and the oil are represented in Figure (3.34).

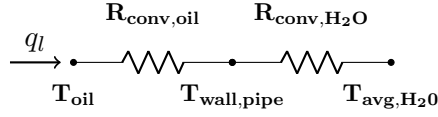


Figure 3.34: Heat transfer modes in water exchanger.

The conduction on the wall of the pipe can be neglected because the pipe material is copper, that has high thermal conductivity, and also has a thin wall.

Using the Newton's law applied in the oil mass, which is given by equation (3.82), it is quantified the exchanged heat due to the convection mode.

$$\dot{Q}_{loss,e} = \bar{h}_{conv,oil} \cdot A_{sur,pipe} \cdot (T_{oil} - T_{pipe,wall}) \quad (3.82)$$

Applying the Newton's law in the water mass, which is given by equation (3.83), it is quantified the exchanged heat due to the convection mode.

$$\dot{Q}_{loss,e} = h_{conv,H_2O} \cdot A_{sur,pipe} \cdot (T_{pipe,wall} - T_{avg,H_2O}) \quad (3.83)$$

Applying the 1st Law of thermodynamics to the water that flows inside the pipe, it matches the variation of the internal energy with the heat loss due the carter. It is shown in equation (3.84).

$$\dot{Q}_{loss,e} = \dot{m}_{H_2O} \cdot c_p \cdot (T_{out,H_2O} - T_{in,H_2O}) \quad (3.84)$$

With $T_{avg,H_2O} = \left(\frac{T_{in,H_2O} + T_{out,H_2O}}{2} \right)$.

Now, it is needed to choose a correlation to determine the convection coefficient of the oil, for this it could be used the book "Fundamentals of Heat and Mass Transfer"[22].

Since the oil has movement given by the shaft in rotation, it is necessary to know if it is enough to generate forced convection or if it is negligible. It could be evaluated by the ratio of the Grashof and Reynolds numbers, as shown in (3.85).

The Grashof number represents the ratio of the buoyancy to viscous forces acting on a fluid and generally it is used in natural convection. The Reynolds number represents the ratio of the inertial forces to viscous forces within a fluid which is subjected to relative internal movement due to different fluid velocities, known as boundary layer in the case of a bounding surface such as the interior of a pipe.

$$\frac{Gr}{Re^2} \quad (3.85)$$

If the ratio (3.85) is < 0.1 the natural convection is negligible and , by the other side, if the result is > 10 the forced convection is negligible.

The numbers can be calculated by solving the equations (3.86) and (3.87).

$$Gr = \frac{g \cdot \beta_{oil} \cdot (T_{oil} - T_{wall,pipe}) \cdot D^3}{\nu_{oil}^2} \quad (3.86)$$

$$Re = \frac{\rho_{oil} \cdot v_{oil} \cdot D}{\mu_{oil}} \quad (3.87)$$

Assuming $v_{oil} = 0.5 \text{ m} \cdot \text{s}^{-1}$, the results for Grashof and Reynolds numbers are shown in Table 3.23.

Table 3.23: Grashof and Reynolds numbers for the oil.

Gr	Re	$\frac{Gr}{Re^2}$
6.66E-6	56.71	2.07E-8

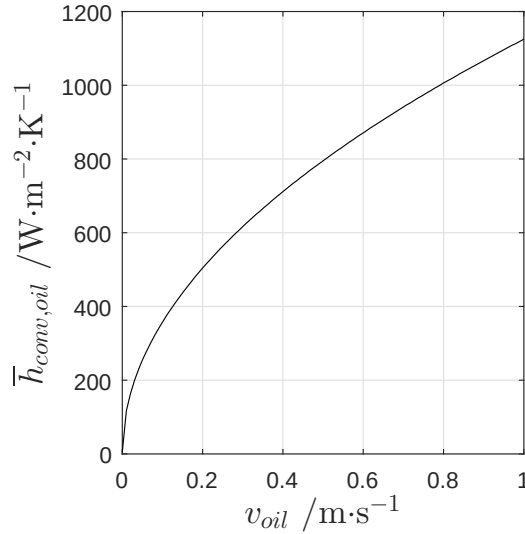
Thus, with results presented above, it is shown that natural convection is may disregarded compared to forced convection.

Correlation for forced convection around cylinders

The chosen correlation, shown in equation (3.88) was developed for external flow around cylinders with circular section and smooth surface.

$$\overline{Nu}_D = \frac{\bar{h}_{conv,oil} \cdot D}{k_{oil}} = 0.3 + \frac{0.62 \cdot Re_D^{1/2} \cdot Pr^{1/3}}{\left[1 + \left(\frac{0.4}{Pr}\right)^{2/3}\right]^{1/4}} \cdot \left[1 + \left(\frac{Re_D}{282000}\right)^{5/8}\right]^{4/5} \quad (3.88)$$

With $Re_D \cdot Pr \geq 0.2$ and fluid properties at film temperature $T_f = \frac{T_{oil} + T_{wall,pipe}}{2}$


Figure 3.35: Forced convection coefficient for variable v_{oil} .

The heat transfer coefficient was calculated for $\theta_{oil} = 100$ °C, $\theta_{in,H_2O} = 20$ °C and $v_{H_2O,max} = 2.5 \text{ m} \cdot \text{s}^{-1}$, which are the temperature and velocity of the piped water available on CETRIB Lab. The pipe diameter chosen is 8 mm.

Applying the equations (3.82),(3.83), (3.84) and (3.88), the results presented in Table 3.24 was obtained after some iterations.

Table 3.24: Heat transfer in water exchanger.

θ_{avg,H_2O} °C	$\theta_{wall,pipe}$ °C	$h_{conv,oil}$ W·m ⁻² ·K ⁻¹	$h_{conv,water}$ W·m ⁻² ·K ⁻¹	L_{pipe} mm	$Q_{loss,e}$ W
20.6	28.4	795	7399	450	653

3.3.2 Heating System

The heating system will be used when it is desired that the test temperature be higher than the stabilization temperature. By the Figure 3.33, which gives an approximation to the heat loss by the carter and assuming that it is needed to heat the oil from 20°C to 100°C and stabilize it, the following model can be implemented in the EES software. The fluid properties are used at θ_{oil} temperature.

Applying the 1st Law of Thermodynamics to the oil, as shown in equations (3.89) and (3.90).

$$\frac{dE_{oil}}{dt} = m \cdot cp \cdot \frac{dT}{dt} \quad (3.89)$$

$$\frac{dE_{oil}}{dt} = \dot{Q}_{resistor} - \dot{Q}_{carter} \quad (3.90)$$

Being, $\frac{dE_{oil}}{dt}$ the variation of the internal energy of the oil with the time, $\dot{Q}_{resistor}$ the heat that the resistor gives and $\dot{Q}_{carter}$ the heat loss by the carter.

Using the instruction presented in equation (3.91) in EES to integrate the expression (3.89):

$$\theta_{oil} = \theta_{room} + \int_0^t \frac{\dot{Q}_{oil}}{m \cdot cp} dt \quad (3.91)$$

Where the cp and ρ are function to the θ_{oil} . Therefore, as m varies with the ρ , the m also varies with the temperature.

To choose a nominal value for the heating resistor, some different $\dot{Q}_{resistor}$ values was tested, as is shown in Figure 3.36, hence it can see the behavior of the heating of the oil.

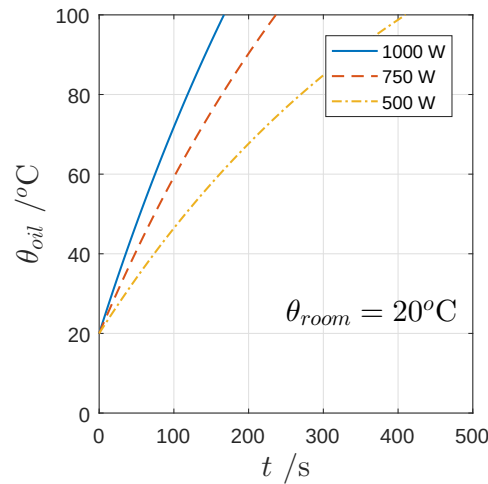


Figure 3.36: The oil heating time with variable $\dot{Q}_{resistor}$.

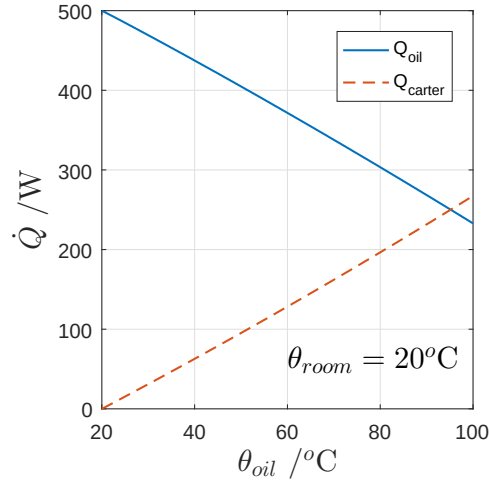


Figure 3.37: The behavior of $\dot{Q}_{oil}$ and $\dot{Q}_{carter}$ with the θ_{oil} .

Assuming that the 500W resistor is the most suitable for this problem, the detailed behavior of different heats for each temperature was plotted in Figure 3.37.

The cartridge heating resistor chosen is the one that produces 500 W of power.

3.3.3 Closed loop control system

The main purpose of this heating system is to set a required temperature, which the system goes there and stabilizes, with the lowest error possible.

The closed loop control is the right type of the control for this system because it allows to set a temperature with feedback given by thermocouple. The closed loop control is shown in Figure 3.38.

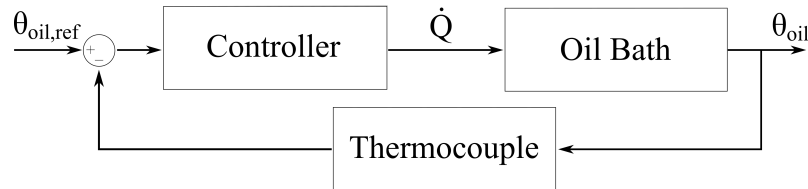


Figure 3.38: Heating closed loop control system.

The control system is closed by the thermocouple that reads the instant temperature and leads it to the comparator, which compares the instant value with the set value, then the controller acts in accordance with chosen procedure. The PID action control with auto-tuning mode, that is very effective for temperature control. The auto-tuning control behavior is presented ahead in section 3.5.

3.4 Standard Components

3.4.1 Pins for the front wall carter

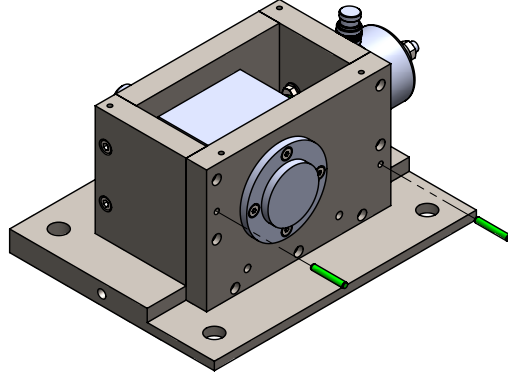


Figure 3.39: Mounting of the pins in the front wall carter.

Verification of the shear strength in the pin, presented in Figure 3.39 with 6mm of diameter:

$$\frac{R_{ps}}{S_f} = \frac{F}{\frac{3}{4} \cdot \frac{\pi \cdot (d_{pin})^2}{4}} \quad (3.92)$$

With the S_f equals to 2 and $R_{ps}=370$ MPa, according to von Mises criterion for the steel class 8.8, the shear force F that can be supported by each pin is 3915 N.

3.4.2 Bolts

ISO 4762 M6 for the carter front wall

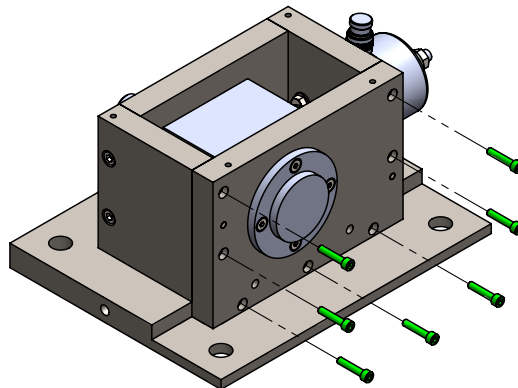


Figure 3.40: Mounting of the M6 bolts in the front wall of the carter.

The bolted connection, presented in Figure 3.40, is designed with base on the book of the course unit “Machine Elements II” for pre-stressed connections [19]. In this case, we

must avoid the shear in the bolts, and for this we need to pre-stress the bolts to prevent sliding of the walls. It must then be verified, in the case of slippage, what pre-stress is required on each bolt.

$$\tau = \mu \cdot N_p \cdot n_p \cdot n_s \quad (3.93)$$

Where τ is the shear force, μ is the coefficient of friction, N_p is the pre-stress on each bolt, n_p is the number of bolts and n_s is the number of the slippage plans.

Assuming that cylinder force is half supported by the front wall and half supported by the back wall. The joints between the front wall and side walls are requested with 20 kN, thus the two guiding pins can support $2 \cdot 3915N = 7830N$. For the equation (3.93), with $\tau = 20000 - 7830 = 12170$ N, $\mu = 0.2$, $n_p = 7$ bolts and 1 slippage plan, the result is shown in equation (3.94).

$$N_p = 8693 \text{ N} \quad (3.94)$$

On the other side, the pre-stress on the bolt is limited by the tension in the bolt core. According to the REApE, that assumes that the core of the bolt can be stressed up to 80% of the yield stress without damage:

$$N_p = 0.8 \cdot R_{pt} \cdot \left[\frac{\pi \cdot (d_{bolt})^2}{4} \right] \quad (3.95)$$

Chosen a 8.8 class, where $R_m = 800$ MPa and $R_{pt} = 640$ MPa. It results in $d_{bolt} \geq 4.65$ mm. Hence, the minimum standard d_{bolt} that verifies the strength is the 6 mm.

ISO 4762 M8 for the carter side walls

This side wall bolts, presented in Figure 3.41 are requested to the compression in service, hence, any bolt diameter meets resistance to yield. However, here the problem is related to the oil tightness, where it has to have enough tightness torque for this. The strength verification is related with the tightness yield.

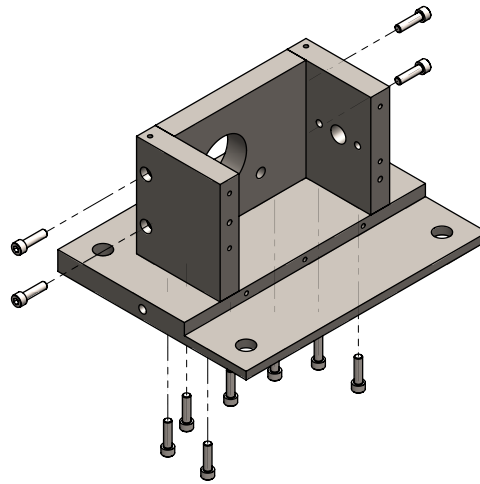


Figure 3.41: Mounting of the M8 bolts in the side walls.

ISO 7046 M6 for the yoke connection with bearing sleeve

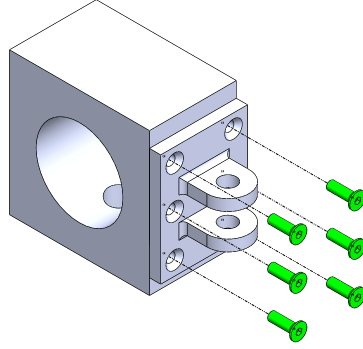


Figure 3.42: Mounting of the ISO 7046 M6 for the yoke connection with bearing sleeve.

The bolted connection, presented in Figure 3.42, is designed based on REApE standard for bolts, which is presented in “Parafusos de Transmissão de Potência e de Ligação” [19]. The strength force is given by the equation (3.96).

$$N_{rd} = N_p \cdot n_s \quad (3.96)$$

In the most severe case, N_{rd} is 40 kN and the number of “bolts” n_s is 6 to prevent the bending of the connection. Then, it is important to verify if the core of each bolt is in safe. According to REApE, the core of the bolt can be stressed up to 80% of the yield stress without damage, equation (3.97).

$$N_p = 0.8 \cdot R_{pt} \cdot \left(\frac{\pi \cdot (d_{bolt})^2}{4} \right) \quad (3.97)$$

Chosen a 8.8 class, where $R_m=800$ MPa and $R_{pt}=640$ MPa and solving the equations (3.96) and (3.97), the results are displayed in Table 3.25.

Table 3.25: Results for the rod according REApE.

n_s	N_{rd} kN	N_p kN	d_{bolt} mm
6	40	6.7	≥ 4.1

It results in $d_{bolt} \geq 4.1$ mm. Hence, the minimum standard d_{bolt} that verifies the strength is the 6 mm.

Bolt 5/16” 18UNC for the cylinder

The support bolts required by the cylinder to support it, presented in Figure 3.43, are according to the Unified Thread Standard. This thread profile is the same as that of ISO metric but the nominal diameter and the pitch are different. Since the bolts are in compression, there is no special concerns about the class, the most common class is acceptable.

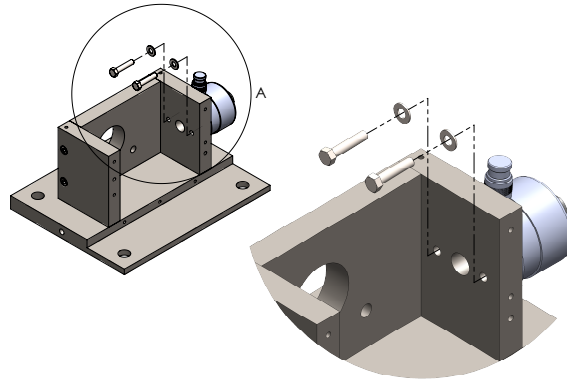
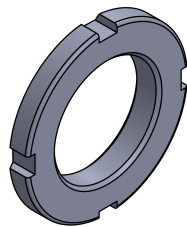


Figure 3.43: Mounting of the 5/16" bolts in the side wall.

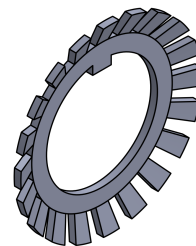
3.4.3 Nuts

SKF KM6 Lock nut and washer MB6 for rolling bearings positioning

Lock nuts are used to locate rolling bearings and other components onto a shaft. They have to be secured to prevent unintentional loosening. This is done, either by a locking device that engages a keyway in the shaft, or by a locking mechanism integrated in the nut. The lock washers are simple, stable and reliable fastening elements which engage a keyway in the shaft. The washer locks the nut in position when one of the washer tabs (tangents) is bent into one of the slots on the nut.



(a) KM6 Lock nut



(b) MB6 Washer

Figure 3.44: Axial locking system

ISO 4033 M10 nut for the end of the application load shaft

The hexagonal nut ISO 4033 grade A was chosen for the diameter of the shaft. It transmits the effort from the cylinder to the shaft.

3.4.4 Seals

Simrit Simmerring BAU

Simmerring BAU is a seal manufactured by Simrit which is made for rotative shafts. This seal can be used with synthetic, mineral oils and greases under 100°C. It also is suitable for circumferential speed below 10 m·s⁻¹.

The Simmerring seal is mounted in the cover, as shown in Figure 3.45.

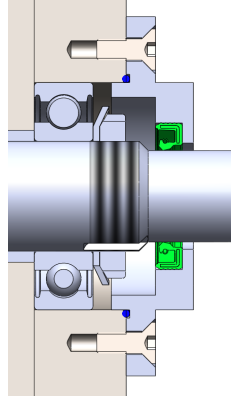


Figure 3.45: Mounting of Simmering BAU seal.

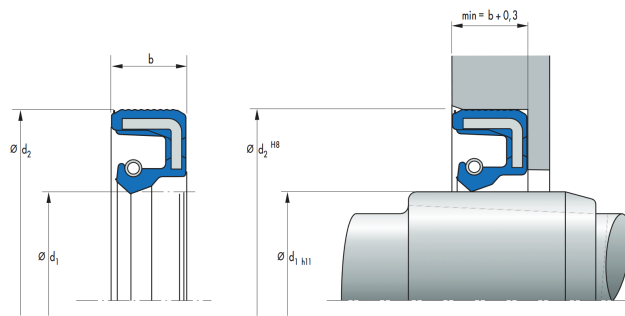


Figure 3.46: Simmerring BAU dimensions

The chosen dimensions presented in Figure 3.46 are shown in Table 3.26:

Table 3.26: Simmerring BAU dimensions

d_1	d_2	b	outer surface
mm	mm	mm	
25	40	7	smooth

Simrit Merkel U-Ring T20

The Merkel U-ring T20 is a seal that has asymmetrical profile, inner lip set back and press fit at the outside diameter. Hence, it present a very good static and dynamic sealing. Its mounting is shown in Figure 3.47, the seal is placed in a counterbore of the wall, to promote the sealing in the rod. This seal is usually used in standard cylinders.

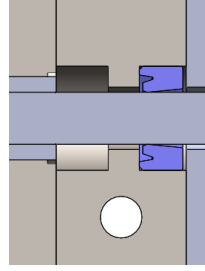


Figure 3.47: Mounting of Merkel T20 seal.

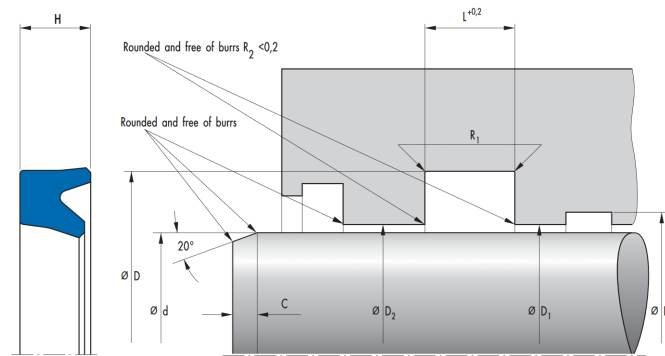


Figure 3.48: Merkel T20 dimensions.

The chosen dimensions presented in Figure 3.48 are shown in Table 3.27:

Table 3.27: Merkel T20 dimensions.

d	D	H	L	R_1
mm	mm	mm	mm	mm
10	20	8.2	9	0.4

Simrit Static Sealing O-Ring

The sealing effect of the O-Ring is attributable to its axial or radial change of shape of its cross-section once installed. This change of shape is achieved by a corresponding designing of the installation space. The resultant reactive force provides the seal with the necessary contact pressure, which is additionally supported by pressure from the medium. For flange and cover seals the cross-section of the ISC O-Ring is axially shaped because the flange presents a rectangular groove as shown in Figure 3.49.

In this test rig, the O-ring seals are used in the two covers of the shaft.

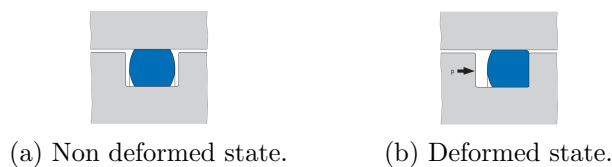


Figure 3.49: O-Ring states.

3.5 Catalogued Parts

3.5.1 Shinko ACS 13A Controller

A controller is the base of any automated system, therefore, to configure and control a temperature with a cooling and heating system, a controller is needed.

The Shinko ACS 13A S/M, presented in Figure 3.50, is already available in CETRIB, so it will be used for the control. This controller has 2 outputs, one of them for heating and another for cooling, which corresponds to the requisites. It also has a connection to PC which is a desirable.



Figure 3.50: Shinko controller ACS-13A.

Returning to the subject discussed in the control for the heating system, the auto-tuning mode in this controller has the following behavior.

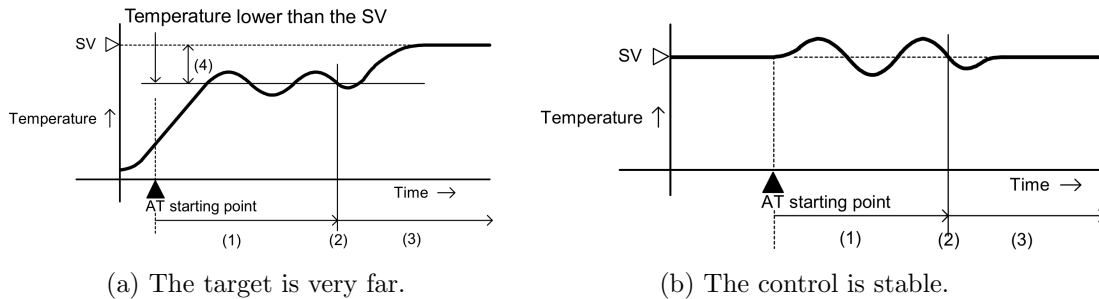


Figure 3.51: Auto-tuning behavior.

Looking at Figure 3.51 and considering the AT the starting point of the auto-tuning, the SV is the set or target value, the AT-bias the value below of the target where the controller will fluctuate around.

The auto tuning procedure, shown in Figures 3.51a and 3.51b, is described in Table 3.28:

Table 3.28: Controller AT procedure.

Step	Procedure
(1)	Begins to calculate the PID constant
(2)	The PID constant is reached
(3)	Control is done by the PID constant set by AT
(4)	AT bias value set by the user

3.5.2 ASCO 2 Way Selenoid Valve

The ASCO selenoid controlled 2 way valve, shown in Figure 3.52, allows to control the water flow in the cooling system. Since the constant time of the cooling is high, which means that the system reacts very slow, therefore an On/Off valve is enough.



Figure 3.52: 2 Way Selenoid Valve [23].

3.5.3 Omega Type K Thermocouple

A thermocouple is a simple, robust and cost-effective temperature sensor used in a wide range of temperature measurement processes. It consists of two dissimilar metal wires, joined at one end. Known for their versatility as temperature sensors, thermocouples are manufactured in a variety of styles, such as thermocouple probes which is used in this project, Figure 3.53.



Figure 3.53: Type K thermocouple probe [24].

3.5.4 Cartridge Type Heating Resistor

Cartridge heaters are most frequently used for heating metal parts by insertion into drilled holes, Figure 3.54. For easy installation, the heaters are made slightly undersize

relative to their nominal diameter. Usually this type of heater may be made with custom dimensions. The cartridge required for this project is the one that has 500 W and the appropriate dimensions in relation with the carter wall dimension.



Figure 3.54: Cartridge heater [25].

3.5.5 ETH DRDL II Torque Transducer

Torque transducer is the component that allows to measure the torque loss in the test rig. It is placed between the motor and the test rig. The torque transducer chosen is the ETH DRDL II that is compact and which is already available in CETRIB. The Figure 3.55 represents the torque transducer.



Figure 3.55: ETH DRDL II Torque Transducer.

3.5.6 Enerpac RCH 120 Hydraulic Cylinder

The cylinder is a key component in this test rig, it should provides a very high force. Therefore, a hydraulic cylinder is chosen, presented in Figure 3.56, because it allows high load capacity and at the same time has small dimensions as can be seen in Table 3.29. A hollow type cylinder allows to apply traction load, which is fundamental for avoiding deviations of the load application direction.



Figure 3.56: Enerpac cylinder RCH-120. [26]

Table 3.29: Principal information of cylinder.

Force	Stroke	Outside Diameter	Hole Diameter	Weight
kN	mm	mm	mm	kg
125	8	69	19.6	1.5

The force applied by the cylinder is proportional to the pressure applied to the oil times the area of the its rod section. To apply pressure to the hydraulic circuit it can be used a hand or foot pump with a press gauge. However, this method is not very accurate so that it is needed to use a load cell to give feedback of actual load applied.

3.5.7 HBM KMR 60 kN Load Cell

Load cell is the component that measures the actual force transmitted by the cylinder to the shaft, giving thus feedback. It should be positioned between the cylinder face and the nut. Hence it should be a annular cell, as presented in Figure 3.57 in order to be passed through the shaft. So that the cell works in compression, which is one of the most accurate types. The load cell dimensions are shown in Table 3.30.

Table 3.30: KMR Load Cell dimensions.

Nominal Force	Outside Diameter	Hole Diameter	Thickness
kN	mm	mm	mm
60	20	10	8



Figure 3.57: KMR load cell.[27]

3.6 Parts production drawings

The test rig assembly drawing as well as all the production drawings for all the designed parts are displayed in Appendix A and B, respectively. Table 3.31 displays the Appendix location for each drawing.

Table 3.31: Assembly and parts production drawings location

Drawing	Appendix
Test rig assembly	A.1
Base	B.1
Front wall	B.2
Right side wall	B.3
Left side wall	B.4
Back wall	B.5
Top frame	B.6
Main shaft	B.7
Hub	B.8
Bearing sleeve	B.9
Rod end	B.10
Yoke	B.11
Rod	B.12
Spacer	B.13
Cylinder spacer	B.14
Front cover	B.15
Back Cover	B.16

Test rig assembly

4.1 Assembly of the carter

The first step of the assembly is the mounting of the back wall, as the Figure 4.1 show. First, the pins are mounted and then the bolts are placed.

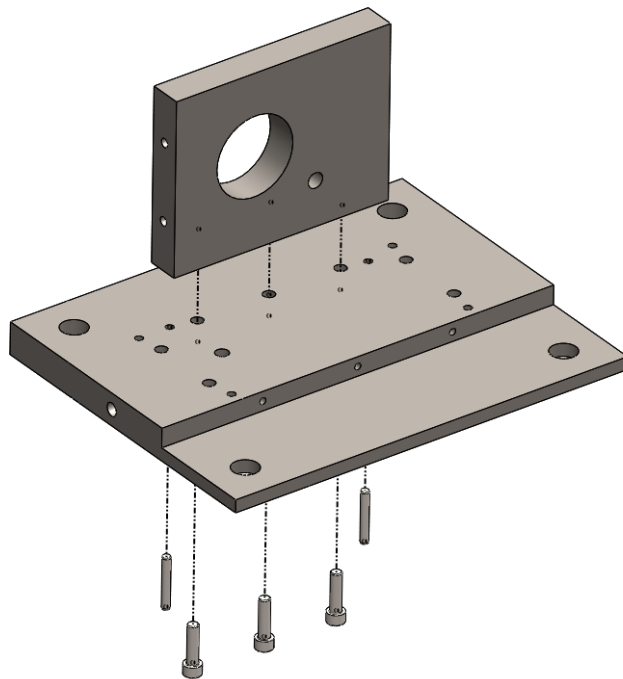
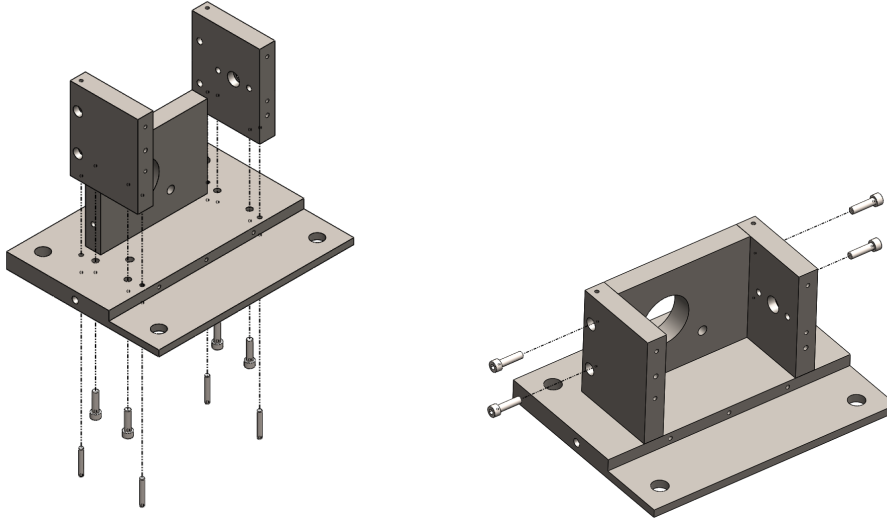


Figure 4.1: Mounting of the back wall to the base plate.

Then, the side walls should be mounted. For this, the side walls are placed leaning against the back wall and the base plate and the pins and bolts are placed, as shown in Figure 4.2a. Then, the connection of the back wall and the side walls should be performed by the bolts, as shown in Figure 4.2b.



(a) Connection of the side walls and base. (b) Connection of the side and back wall.

Figure 4.2: Side walls mounting.

4.2 Mounting of the rolling bearings in the carter

The mounting procedure is presented in detail below. This procedure had as premise the easy and quick assembly. As previously shown, the first draft, presented in Figure 3.2, had already been made based on this assembly routine. Therefore, the parts have been designed to meet this important requirement.

The mounting process begins with the axial positioning of the rolling bearings on the main shaft, these are mounted with slight interference fit, two on each side separated by a spacer ring, as shown in Figure 4.3. The locknuts and the washers axially lock the mounting and if needed they could give the pre-load to rolling bearings.

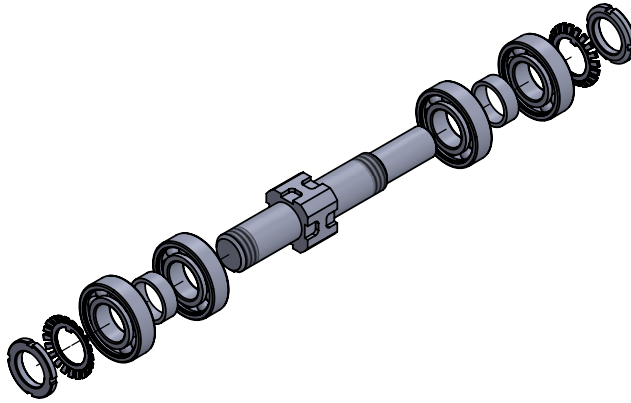


Figure 4.3: Mounting the rolling bearings on the main shaft.

After that, the shaft with rolling bearings is placed in the front wall, which is the moving wall, as represented in Figure 4.4a. The coverslips are placed in the front wall with the main role of prevent the axial movement of one of the outer rolling bearing and consequently the axial movement of the shaft, as shown in Figure 4.4b. The other outer bearing is leaved free, thus allowing the movement generated by the thermal expansion.

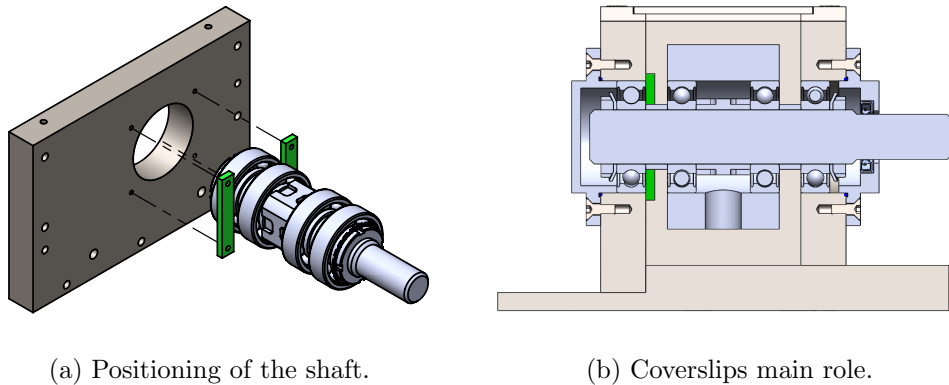


Figure 4.4: Mounting the main shaft with bearings on the front wall.

Then, the bearing sleeve and the yoke, represented in Figure 4.5, are mounted over the rolling bearings with sliding fit. The yoke is placed once in the sleeve and after that no longer needs to be disassembled.

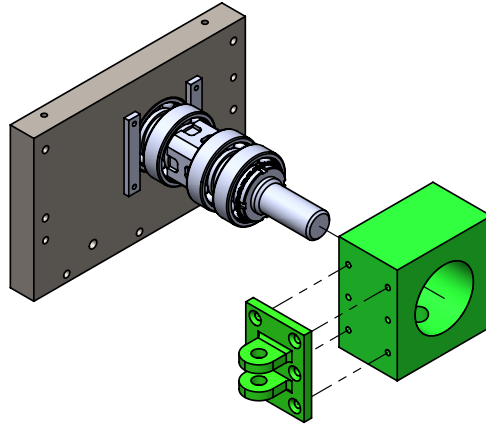


Figure 4.5: Mounting the bearing sleeve on the main shaft.

After that, the front wall is positioned on the side walls by the guiding pins. Being the rolling bearings easily placed on two walls because they are mounted with sliding fit. Finally, the covers are mounted on the front and back walls and the clevis pin is placed in the knuckle joint concluding thus the mounting process.

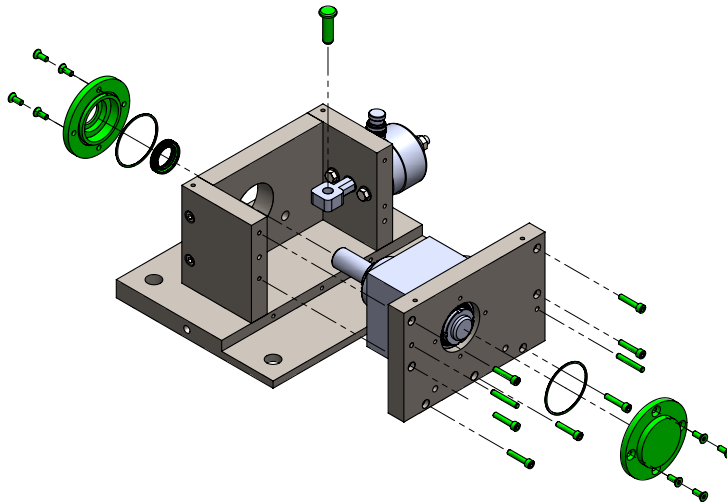
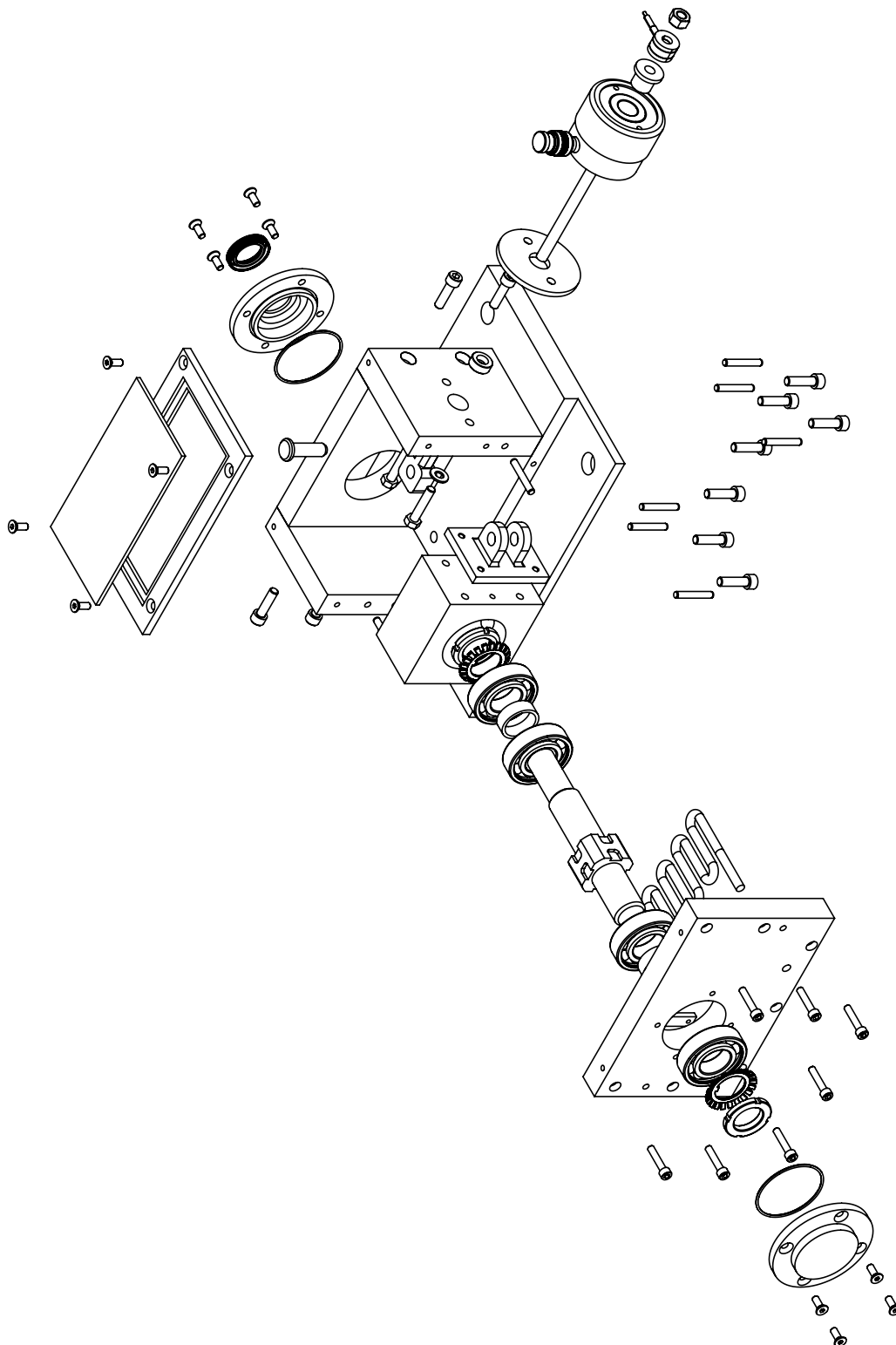


Figure 4.6: Front wall mounting and clevis pin placement.

The rolling bearings are mounted and positioned in the main shaft, the bearing sleeve which applies the radial force in bearings is placed and connected by the knuckle joint to the load system.

The simplicity of this assembly is based on the load system which was arranged so that it does not need to be mounted on all test preparations. The knuckle joint clearly simplified this process because with the removal of the pin, the load system is totally disconnected from the main shaft and the dismounting is quick.

4.3 Exploded view



Conclusions and Future Works

5.1 Conclusion

The main objectives of this dissertation have been met.

The test rig developed is capable of:

- Test different rolling bearing types;
- Impose the same radial load both on the test and the slave rolling bearings;
- Submit the test and the slave rolling bearings to the same operating conditions in terms of load, speed, temperature and lubricant feed;
- Control the oil sump temperature between 20 and 100°C;
- Impose a large spectrum of radial loads, from 0 up to 40 kN;
- Impose a large spectrum of speeds, from 0 up to 3000 rpm;
- Impose a large spectrum internal torque loss, from 0 up to 5 Nm.

5.2 Future Works

- Define the machining sequence and build the test rig;
- Perform tests to calibrate the machine;
- Study the value of misalignment in the bearings and their influence;
- Control system for the load application system;
- Extractor bearings tool customized to reach the inner raceway of the rolling bearings;
- Make a gabari with adjustment to the hub that serves as support for the mounting and allows tightening the locknuts.

5.3 Work final comments

The accomplishment of the present dissertation marks my conclusion of the course of Mechanical Engineering. In this work I had a great approximation to the project realized in the industries, where a mechanical equipment was designed from its root to the part mechanical drawings. In the course of the work, powerful design programs (KISSsoft), computation (MATLAB and EES) and 3D drawing (SolidWorks) were used. About the standardization used, some ISO and DIN standards were followed, with particular emphasis on the design process used under FKM standardization, which is a great tool for my future professional life.

In short, the main and optional objectives were achieved, so I feel very proud with this successfully work. I would like to see my test rig in operation performing daily tests. It was with great pleasure that I worked in these last five months in this Institution, CETRIB-INEGI, in order to help with one more test rig. I would like to show, one more time, my gratitude to the whole for their support, for the many knowledge and the resources that have been provided to me.

References

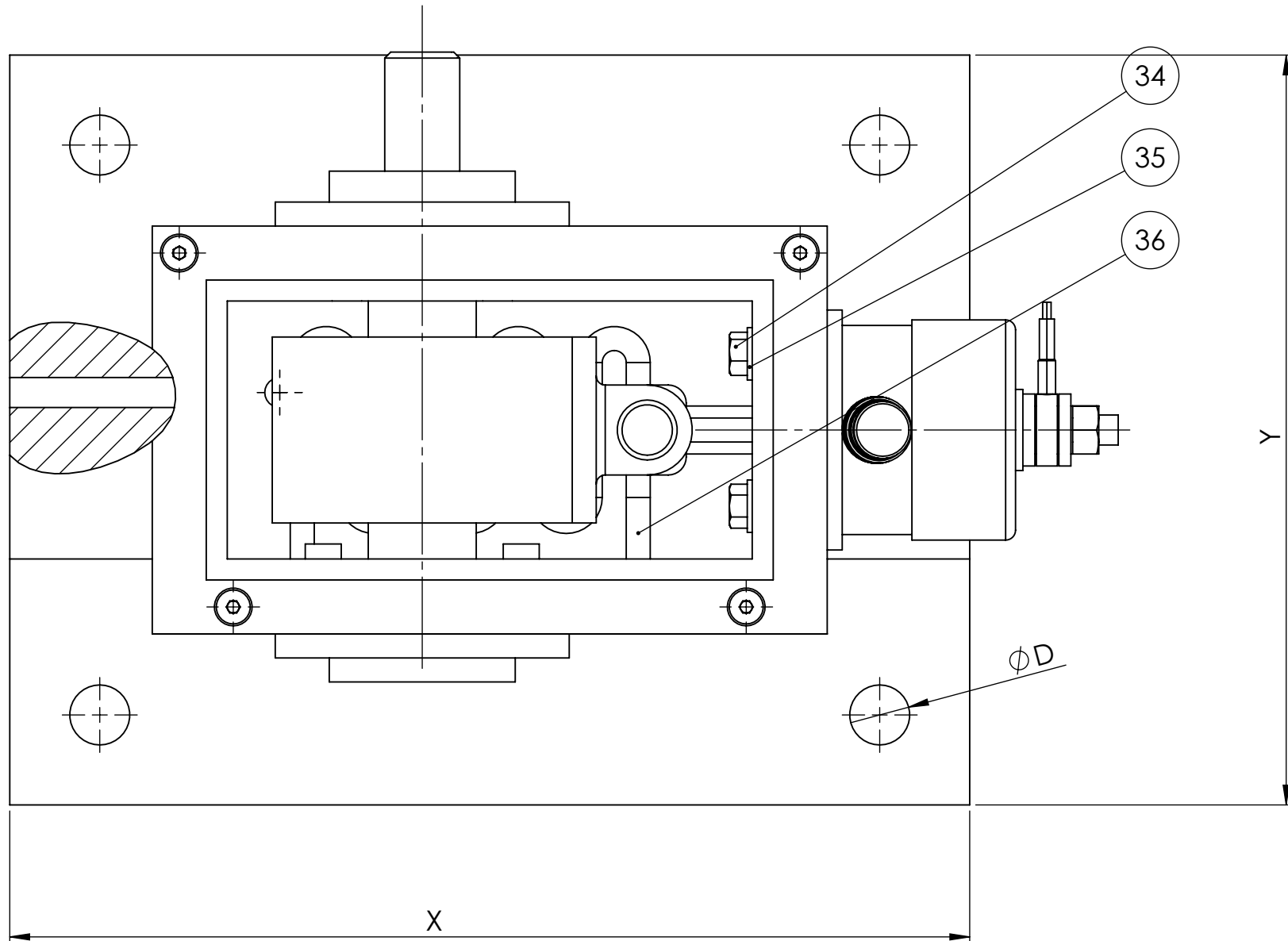
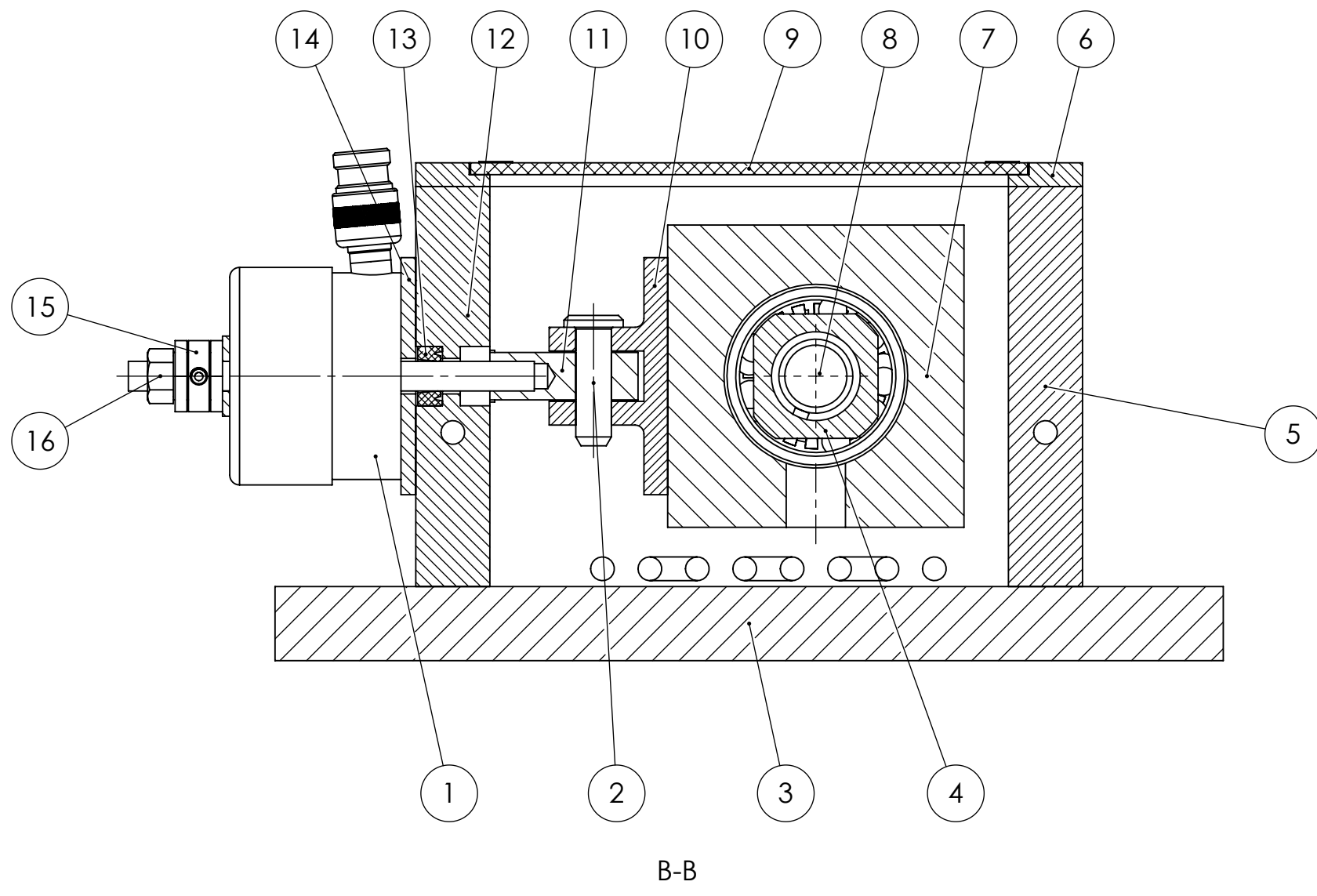
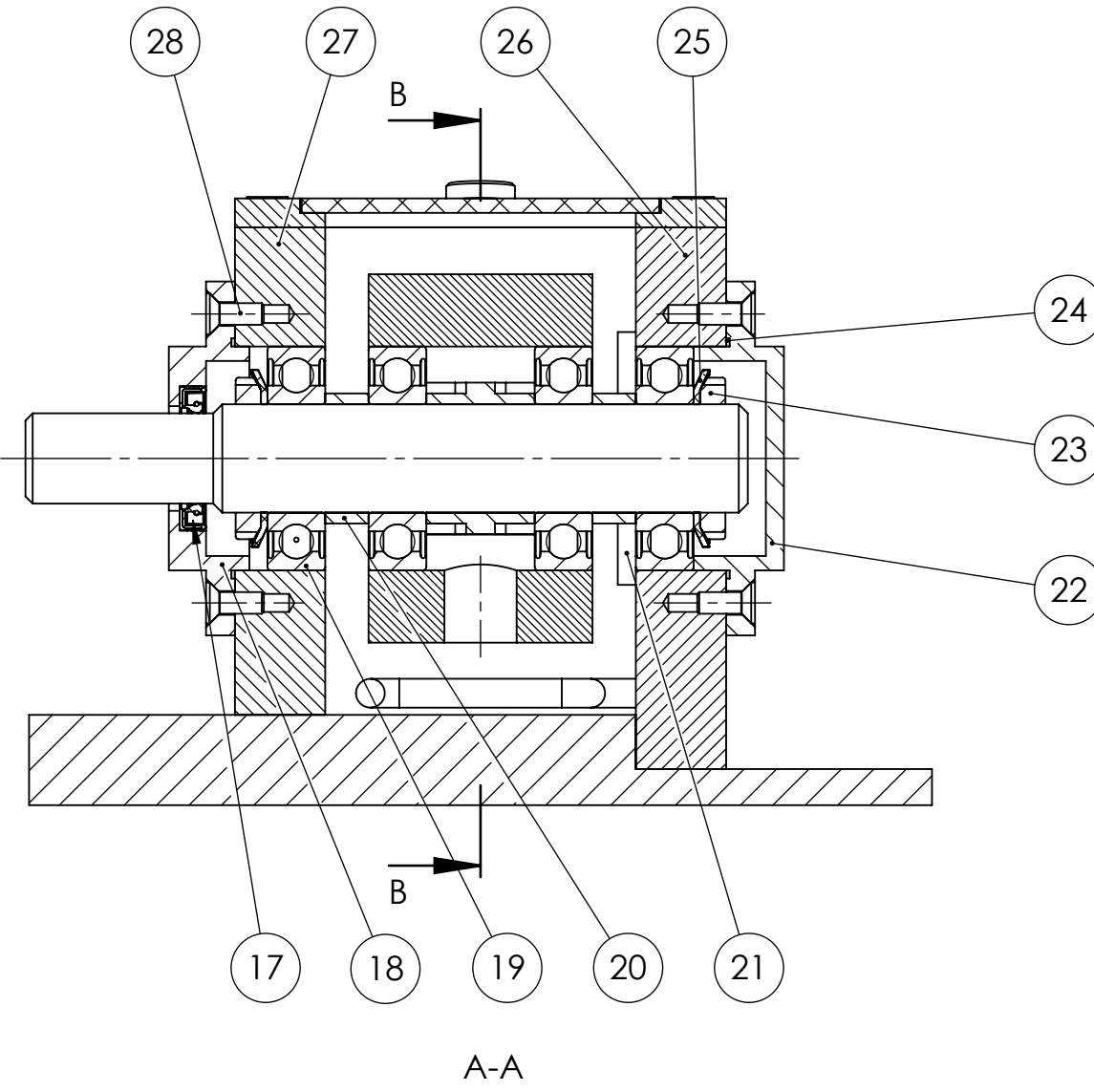
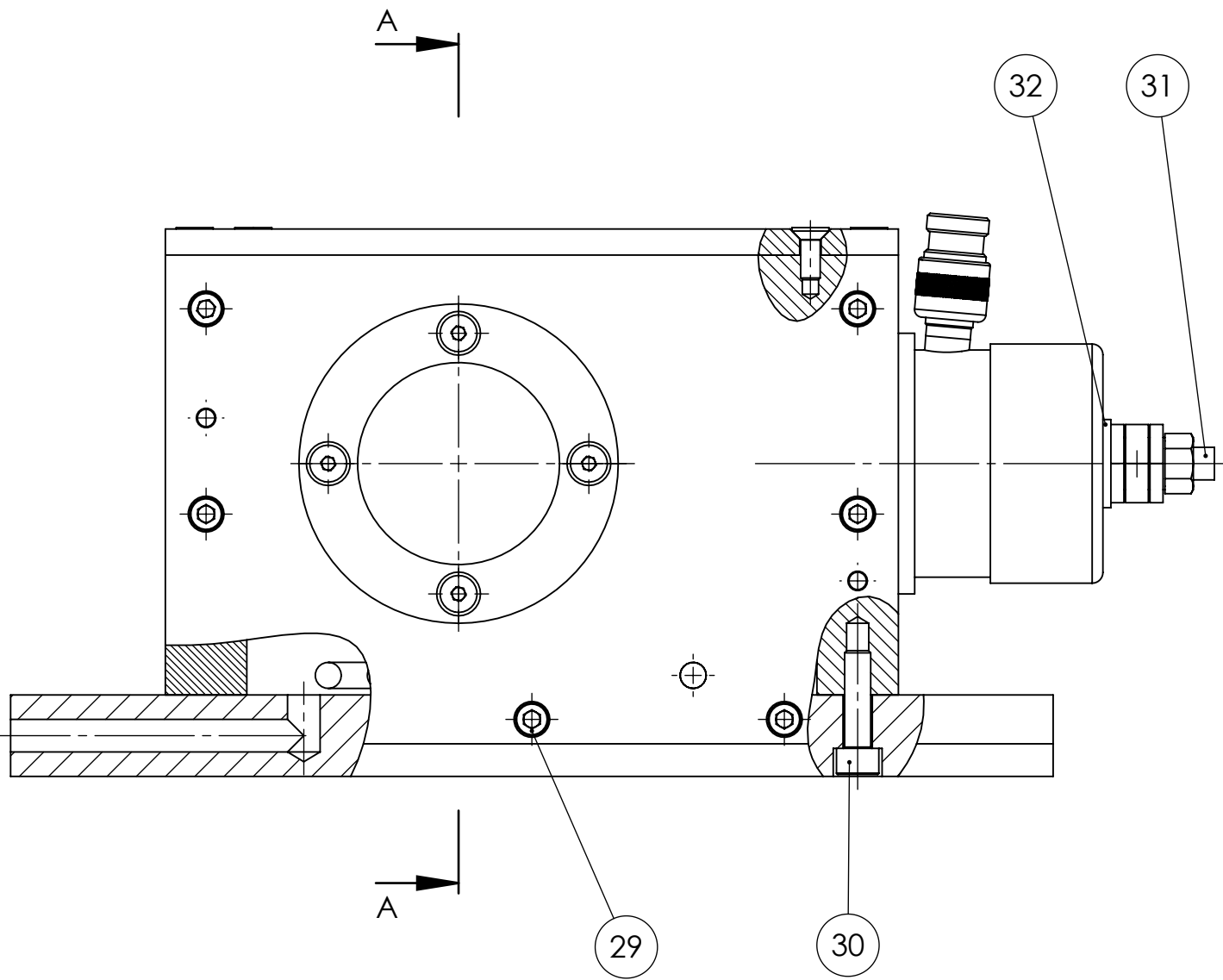
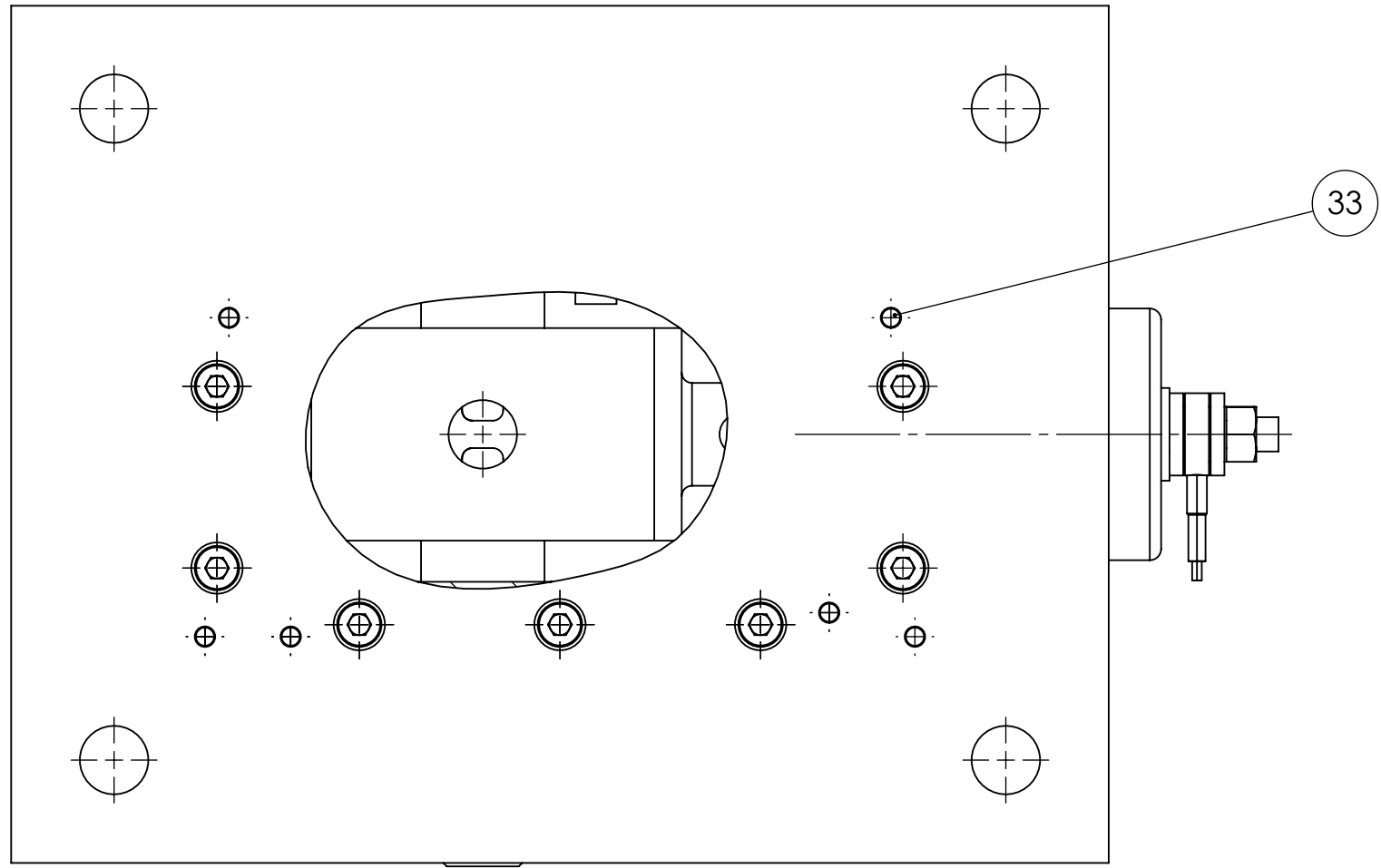
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Test rig assembly drawing

A.1 Test rig assembly drawing



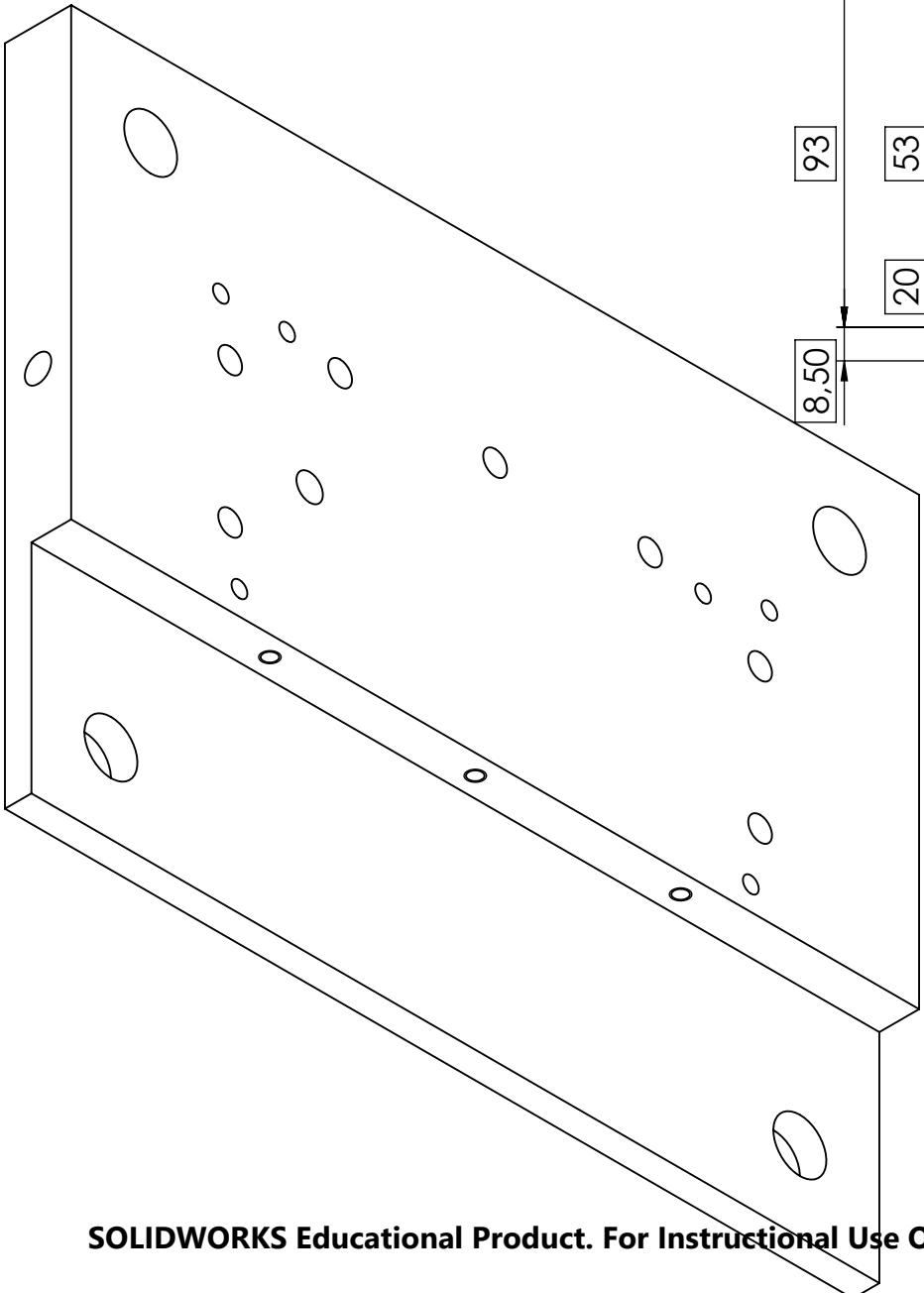
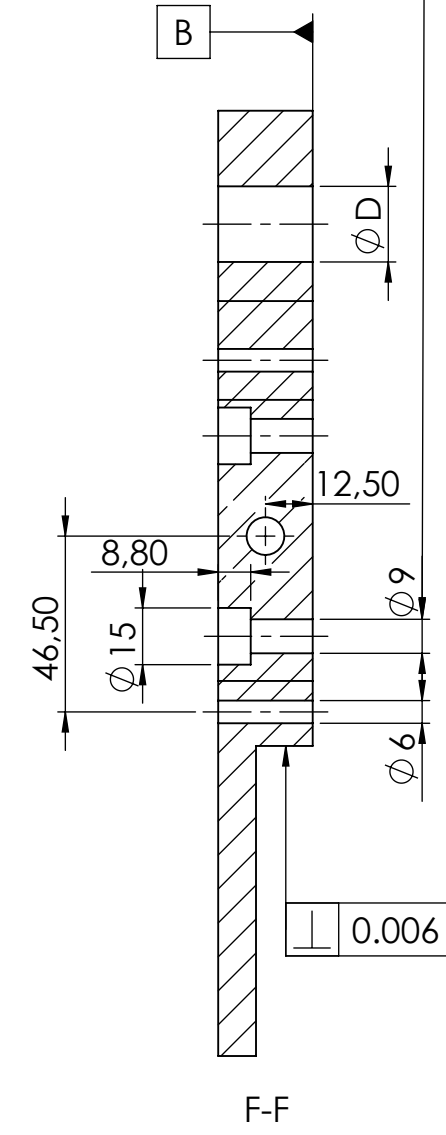
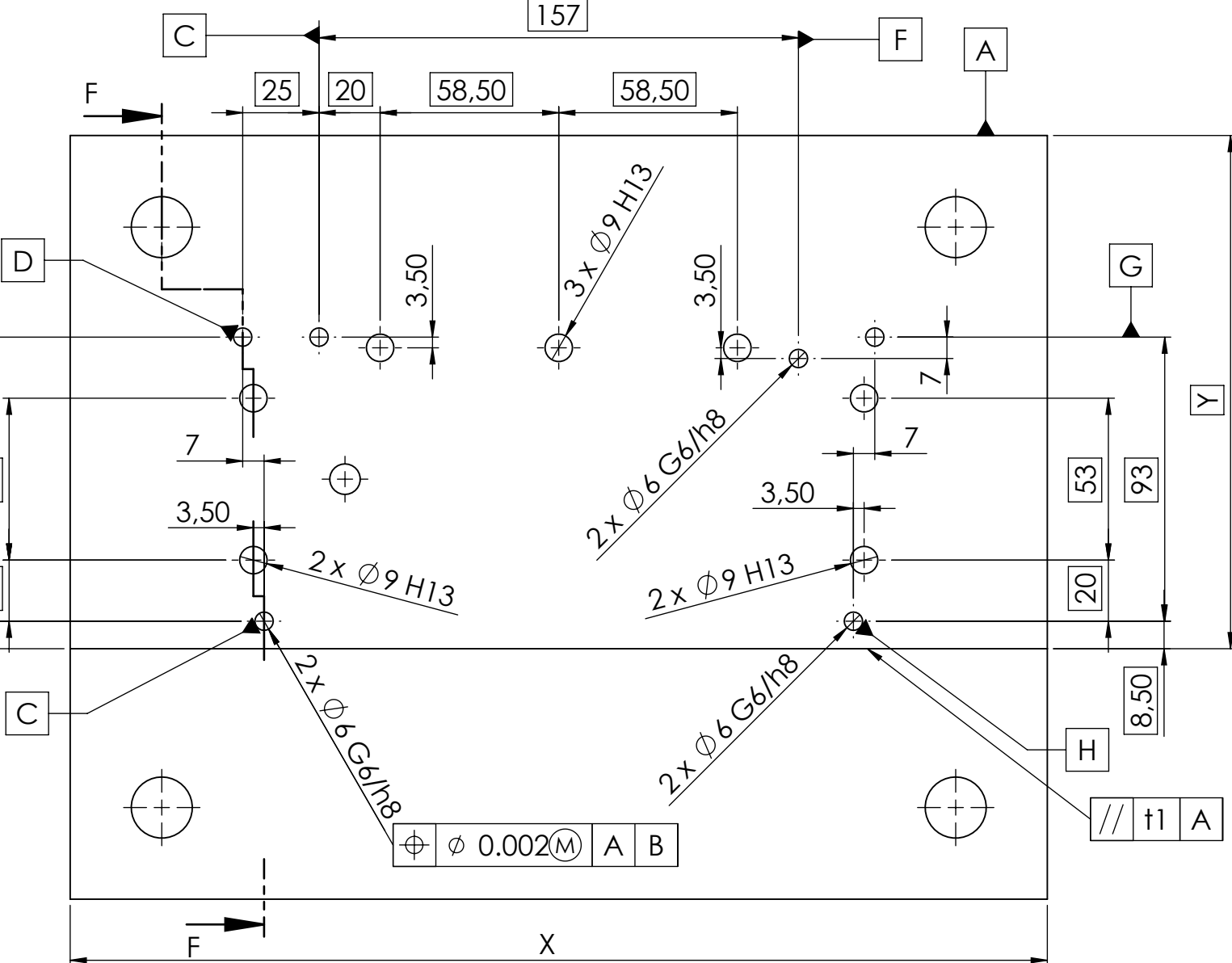
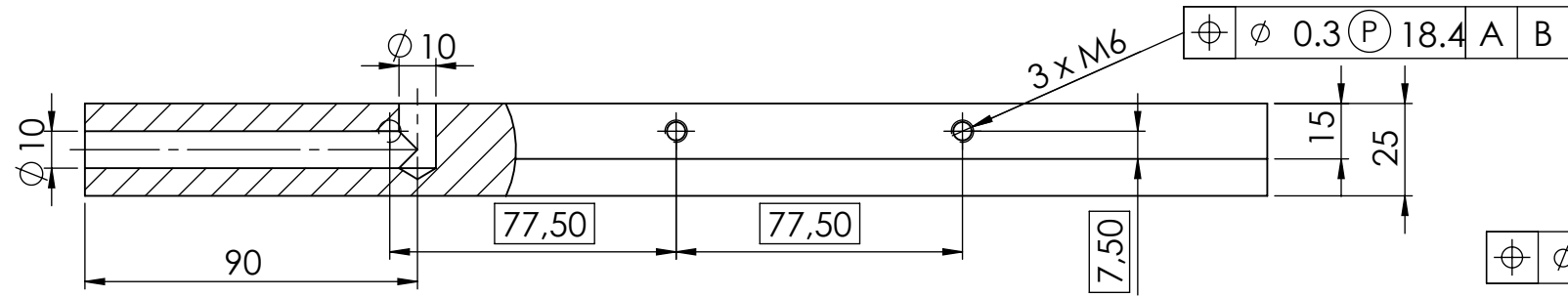
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2	Pin	ISO 2341 - A - 12 x 40 - St	1
3	Base	Material CK 45 K	1
4	Hub	Material 34 CrNiMo6	1
5	Left Wall	Material CK 45 K	1
6	Top Frame	Material CK 45 K	1
7	Bearing Sleeve	Material CK 45 K	1
8	Shaft	Material 34 CrNiMo6	1
9	Polycarbonate Plate		1
10	Yoke	Material 34 CrNiMo6	1
11	Rod End	Material 34 CrNiMo6	1
12	Right Wall	Material CK 45 K	1
13	Rod Seal	Simrit Merkel T20	1
14	Cylinder Spacer	Material CK 45 K	1
15	Load Cell	HBM KMR 60 kN	1
16	Rod Nut	ISO - 4033 - M10 - W - N	1
17	Seal	Simrit Simmerring BAU	1
18	Back Cover	Material CK 45 K	1
19	Ball Bearing		4
20	Spacer	Material CK 45 K	2
21	Bearing support	Material CK 45 K	2
22	Front Cover	Material CK 45 K	1
23	Lock Nut	SKF KM6	2
24	O-Ring Seal	Simrit	2
25	Lock Washer	SKF MB6	2
26	Front Wall	Material CK 45 K	1
27	Back Wall	Material CK 45 K	1
28	Countersunk M6 Bolt	ISO 10642 - M6 x 16 - 16N	12
29	Cyl. Head M6 Bolt	ISO 4762 M6 x 30 - 30N	7
30	Cyl. Head M8 Bolt	ISO 4762 M8 x 30 - 30N	11
31	Rod	Material 34 CrNiMo6	1
32	Cylinder hub	Material CK 45 K	1
33	Guiding Pin	ISO 8734 - 6 x 40 - A - St	8
34	Hex. 5/16" Bolt	HBOLT 0.3125-24x1.5x0.875-N	2
35	Washer	ISO 7089 M8 200HV	2
36	Cooler	Copper Pipe	1

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Finish:		Radial Bearings Test Rig	
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Approx Weight: Kg		Company Confidential	Checked/Approved by:
Projection Method: FIRST ANGLE		Legal Owner:	Part Number:
Sheet Size: A1		FEUP	Drawing Number:
		Rua Dr. Roberto Frias s/n 4200-465 Porto, Portugal	Sheet: 1 of 1
			Revision:

Appendix B

Production drawings

B.1 Base



Note:
The X, Y and D dimensions
should be determined
according to the place where
the test rig will be mounted.

✓ Ra 1.6

SCALE 1 : 2	Geometric tolerance ISO 8015 General tolerance ISO 2768 - mH ISO 13920 Roughness ISO 1302 Chamfer ISO 13715		Material: Ck 45 k	
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	Date	Name		
Prepared				
Verified				
Aproved				
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		Number:		
Rev.:	Date: 2017-06-15	Lang: en	Sheet: 1/1	

B.2 Front wall

B.3 Right side wall

8 7 6 5 4 3 2 1

F F

E E

D D

C C

B B

A A

$\oplus \phi 0.002 \text{ (M)}$ A B C

$\oplus \phi 0.5 \text{ (P)} 16.2$ A D - E

$\oplus \phi 0.5 \text{ (P)} 16.2$ A B C

$\oplus \phi 0.3 \text{ (P)} 18.4$ B F

$\oplus \phi 0.002 \text{ (M)}$ A B C

$Ra 1.6$

Note:
L dimension should be
chosen according to the
cartridge heater

SCALE 1 : 2	Geometric tolerance ISO 8015 General tolerance ISO 2768 - mH ISO 13920		Material: Ck 45 k	
	Roughness ISO 1302 Chamfer ISO 13715		Weight:	
	Data	Nome		
Prepared				
Verified				
Aproved				
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				Lang: en
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B.4 Left side wall

F

E

D

C

B

A

F

E

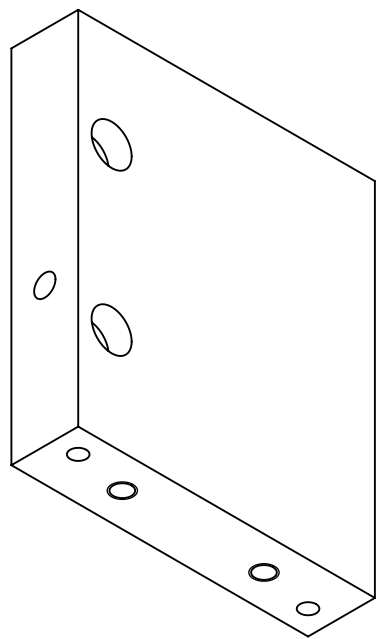
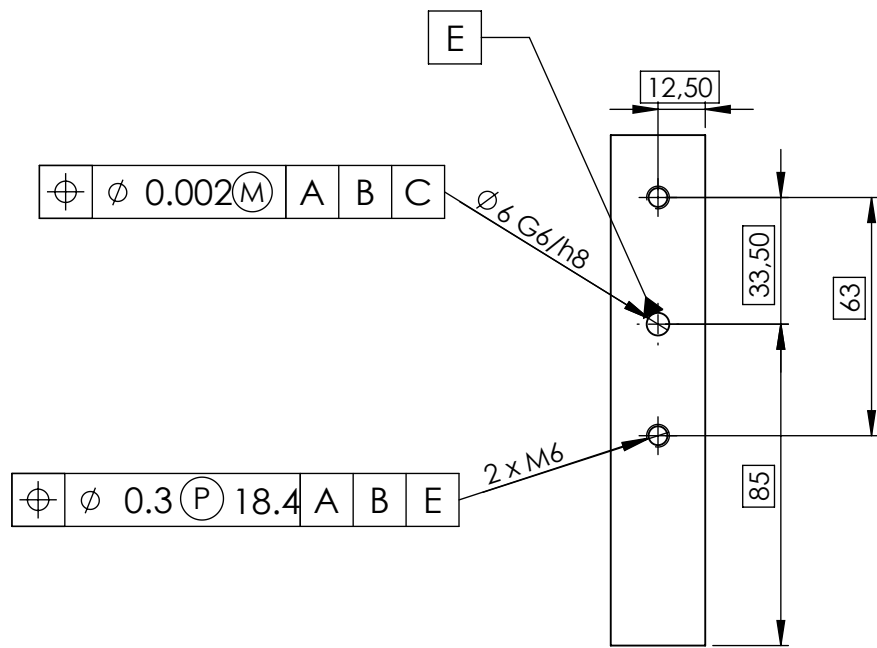
D

C

B

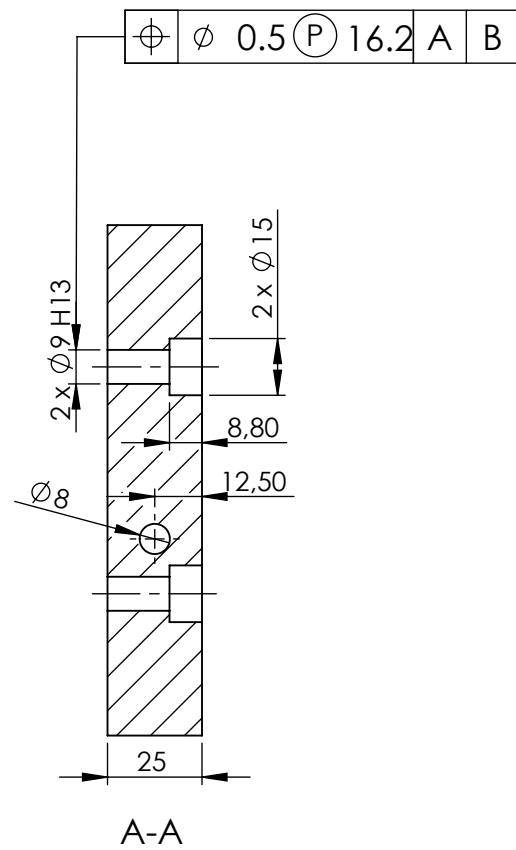
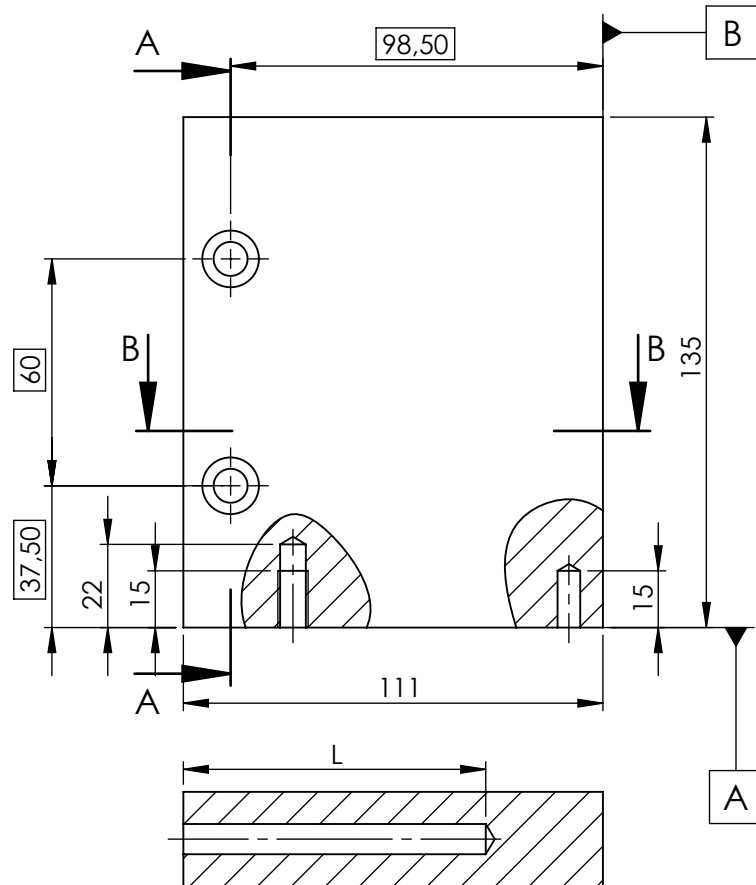
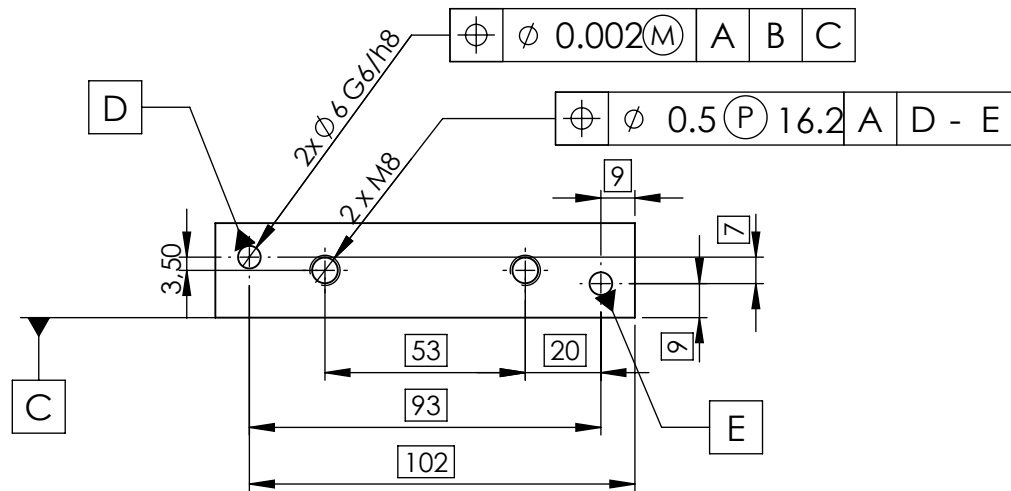
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8 7 6 5 4 3 2 1



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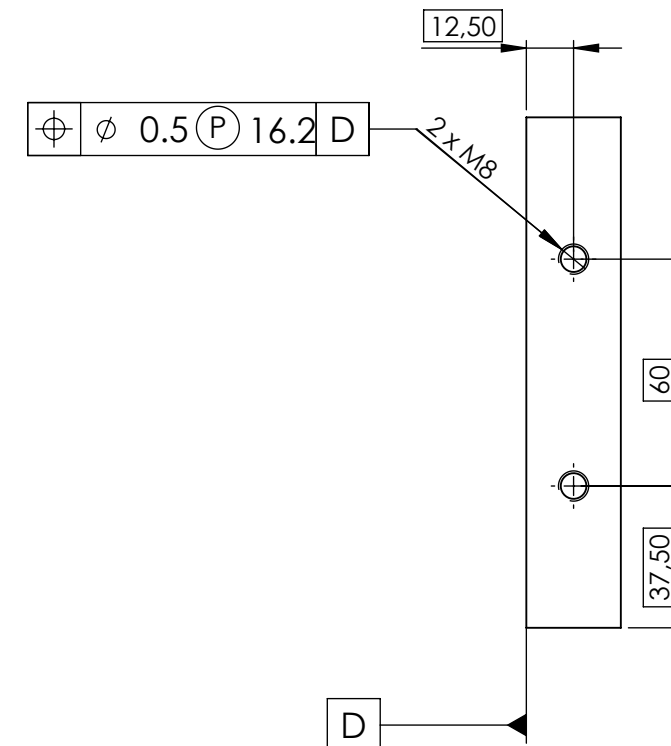
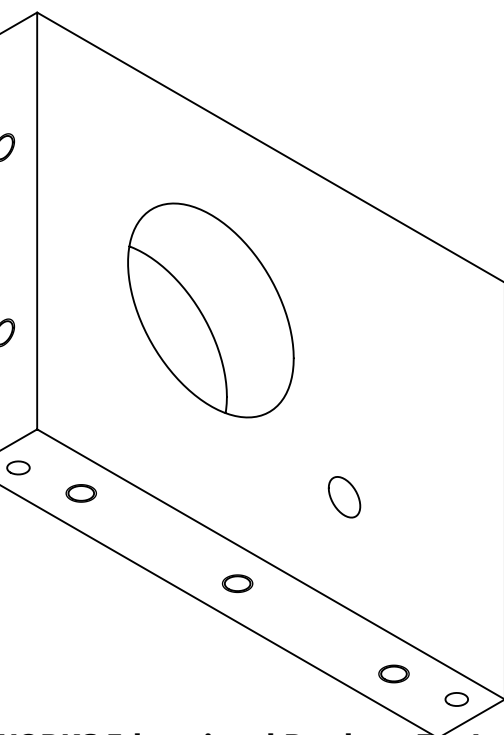
Note:
L dimension should be
chosen according to the
cartridge heater



Ra 1.6

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	Data	Nome		
Prepared				
Verified				
Aproved				
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FEUP		Left Side Wall		
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			Sheet: 1/1	

B.5 Back wall

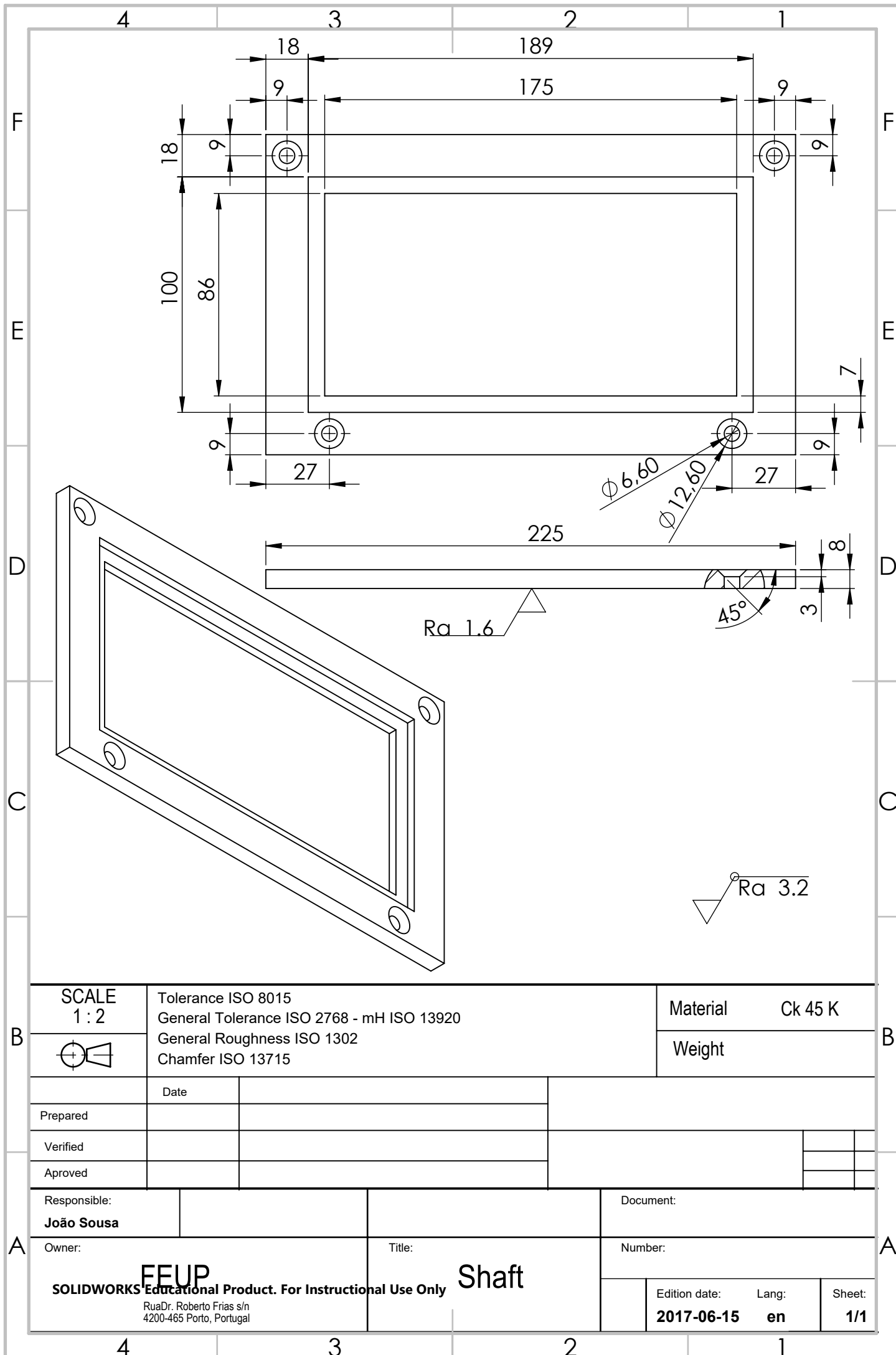


Ra 1.6

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		Data	Nome						
Prepared									
Verified									
Approved									
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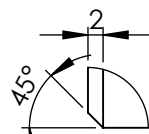
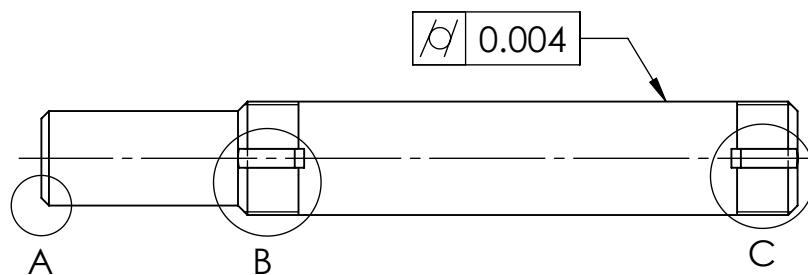
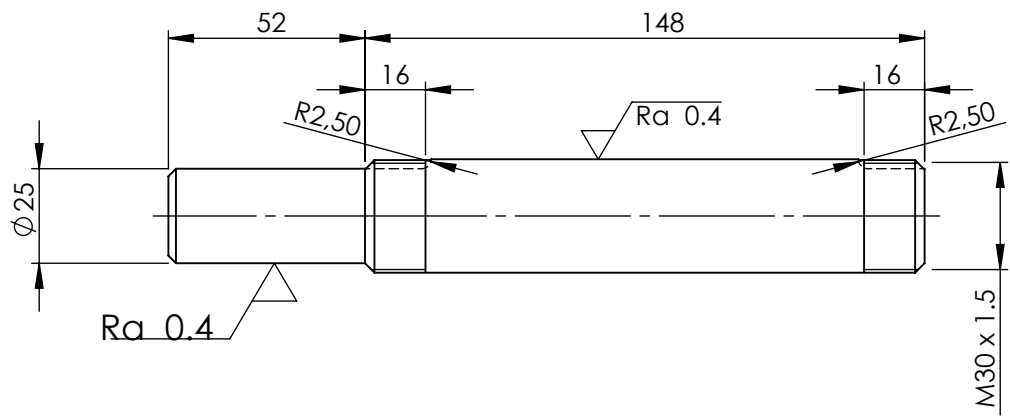
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B.6 Top frame

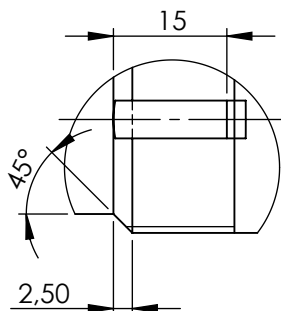


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João Sousa							
Owner:			Title:		Number:		
FEUP SOLIDWORKS Educational Product. For Instructional Use Only RuaDr. Roberto Frias s/n 4200-465 Porto, Portugal			Shaft				
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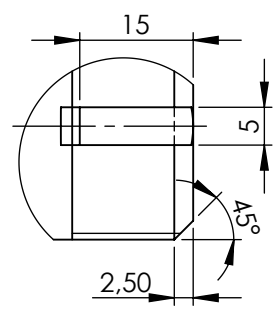
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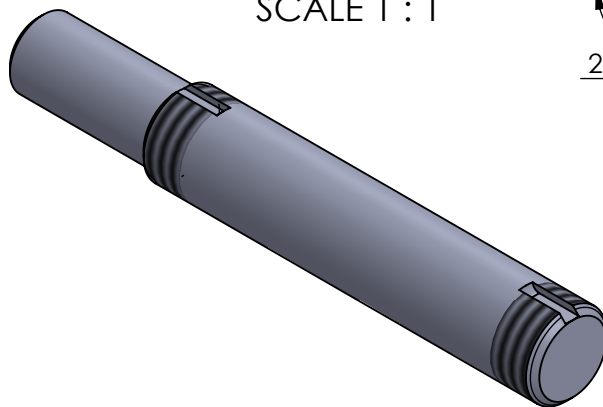
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B
SCALE 1 : 1

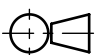


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SCALE 1 : 1



$Ra 3.2$

SCALE
1 : 2



Tolerance ISO 8015
General Tolerance ISO 2768 - mH ISO 13920
General Roughness ISO 1302
Chamfer ISO 13715

Material 34 CrNiMo 6

Weight

Date

Prepared

Verified

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Responsible:

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Title:

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Main Shaft

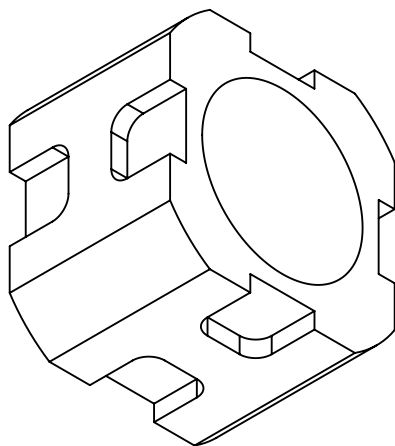
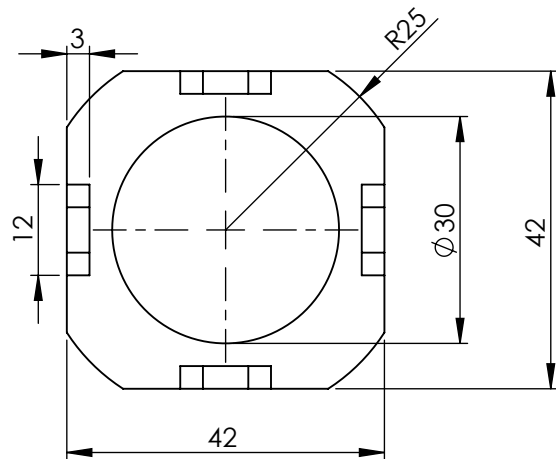
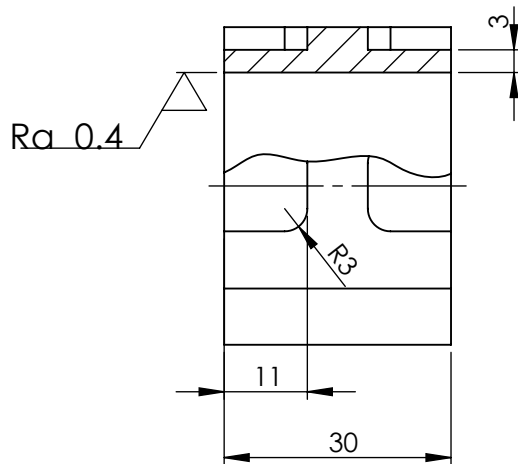
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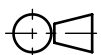
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B.8 Hub



✓ Ra 3.2

SCALE
1:1



Tolerance ISO 8015
General Tolerance ISO 2768 - mH ISO 13920
General Roughness ISO 1302
Chamfer ISO 13715

Material 34 CrNiMo 6

Weight

Date

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Verified

Aproved

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Hub

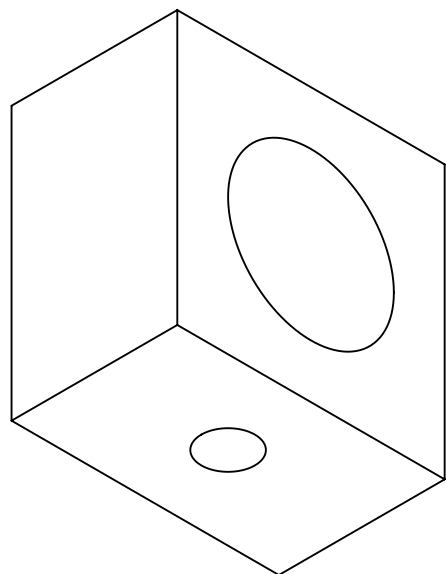
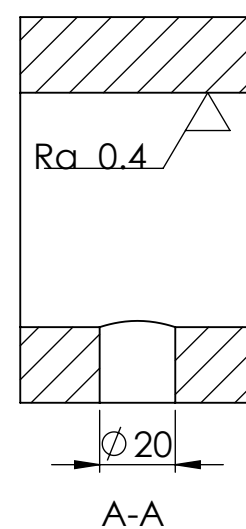
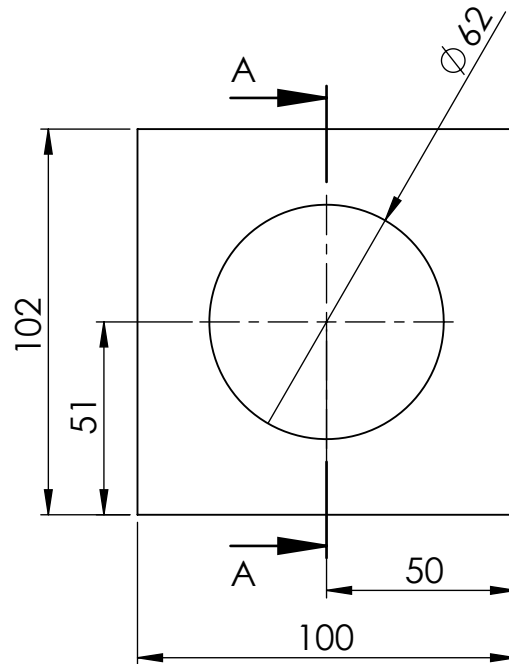
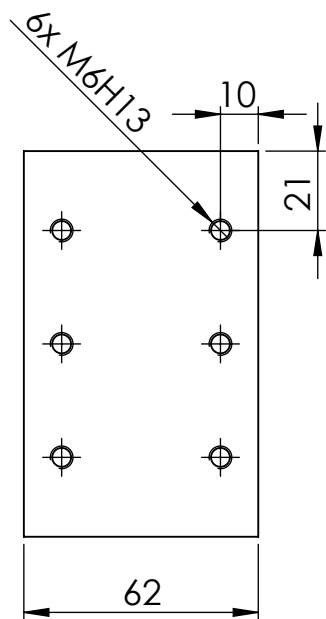
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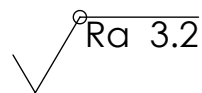
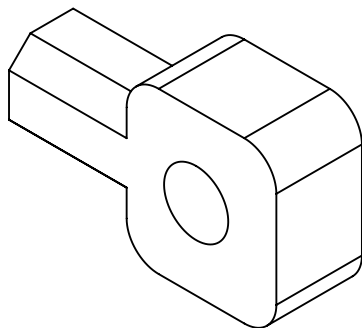
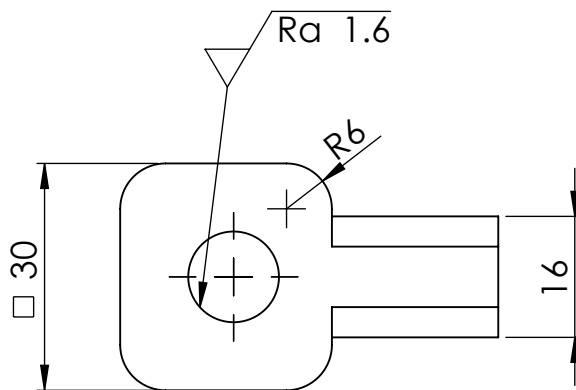
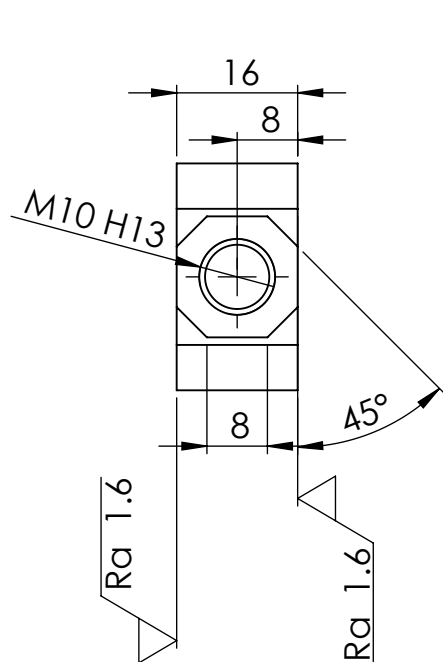
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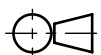
✓ Ra 3.2

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	Verified					
	Approved					
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Owner:		Title: Bearing Sleeve		Number:		
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B.10 Rod end



SCALE
1:1



Tolerance ISO 8015
General Tolerance ISO 2768 - mH ISO 13920
General Roughness ISO 1302
Chamfer ISO 13715

Material 34 CrNiMo 6

Weight

Date

Prepared

Verified

Aproved

Responsible:

João Sousa

Document:

Owner:

Title:

Number:

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Rod End

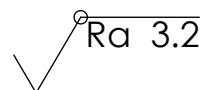
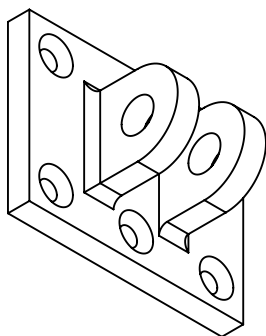
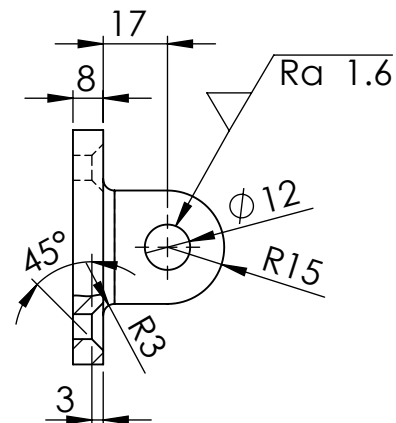
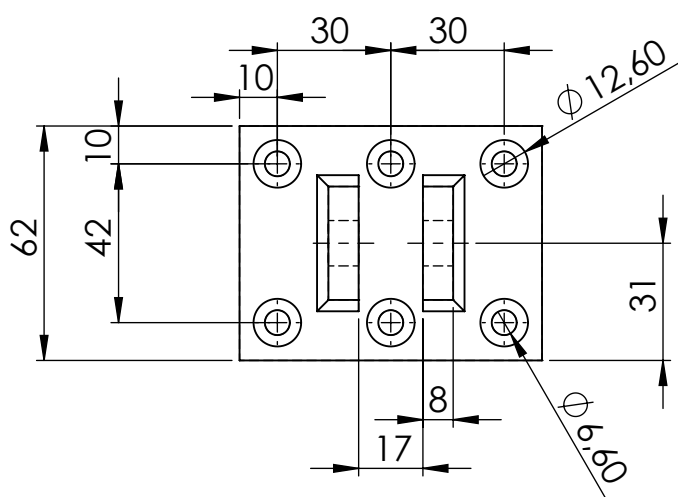
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B.11 Yoke



SCALE
1 : 2



Tolerance ISO 8015
General Tolerance ISO 2768 - mH ISO 13920
General Roughness ISO 1302
Chamfer ISO 13715

Material 34 CrNiMo 6

Weight

Date

Prepared

Verified

Aproved

Responsible:

João Sousa

Document:

Owner:

Title:

Number:

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Yoke

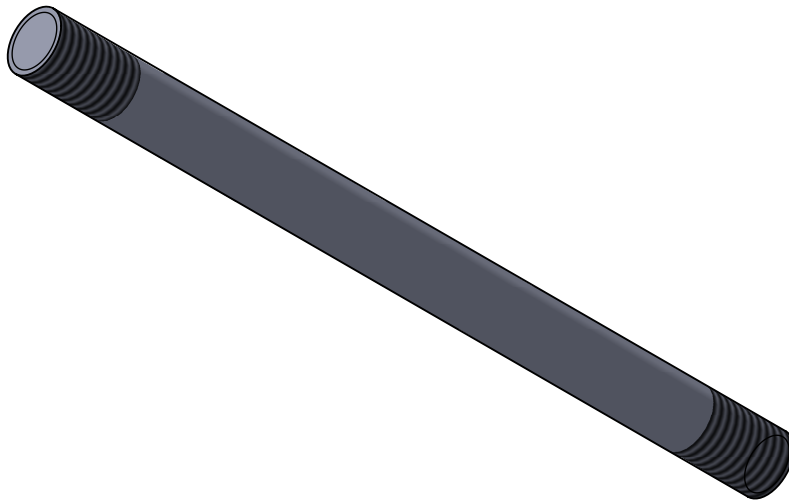
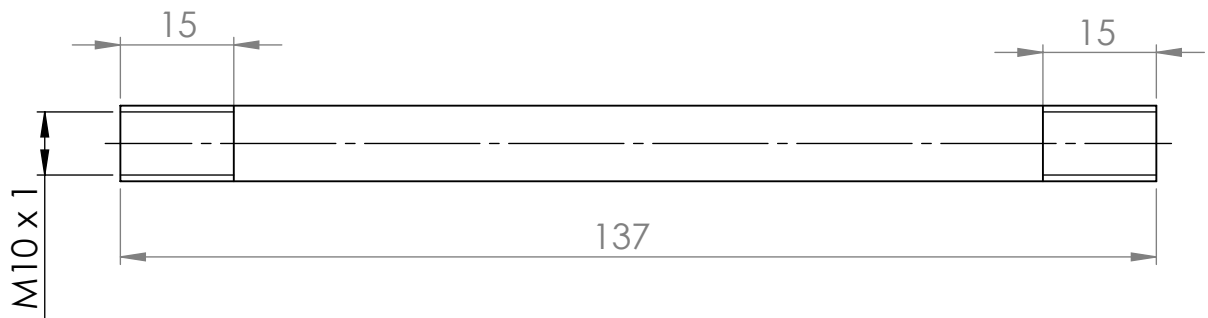
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B.12 Rod



✓ Ra 1.6

SCALE
1 : 1

Tolerance ISO 8015
General Tolerance ISO 2768 - mH ISO 13920
General Roughness ISO 1302
Chamfer ISO 13715

Material 34 CrNiMo 6

Weight



Date

Prepared

Verified

Approved

Responsible:

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Title:

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Rod

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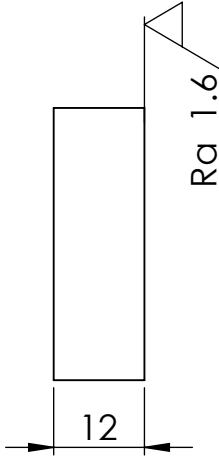
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4200-465 Porto, Portugal

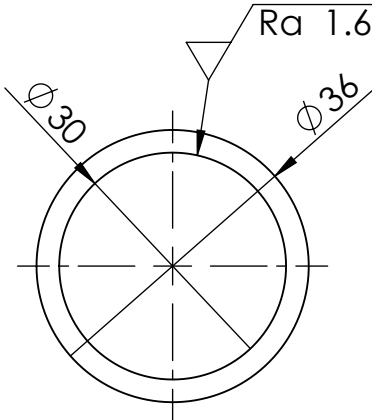
B.13 Spacer

	4	3	2	1	
F					F
E					E
D					D
C					C
B					B
A					A



12

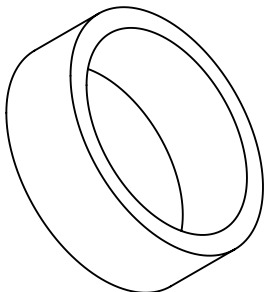
Ra 1.6

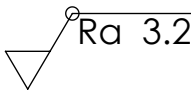


Ø30

Ø36

Ra 1.6

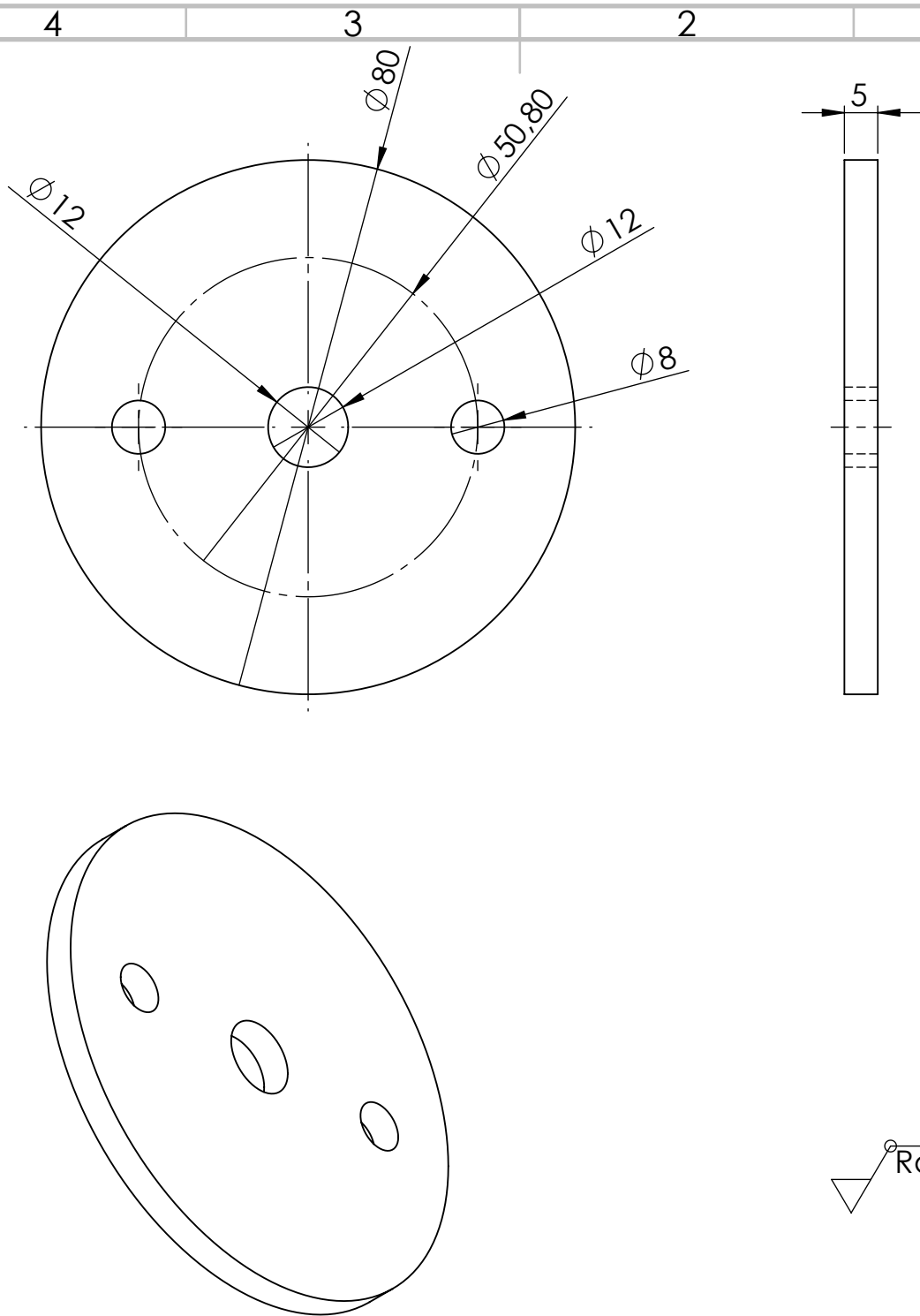




Ra 3.2

SCALE 1 : 1 	Tolerance ISO 8015 General Tolerance ISO 2768 - mH ISO 13920 General Roughness ISO 1302 Chamfer ISO 13715			Material Ck 45 K	
				Weight	
	Date				
Prepared					
Verified					
Aproved					
Responsible: João Sousa			Document:		
Owner:		Title: FEUP SOLIDWORKS Educational Product. For Instructional Use Only Rua Dr. Roberto Frias s/n 4200-465 Porto, Portugal Spacer		Number:	
		Edition date: 2017-06-15	Lang: en	Sheet: 1/1	

B.14 Cylinder spacer



SCALE
1:1

Tolerance ISO 8015
General Tolerance ISO 2768 - mH ISO 13920
General Roughness ISO 1302
Chamfer ISO 13715

Material Ck 45 K

Weight



Date

Prepared

Verified

Aproved

Responsible:

João Sousa

Document:

Owner:

Title:

Number:

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4200-465 Porto, Portugal

Cylinder Spacer

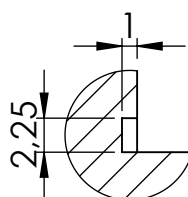
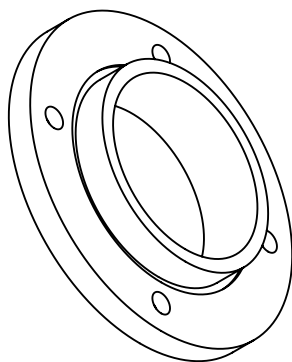
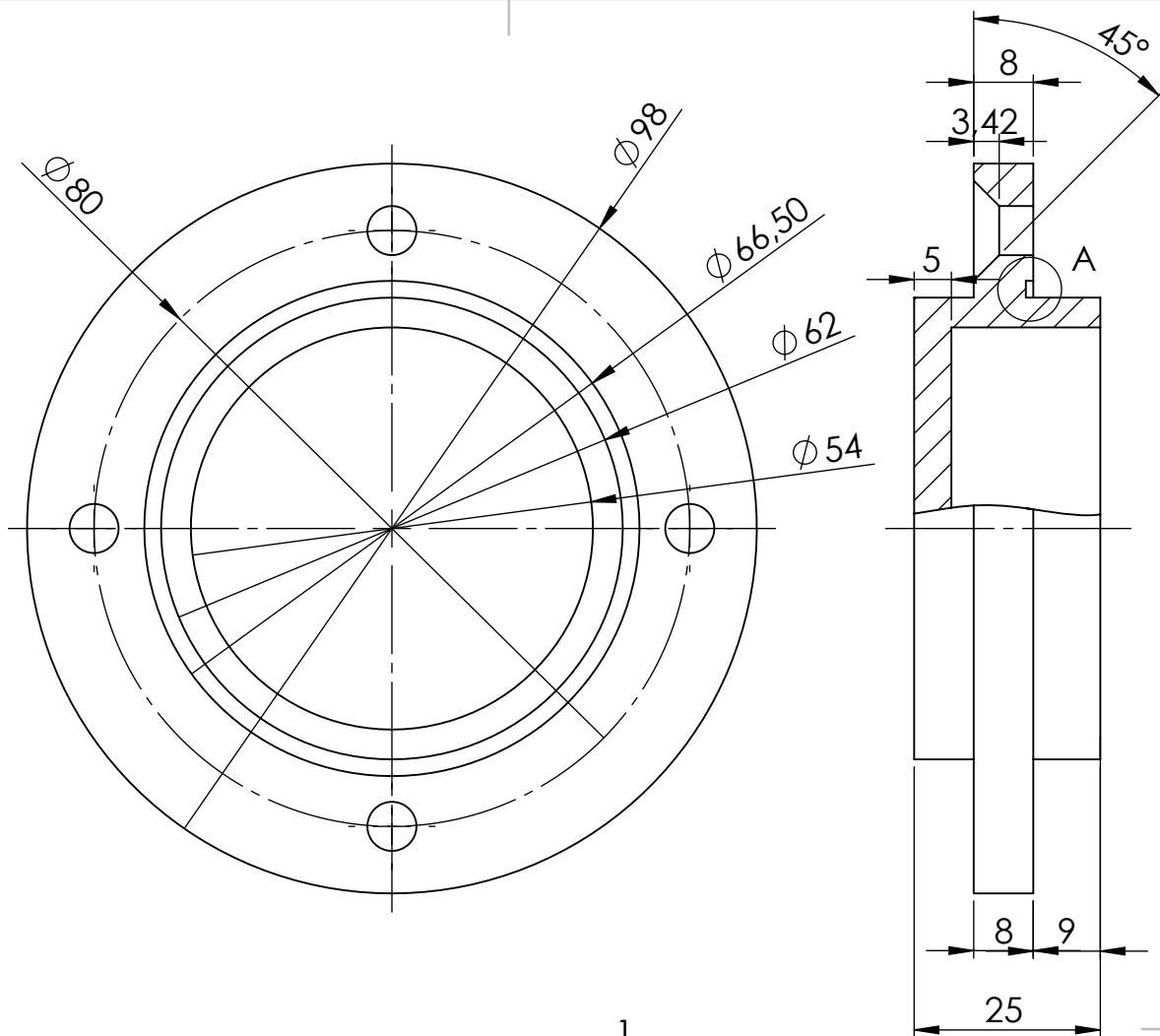
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2017-06-15 en

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1/1

B.15 Front cover

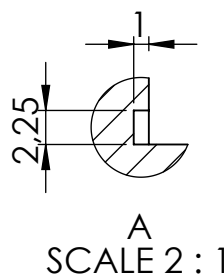
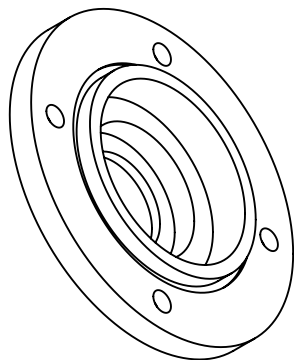
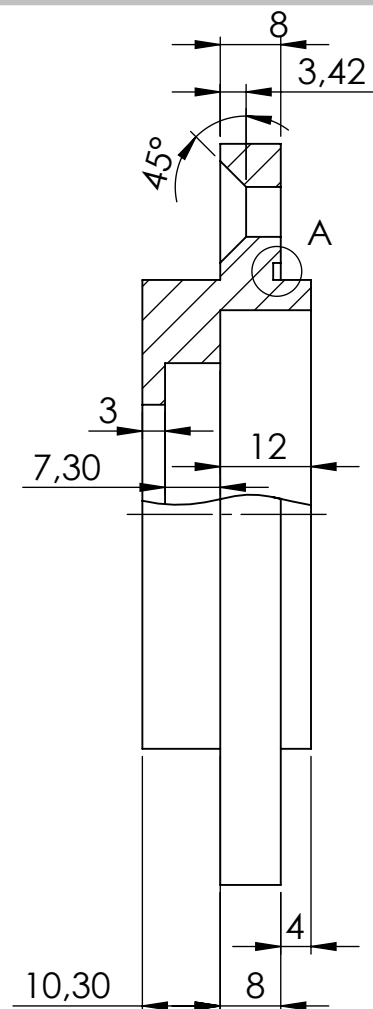
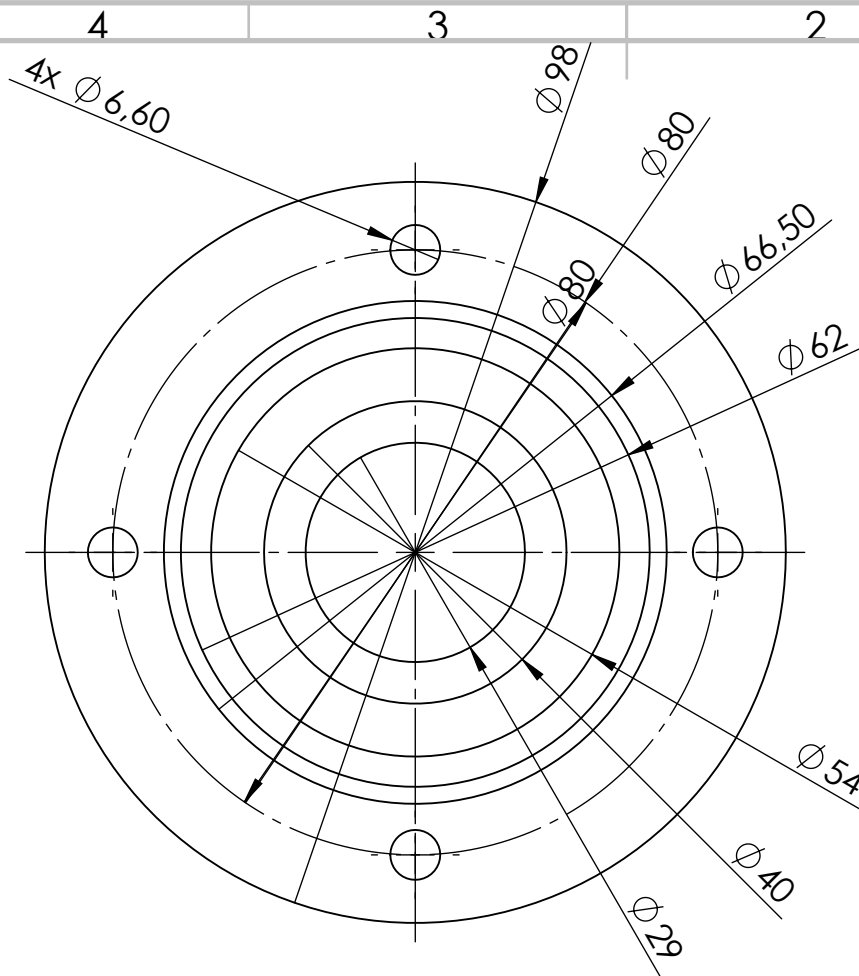


A
SCALE 2 : 1

✓ Ra 3.2

B	SCALE 1 : 1		Tolerance ISO 8015 General Tolerance ISO 2768 - mH ISO 13920 General Roughness ISO 1302 Chamfer ISO 13715		Material Ck 45 K	
					Weight	
A	Prepared	Date				
	Verified					
	Aproved					
	Responsible: João Sousa		Document:			
	Owner:	Title: Front Cover		Number:		
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B.16 Back cover



✓ Ra 3.2

B	SCALE 1 : 1		Tolerance ISO 8015 General Tolerance ISO 2768 - mH ISO 13920 General Roughness ISO 1302 Chamfer ISO 13715		Material Ck 45 K	
					Weight	
	Date					
	Prepared					
	Verified					
	Aproved					
A	Responsible: João Sousa				Document:	
	Owner:		Title: Back Cover		Number:	
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Materials

After some iterations in the design of each component, two steels was chosen to build the test rig. For the sake of simplicity, the materials are presented here, so it is possible to make reference to their properties throughout the report.

C.1 Steel - DIN Ck 45 K

The Ck 45 K Steel is a common carbon steel. It presents fair mechanical properties it is widely available and has a low cost, being generally applied in mechanical components, hence, it was chosen to build the carter and the components with low requirements. The properties of Ck45 K steel are presented in Table C.1.

Table C.1: Ck 45 K mechanical properties.

Mechanical Properties			
Hardness	R_m	$R_p0.2\%$	A
HBW (max.)	MPa	MPa	%
186	≥ 580	≥ 325	≥ 4

Where R_m is the rupture stress, $R_p0.2\%$ is the yield stress and A is the maximum strain.

C.2 Steel - 34 CrNiMo 6

For the shaft construction, 34 CrNiMo 6 nitrided steel was chosen because it has very good mechanical properties Table C.2, allowing the shaft to be capable of resisting higher loads with high fatigue life.

Table C.2: 34 CrNiMo 6 mechanical properties.

Mechanical Properties		
R_m	$R_p0.2\%$	A
MPa	MPa	%
≥ 1200	≥ 1000	≥ 9

Note that the yield stress at 0.2% strain can be related to the yield stress at different

stress states, Equation (C.1).

$$R_p 0.2\% = R_{pt} = R_{pc} = R_{pb} \quad (\text{C.1})$$

Where R_{pt} is the yield stress in tension, R_{pc} is the yield stress in compression, and R_{pb} is the yield stress in bending.

Shear stress, R_{ps} , according to von Mises yield criterion can be given by the Equation (C.2).

$$R_{ps} = \frac{R_{pt}}{\sqrt{3}} \quad (\text{C.2})$$

Four-point bending analysis

Considering that the shaft is supported by A and D supports, the load is applied in the positions B and C and the shaft is symmetric. The shaft loading is presented in the Figure D.1.

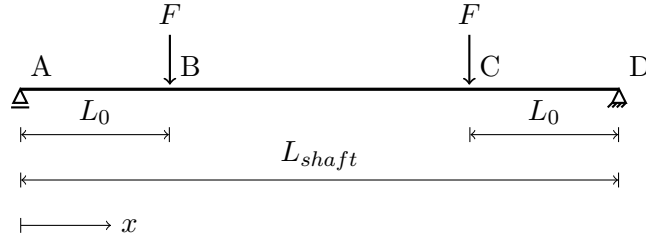


Figure D.1: Four-point bending scheme.

To find the reaction force of the supports A and D it is needed to make a sum of bending moments in each support, as shown in Equations (D.1) and (D.2).

$$\sum M_A = 0 \quad (D.1)$$

$$\sum M_D = 0 \quad (D.2)$$

The Equations (D.1) and (D.2) can be rewritten as in the Equations (D.3) and (D.4) respectively, by the application of the bending moments.

$$-F \cdot L_0 - F \cdot (L_{shaft} - L_0) + F_D^y \cdot L_{shaft} = 0 \quad (D.3)$$

$$-F_A^y \cdot L_{shaft} + F \cdot (L_{shaft} - L_0) + F \cdot L_0 = 0 \quad (D.4)$$

Being the F_A^y and F_D^y the vertical reaction forces of the supports A and D, respectively. The amount of bending moment $F \cdot L_0$ cancels in each equation. Thus, each reaction force is equal to the force applied, as Equations (D.5) and (D.6) shown.

$$F_D^y = F \quad (D.5)$$

$$F_A^y = F \quad (D.6)$$

About the section loads, they are obtained by the load balance for the three different zones such as A-B, B-C and C-D. The Equations (D.7), (D.8) and (D.9) and the Figure D.2 show this.

$$V = F, \quad \text{for} \quad 0 < x < L_0 \quad (\text{D.7})$$

$$V = 0, \quad \text{for} \quad L_0 < x < L - L_0 \quad (\text{D.8})$$

$$V = -F, \quad \text{for} \quad L - L_0 < x < L_{shaft} \quad (\text{D.9})$$

Converting the shear loads in the sections into a load diagram, it is represented as in Figure D.2.

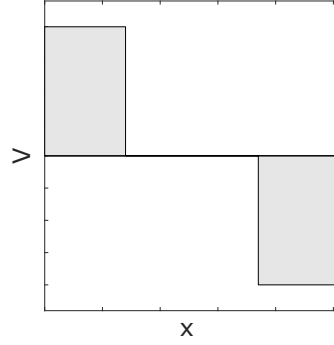


Figure D.2: Shear diagram.

About the bending moment at each section, it can be obtained by the balance of bending moments, which is done in the Equations (D.10), (D.11) and (D.12). There are three different kind of evolution of stress along the length and in the center reaches the maximum stress level, as can be seen in Figure D.3.

$$M_b = F \cdot x, \quad \text{for} \quad 0 < x < L_0 \quad (\text{D.10})$$

$$M_b = F \cdot L_0, \quad \text{for} \quad L_0 < x < L - L_0 \quad (\text{D.11})$$

$$M_b = F \cdot (L_{shaft} - x), \quad \text{for} \quad L - L_0 < x < L_{shaft} \quad (\text{D.12})$$

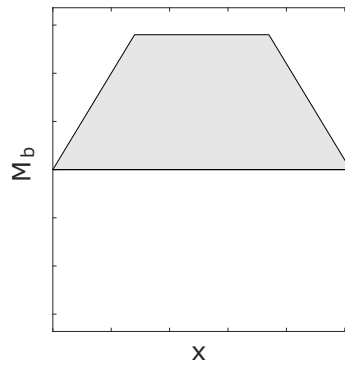


Figure D.3: Bending diagram.

As expected, the maximum bending moment occurs at the central zone, which shows that L_0 has an important influence and should be as small as possible to cause the smallest stress possible.

KISSsoft Reports

E.1 KISSsoft Reports - Shaft

E.1.1 KISSsoft Report - Shaft with hub mounted with interference fit

Name : veio30_12

Changed by: up201106920

on: 11.06.2017

at: 22:54:41

Important hint: At least one warning has occurred during the calculation:

1-> The required service life of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' is not achieved!
The static safety is low (in range 0.5 - 2.0).

Please check whether these values are acceptable or not.

2-> The required service life of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' is not achieved!
The static safety is low (in range 0.5 - 2.0).

Please check whether these values are acceptable or not.

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	Shaft 1
Drawing	
Initial position (mm)	0.000
Length (mm)	118.000
Speed (1/min)	1500.00
Sense of rotation: clockwise	
Material	34 CrNiMo 6 (2)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	0.653
(Notice: Weight stands for the shaft only without considering the gears)	
Weight of shaft, including additional masses (kg)	0.653
Mass moment of inertia (kg*mm ²)	73.473
Momentum of mass GD2 (Nm ²)	0.003
Position in space (°)	0.000
Gears mounted with stiffness according to ISO	
Consider deformations due to shearing	
Shear correction coefficient	1.100
Contact angle of rolling bearings is considered	
Tolerance field: Mean value	
Housing material	C45 (1)
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature of housing (°C)	80.000

Thermal housing reference point (mm)	0.000
Reference temperature (°C)	20.000

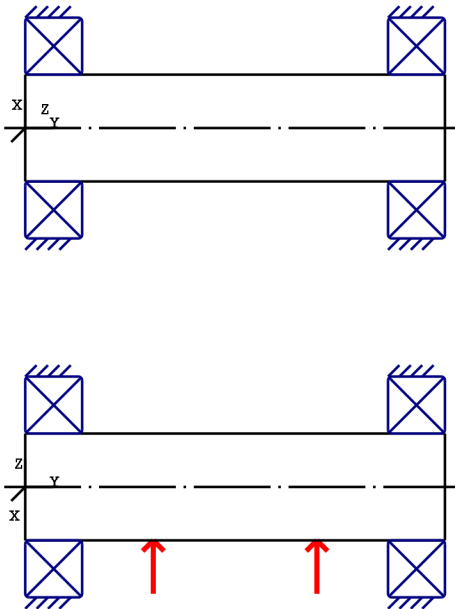


Figure: Load applications

Shaft definition (Shaft 1)

Outer contour

Cylinder (veio)		0.000mm ... 118.000mm
Diameter (mm)	[d]	30.0000
Length (mm)	[l]	118.0000
Surface roughness (µm)	[Rz]	2.8000
Interference fit (Interference fit)		44.000mm ... 74.000mm
l=30.00 (mm), Typ=1, Turned (Ra=3.2µm/125µin)		

Forces

Type of force element	Centric force	
Label in the model	carga1	
Position on shaft (mm)	[y _{local}]	36.0000
Position in global system (mm)	[y _{global}]	36.0000
Length of load application (mm)		16.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000

Shearing force X (N)	0.0000
Shearing force Z (N)	20000.0000
Bending moment X (Nm)	0.0000
Bending moment Z (Nm)	0.0000

Type of force element		Centric force
Label in the model		carga2
Position on shaft (mm)	[ylocal]	82.0000
Position in global system (mm)	[yglobal]	82.0000
Length of load application (mm)		16.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		20000.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

Bearing

Label in the model		Rolling bearing
Bearing type		SKF *NJ 206 ECJ
Bearing type		Cylindrical roller bearing (single row)
Bearing position (mm)	[ylocal]	8.000
Bearing position (mm)	[yglobal]	8.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	62.000
Width (mm)	[b]	16.000
Corner radius (mm)	[r]	1.000
Basic static load rating	[C ₀]	36.500
Basic dynamic load rating	[C]	44.000
Fatigue load rating	[C _U]	4.600
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Label in the model		Rolling bearing
Bearing type		SKF *NJ 206 ECJ
Bearing type		Cylindrical roller bearing (single row)
Bearing position (mm)	[ylocal]	110.000
Bearing position (mm)	[yglobal]	110.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	62.000
Width (mm)	[b]	16.000
Corner radius (mm)	[r]	1.000
Basic static load rating	[C ₀]	36.500
Basic dynamic load rating	[C]	44.000
Fatigue load rating	[C _U]	4.600
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Results

Shaft

Maximum deflection (μm)	105.793
Position of the maximum (mm)	56.000
Mass center of gravity (mm)	59.000
Total axial load (N)	0.000
Torsion under torque ($^\circ$)	0.000

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	[u _A]	10.00	μm
Rolling bearings, classical calculation (contact angle considered)			

Shaft 'Shaft 1' Rolling bearing 'Rolling bearing'

Position (Y-coordinate)	[y]	8.00	mm
Equivalent load	[P]	20.00	kN
Equivalent load	[P ₀]	20.00	kN
Life modification factor for reliability[a ₁]		1.000	
Nominal bearing service life	[L _{nh}]	153.96	h
Static safety factor	[S ₀]	1.83	
Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	-19.997	kN
Bearing reaction force	[F _r]	19.997	kN (-90°)
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _{rr}]	0.103	Nm
Sliding moment of friction	[M _{sl}]	0.056	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 10000/1 EN:2013			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.159	Nm
Power loss	[P _{loss}]	24.995	W

The moment of friction is calculated according to the details in SKF Catalog 2013.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

Displacement of bearing	[u _x]	0.000	μm
Displacement of bearing	[u _y]	0.000	μm
Displacement of bearing	[u _z]	16.250	μm
Displacement of bearing	[u _r]	16.250	μm (90°)
Misalignment of bearing	[r _x]	2.498	mrad (8.59')
Misalignment of bearing	[r _y]	-0.000	mrad (0')
Misalignment of bearing	[r _z]	-0.000	mrad (0')
Misalignment of bearing	[r _r]	2.498	mrad (8.59')

Shaft 'Shaft 1' Rolling bearing 'Rolling bearing'

Position (Y-coordinate)	[y]	110.00	mm
Equivalent load	[P]	20.00	kN
Equivalent load	[P ₀]	20.00	kN
Life modification factor for reliability[a ₁]		1.000	

Nominal bearing service life	[L _{nh}]	153.96	h
Static safety factor	[S ₀]	1.83	
Bearing reaction force	[F _x]	-0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	-19.997	kN
Bearing reaction force	[F _r]	19.997	kN (-90°)
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _{rr}]	0.103	Nm
Sliding moment of friction	[M _{sl}]	0.056	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 10000/1 EN:2013			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.159	Nm
Power loss	[P _{loss}]	24.995	W

The moment of friction is calculated according to the details in SKF Catalog 2013.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

Displacement of bearing	[u _x]	0.000	μm
Displacement of bearing	[u _y]	0.000	μm
Displacement of bearing	[u _z]	16.250	μm
Displacement of bearing	[u _r]	16.250	μm (90°)
Misalignment of bearing	[r _x]	-2.498	mrad (-8.59')
Misalignment of bearing	[r _y]	-0.000	mrad (0')
Misalignment of bearing	[r _z]	0.000	mrad (0')
Misalignment of bearing	[r _r]	2.498	mrad (8.59')

Damage (%) [H] (20000.000)

No.	B1	B2
	19999.99	9999.99

Σ9999.99 9999.99

Utilization (%) [H] (20000.000)

	B1	B2
	430.624	30.62

B1: Rolling bearing

B2: Rolling bearing

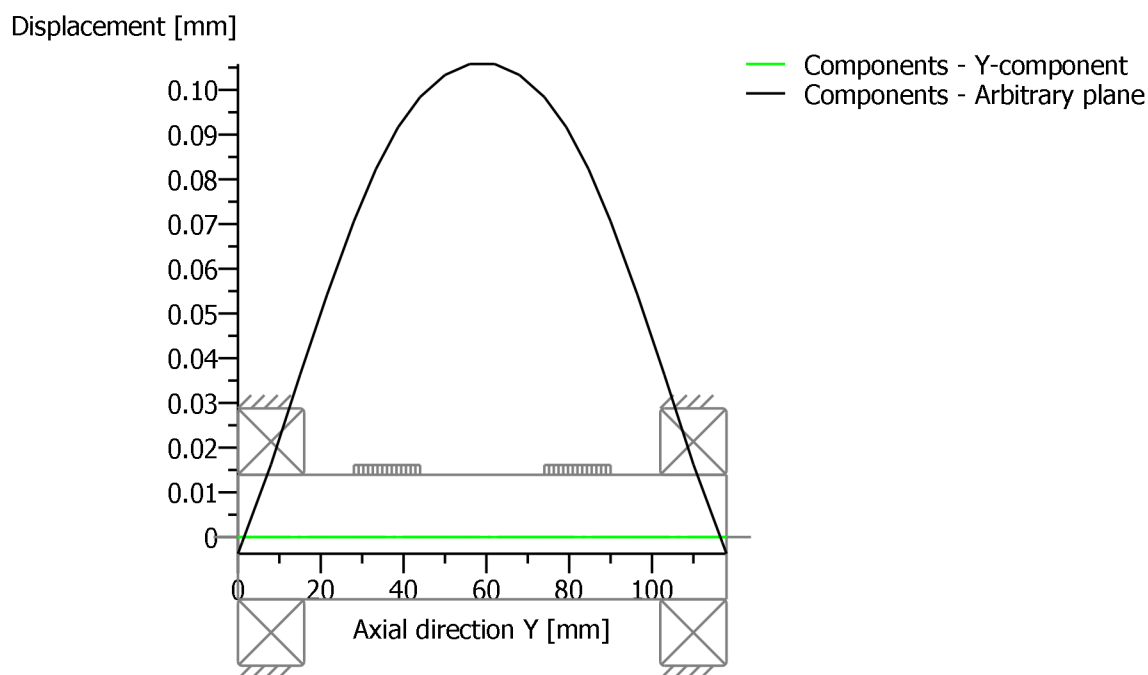
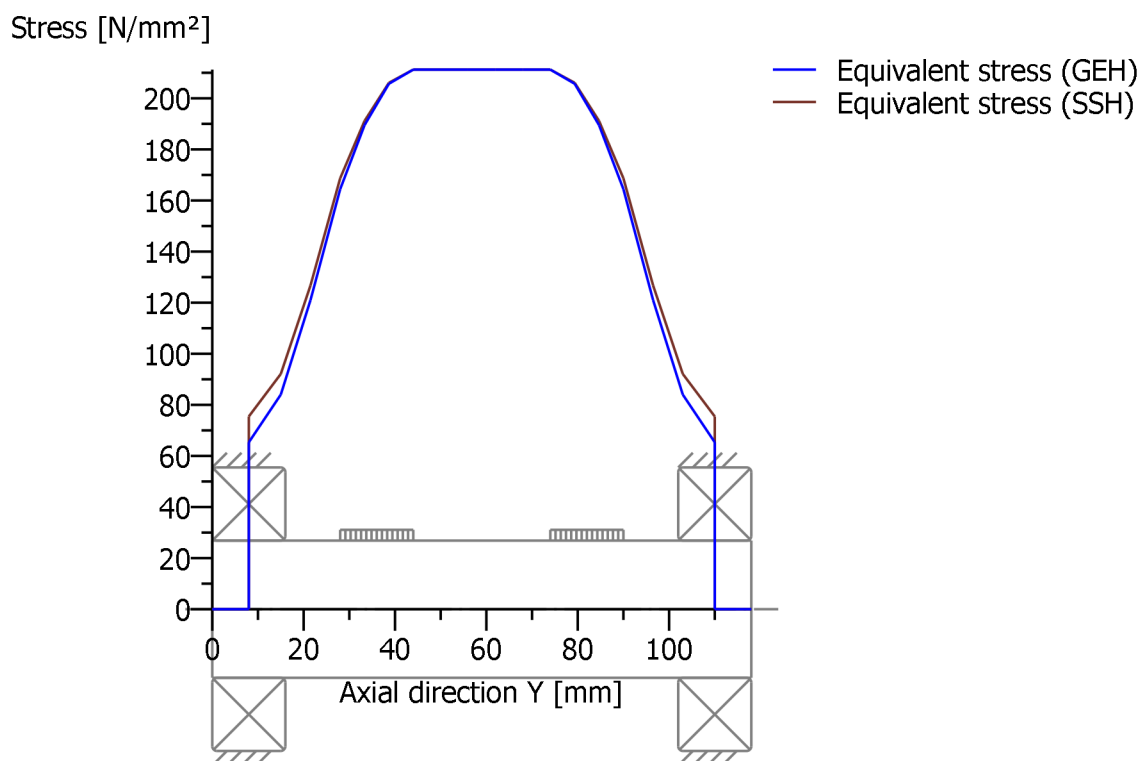


Figure: Deformation (bending etc.) (Arbitrary plane 90 120)



Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\sigma_V = ((\sigma_B + \sigma_{Z,D})^2 + 3 \cdot (\tau_T + \tau_S)^2)^{1/2}$

SSH(Tresca): $\sigma_V = ((\sigma_B - \sigma_{Z,D})^2 + 4 \cdot (\tau_T + \tau_S)^2)^{1/2}$

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Edition, 2012)

Summary

Shaft 1

Material	34 CrNiMo 6 (2)
Material type	Through hardened steel
Material treatment	flame/ind. hardened
Surface treatment	No

Calculation of endurance limit and the static strength

Rolled steel, through hardening steel

Overload case F2 (chapter 2.4.2): Constant stress ratio

Safety number according Chapter 2.5	[jF]	1.20
Safety number according Chapter 1.5	[jm]	1.60
Safety number according Chapter 1.5	[jp]	1.20
Safety number according Chapter 1.5	[jmt]	1.20
Safety number according Chapter 1.5	[jpt]	1.00

Safety number according Chapter 1.5	[jG]	1.00
-------------------------------------	------	------

Cross section	Pos (Y co-ord) (mm)	
A-A	44.00	Interference fit
B-B	74.00	Interference fit

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
A-A	2.80	1.00	0.97	1.24	2.81	3.45
B-B	2.80	1.00	0.97	1.24	2.81	3.45

Nominal safety:	1.20	1.20	1.60
-----------------	------	------	------

Abbreviations:

Kfb: Notch factor bending

KRs: Surface factor

ALGmax: Highest utilization

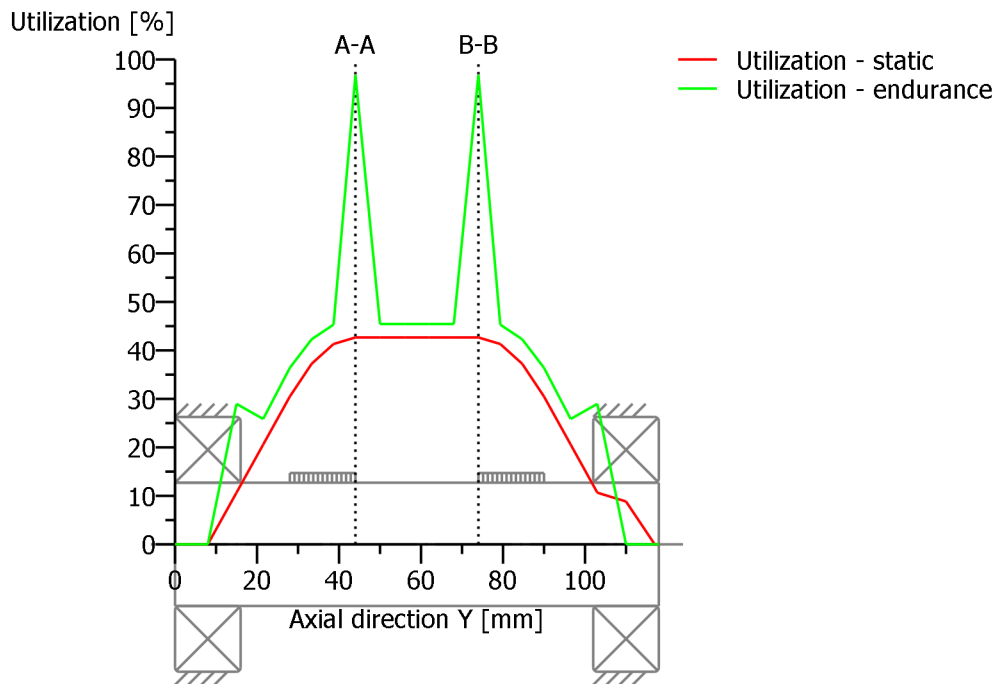
SD: Safety endurance limit

SS: Safety against yield point

SB: Safety against tensile stress

Utilization (%) [Smin/S]

Cross section	static	endurance
A-A	42.669	96.762
B-B	42.669	96.762
Maximum utilization (%)	[A]	96.762



Utilization = S_{min}/S (%)

Figure: Strength

Calculation details

General statements

Label	Shaft 1		
Drawing			
Length (mm)	[l]	118.00	
Speed (1/min)	[n]	1500.00	

Material	34 CrNiMo 6 (2)
Material type	Through hardened steel
Material treatment	flame/ind. hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.000	1.700	1.000	1.000
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, through hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm ²)	[R _m ,N]	1200.00
Yield point (N/mm ²)	[R _p ,N]	1000.00
Fatigue limit (N/mm ²)	[σ _{zd} WN]	540.00
Fatigue limit (N/mm ²)	[σ _b WN]	570.00

Fatigue limit (N/mm ²)	[τ_t WN]	340.00		
Fatigue limit (N/mm ²)	[τ_s WN]	310.00		
Breaking elongation (%)	[A]	9.00		
Reference diameter (mm)	[deffNm, deffNp]	16.00	16.00	
Required life time	[H]	20000.00		
Number of load cycles	[NL]	1800000000		
Endurance limit for single stage use				
Temperature (°C)	[Temperatur]	20.000		
Temperature duration (h)	[TemperaturD]	10000.000		
Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[σ_Z , σ_D , f_r , R_{pmax}]	1.000	1.000	0.577	1150.0
[fWt, fWs]	0.577	0.450		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsigs, RmNmin]	0.500	2700.0	0.220	400.0
[MS, MT]	0.2805	0.1619		
[$k\sigma$, $k\tau$]	15	25		
[kD σ , kD τ]	0	0		
[ND σ , ND τ]	1e+006	1e+006		
[ND σ II, ND τ II]	0	0		

Thickness of raw material (mm) [d.eff] 35.00

Material data calculated acc. FKM directive with Kdm, Kdp

Geometric size factors (Kdm, Kdp) calculated from raw diameter

Material strength calculated from size of raw material

Constants	[adm, adp]	0.330	0.390
Size factors	[Kdm, Kdp]	0.906	0.887
Tensile strength (N/mm ²)	[Rm]	1087.11	
Yield point (N/mm ²)	[Rp]	886.95	
σ_{zdW} (N/mm ²)	[σ_{zdW}]	489.20	
σ_{bW} (N/mm ²)	[σ_{bW}]	516.38	
τ_tW (N/mm ²)	[τ_tW]	308.01	
τ_sW (N/mm ²)	[τ_sW]	280.84	

Overload case F2 (chapter 2.4.2): Constant stress ratio

Safety number according Chapter 1.5	[jm]	1.60
Safety number according Chapter 1.5	[jp]	1.20
Safety number according Chapter 1.5	[jmt]	1.20
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 2.5	[jF]	1.20
Safety number according Chapter 1.5	[jG]	1.00

Cross section 'A-A' Interference fit

Comment

Position (Y-Coordinate) (mm)	[y]	44.000
External diameter (mm)	[da]	30.000
Inner diameter (mm)	[di]	0.000

Notch effect	Interference fit				
Characteristics:	Firm interference fit				
Mean roughness (μm)	[Rz]	2.800			
	Tension/Compression Bending Torsion Shearing				
Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]	0.0	0.0	0.0	0.8	
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	559.9	0.0	0.0	
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]		0.0	951.9	0.0	0.8
Cross section, moment of resistance: (mm²)					
[A, Wb, Wt, A]	706.9	2650.7	5301.4	706.9	

Stresses: (N/mm^2)				
[σ_m , σ_b , τ_m , τ_s]	0.000	0.000	0.000	0.002
[σ_a , σ_b , τ_a , τ_s]	0.000	211.240	0.000	0.000
[σ_{max} , $\sigma_{b\text{max}}$, τ_{max} , $\tau_{s\text{max}}$]	0.000	359.108	0.000	0.002

FATIGUE PROOF:		
Total safety factor according chapter 2.5.3	[jD]	1.200
(Formula: $jD = jF \cdot jG / KTD$)		

		Tension/Compression Bending Torsion Shearing			
Notch effect coefficient	[β (dB)]	2.900	2.900	1.855	1.428
[dB] (mm) 40.0, [rB] (mm) 2.4, [r] (mm) 1.8					
Support number	[n(r)]	1.142	1.142	1.138	1.138
Support number	[n(rB)]	1.132	1.132	1.120	1.120
Support number	[n(d)]	1.026	1.026	1.039	1.039
Mechanical material support factor	[nwm]	1.030	1.030	1.030	1.030
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.875	2.801	1.756	1.404
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Coefficient of surface strengthening	[KV]	1.500	1.500	1.500	1.200
Design coefficient	[KWK]	1.917	1.867	1.171	1.170
Fatigue limit of part (N/mm^2)	[SWK]	255.239	261.972	239.853	239.979

Calculation with principal mean stress:					
Mean stress coefficient	[KAK]	0.667	1.000	0.781	0.781
Permissible amplitude (N/mm^2)	[SAK]	170.221	261.971	187.245	187.343
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm^2)	[SBK]	170.221	261.971	187.245	187.343
Rate of utilization	[aBK]	0.000	0.968	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm^2)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.968

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm^2)	[SmV_2]	0.003
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.968
---------------------	----------	-------

Safety endurance limit assessment	[S.Dauer]	1.240
Required safety	[jD]	1.200
Result (%)	[S/jD]	103.3

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3	[jges]	1.305
(Formula: $jges = jG \cdot \max(jm/KTm \cdot Rp/Rm, jp/KTp, jmt/KTm \cdot Rp/Rm, jpt/KTtp)$)		

		Tension/Compression Bending Torsion Shearing			
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.1387	1.1387	1.0000
Strength of part (N/mm ²)	[SSK]	886.95	1009.95	583.09	512.08
Rate of utilization	[aSK]	0.000	0.464	0.000	0.000

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm ²)	[SvBn]	359.108
Rate of utilization	[aSKvBn]	0.464

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm ²)	[SvQn]	0.003
Rate of utilization	[aSKvQn]	0.000

Highest utilization	[aSKmax]	0.464
---------------------	----------	-------

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	3.447
Required safety	[jm/Ktm]	1.600
Result (%)	[S/jm]	215.4

Safety against yield point	[S.Rp]	2.812
Required safety	[jp/KTp]	1.200
Result (%)	[S/jp]	234.4

Cross section 'B-B' Interference fit

Comment		
Position (Y-Coordinate) (mm)	[y]	74.000
External diameter (mm)	[da]	30.000
Inner diameter (mm)	[di]	0.000
Notch effect		Interference fit
Characteristics:	Firm interference fit	
Mean roughness (µm)	[Rz]	2.800

		Tension/Compression Bending Torsion Shearing			
Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]		0.0	0.0	0.0	0.8
Amplitude [Fzda, Mba, Ta, Fqa]		0.0	559.9	0.0	0.0
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]			0.0	951.9	0.0
Cross section, moment of resistance: (mm ²)					0.8
[A, Wb, Wt, A]		706.9	2650.7	5301.4	706.9

Stresses: (N/mm ²)					
[σmz, σmb, τmt, τms]		0.000	0.000	0.000	0.002
[σaz, σab, τat, τas]		0.000	211.240	0.000	0.000

[σ_{zmax}, σ_{bmax}, τ_{tmax}, τ_{smax}] 0.000 359.108 0.000 0.002

FATIGUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.200

(Formula: $jD = jF \cdot jG / KTD$)

		Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient	[β(dB)]	2.900	2.900	1.855	1.428
[dB] (mm) 40.0,	[rB] (mm) 2.4,	[r] (mm) 1.8			
Support number	[n(r)]	1.142	1.142	1.138	1.138
Support number	[n(rB)]	1.132	1.132	1.120	1.120
Support number	[n(d)]	1.026	1.026	1.039	1.039
Mechanical material support factor	[nwm]	1.030	1.030	1.030	1.030
The support factor is determined with the support factor as defined by Stielers.					
Notch effect coefficient beta	[Kf]	2.875	2.801	1.756	1.404
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Coefficient of surface strengthening	[KV]	1.500	1.500	1.500	1.200
Design coefficient	[KWK]	1.917	1.867	1.171	1.170
Fatigue limit of part (N/mm ²)	[SWK]	255.239	261.972	239.853	239.979

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.667	1.000	0.781	0.781
Permissible amplitude (N/mm ²)	[SAK]	170.221	261.971	187.245	187.343
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm ²)	[SBK]	170.221	261.971	187.245	187.343
Rate of utilization	[aBK]	0.000	0.968	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.968

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_2]	0.003
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.968
Safety endurance limit assessment	[S.Dauer]	1.240
Required safety	[jD]	1.200
Result (%)	[S/jD]	103.3

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.305

(Formula: $jges = jG \cdot \text{Max}(j_m / K T m \cdot R_p / R_m, j_p / K T p, j_{mT} / K T m \cdot R_p / R_m, j_{pT} / K T p)$)

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]	1.700	1.330		
Plastic support number	[npl]	1.0000	1.1387	1.1387	1.0000
Strength of part (N/mm ²)	[SSK]	886.95	1009.95	583.09	512.08
Rate of utilization	[aSK]	0.000	0.464	0.000	0.000

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm ²)	[SvBn]	359.108
--	--------	---------

Rate of utilization	[aSKvBn]	0.464
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm ²)	[SvQn]	0.003
Rate of utilization	[aSKvQn]	0.000
Highest utilization	[aSKmax]	0.464
Safety for fracture and yield stresses:		
Safety against fracture	[S.Rm]	3.447
Required safety	[jm/Ktm]	1.600
Result (%)	[S/jm]	215.4
Safety against yield point	[S.Rp]	2.812
Required safety	[jp/KTp]	1.200
Result (%)	[S/jp]	234.4

Important remarks concerning strength calculation according to

FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization ≤ 1
- Currently the following restrictions still apply::
Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
 $\beta_S = 1.0 + (\beta_T - 1.0) / 2.0$ (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with $p = 20\text{MPa}$
- Proven safety: Effective safety according to special formula,
condition: safety > required safety or result > 100%

End of Report

lines: 578

E.1.2 KISSsoft Report - Shaft with hub mounted with slight interference fit

Name : veio30_reduzido

Changed by: up201106920

on: 11.06.2017

at: 17:21:32

Important hint: At least one warning has occurred during the calculation:

1-> The required service life of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' is not achieved!
The static safety is low (in range 0.5 - 2.0).

Please check whether these values are acceptable or not.

2-> The required service life of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' is not achieved!
The static safety is low (in range 0.5 - 2.0).

Please check whether these values are acceptable or not.

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	Shaft 1
Drawing	
Initial position (mm)	0.000
Length (mm)	124.000
Speed (1/min)	1500.00
Sense of rotation: clockwise	
Material	34 CrNiMo 6 (2)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	0.686
(Notice: Weight stands for the shaft only without considering the gears)	
Weight of shaft, including additional masses (kg)	0.686
Mass moment of inertia (kg*mm ²)	77.209
Momentum of mass GD ² (Nm ²)	0.003
Position in space (°)	0.000
Gears mounted with stiffness according to ISO	
Consider deformations due to shearing	
Shear correction coefficient	1.100
Contact angle of rolling bearings is considered	
Tolerance field: Mean value	
Housing material	C45 (1)
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature of housing (°C)	80.000

Thermal housing reference point (mm) 0.000
Reference temperature (°C) 20.000

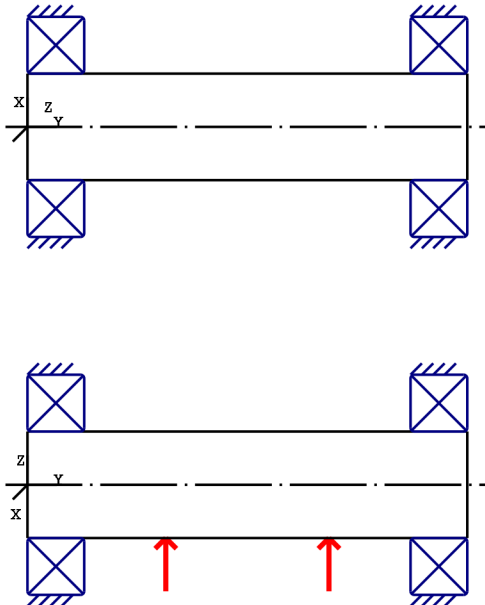


Figure: Load applications

Shaft definition (Shaft 1)

Outer contour

Cylinder (veio)		0.000mm ... 124.000mm
Diameter (mm)	[d]	30.0000
Length (mm)	[l]	124.0000
Surface roughness (µm)	[Rz]	1.8000

Interference fit (Interference fit) 47.000mm ... 77.000mm
l=30.00 (mm), Typ=0, Turned (Ra=3.2µm/125µin)

Forces

Type of force element	Centric force	
Label in the model	carga1	
Position on shaft (mm)	[y _{local}]	39.0000
Position in global system (mm)	[y _{global}]	39.0000
Length of load application (mm)		16.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000

Shearing force X (N)	0.0000
Shearing force Z (N)	20000.0000
Bending moment X (Nm)	0.0000
Bending moment Z (Nm)	0.0000

Type of force element		Centric force
Label in the model		carga2
Position on shaft (mm)	[ylocal]	85.0000
Position in global system (mm)	[yglobal]	85.0000
Length of load application (mm)		16.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		20000.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

Bearing

Label in the model		Rolling bearing
Bearing type		SKF *NJ 206 ECJ
Bearing type		Cylindrical roller bearing (single row)
Bearing position (mm)	[ylocal]	8.000
Bearing position (mm)	[yglobal]	8.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	62.000
Width (mm)	[b]	16.000
Corner radius (mm)	[r]	1.000
Basic static load rating	[C ₀]	36.500
Basic dynamic load rating	[C]	44.000
Fatigue load rating	[C _U]	4.600
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Label in the model		Rolling bearing
Bearing type		SKF *NJ 206 ECJ
Bearing type		Cylindrical roller bearing (single row)
Bearing position (mm)	[ylocal]	116.000
Bearing position (mm)	[yglobal]	116.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	62.000
Width (mm)	[b]	16.000
Corner radius (mm)	[r]	1.000
Basic static load rating	[C ₀]	36.500
Basic dynamic load rating	[C]	44.000
Fatigue load rating	[C _U]	4.600
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Results

Shaft

Maximum deflection (μm)	125.119
Position of the maximum (mm)	59.000
Mass center of gravity (mm)	62.000
Total axial load (N)	0.000
Torsion under torque ($^{\circ}$)	0.000

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	[u_A]	10.00	μm
Rolling bearings, classical calculation (contact angle considered)			

Shaft 'Shaft 1' Rolling bearing 'Rolling bearing'

Position (Y-coordinate)	[y]	8.00	mm
Equivalent load	[P]	20.00	kN
Equivalent load	[P_0]	20.00	kN
Life modification factor for reliability [a_1]		1.000	
Nominal bearing service life	[L_{nh}]	153.96	h
Static safety factor	[S_0]	1.83	
Bearing reaction force	[Fx]	0.000	kN
Bearing reaction force	[Fy]	0.000	kN
Bearing reaction force	[Fz]	-19.997	kN
Bearing reaction force	[Fr]	19.997	kN (-90°)
Oil level	[H]	0.000	mm
Rolling moment of friction	[M_{rr}]	0.103	Nm
Sliding moment of friction	[M_{sl}]	0.056	Nm
Moment of friction, seals	[M_{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 10000/1 EN:2013			
Moment of friction flow losses	[M_{drag}]	0.000	Nm
Torque of friction	[M_{loss}]	0.159	Nm
Power loss	[P_{loss}]	24.994	W

The moment of friction is calculated according to the details in SKF Catalog 2013.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

Displacement of bearing	[u_x]	0.000	μm
Displacement of bearing	[u_y]	0.000	μm
Displacement of bearing	[u_z]	16.250	μm
Displacement of bearing	[u_r]	16.250	μm (90°)
Misalignment of bearing	[r_x]	2.882	mrاد (9.91')
Misalignment of bearing	[r_y]	0.000	mrاد (0')
Misalignment of bearing	[r_z]	-0.000	mrاد (0')
Misalignment of bearing	[r_l]	2.882	mrاد (9.91')

Shaft 'Shaft 1' Rolling bearing 'Rolling bearing'

Position (Y-coordinate)	[y]	116.00	mm
Equivalent load	[P]	20.00	kN
Equivalent load	[P_0]	20.00	kN
Life modification factor for reliability [a_1]		1.000	

Nominal bearing service life	[L _{nh}]	153.96	h
Static safety factor	[S ₀]	1.83	
Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	-19.997	kN
Bearing reaction force	[F _r]	19.997	kN (-90°)
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _{rr}]	0.103	Nm
Sliding moment of friction	[M _{sl}]	0.056	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 10000/1 EN:2013			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.159	Nm
Power loss	[P _{loss}]	24.994	W

The moment of friction is calculated according to the details in SKF Catalog 2013.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

Displacement of bearing	[u _x]	0.000	μm
Displacement of bearing	[u _y]	0.000	μm
Displacement of bearing	[u _z]	16.250	μm
Displacement of bearing	[u _r]	16.250	μm (90°)
Misalignment of bearing	[r _x]	-2.882	mrad (-9.91')
Misalignment of bearing	[r _y]	-0.000	mrad (0')
Misalignment of bearing	[r _z]	-0.000	mrad (0')
Misalignment of bearing	[r _r]	2.882	mrad (9.91')

Damage (%) [H] (20000.000)

No.	B1	B2
	19999.99	9999.99

Σ9999.99 9999.99

Utilization (%) [H] (20000.000)

	B1	B2
	430.61	430.61

B1: Rolling bearing

B2: Rolling bearing

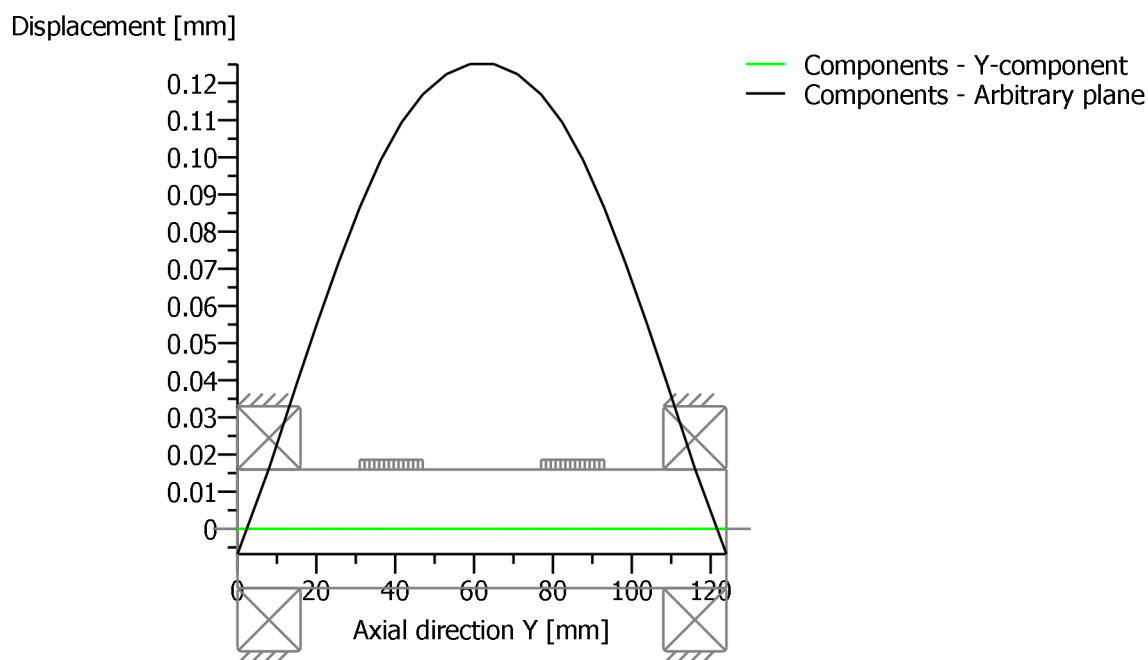
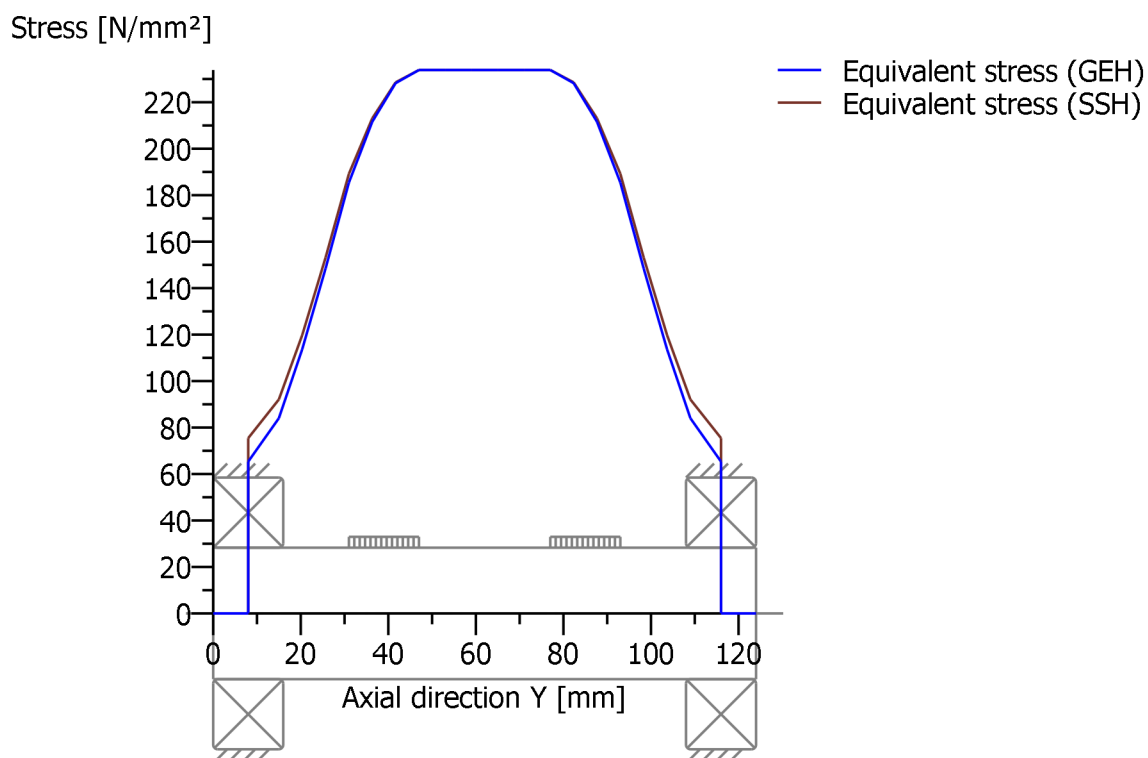


Figure: Deformation (bending etc.) (Arbitrary plane 90 120)



Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\sigma_V = ((\sigma_B + \sigma_Z, D)^2 + 3 \cdot (\tau_T + \tau_S)^2)^{1/2}$

SSH(Tresca): $\sigma_V = ((\sigma_B - \sigma_Z, D)^2 + 4 \cdot (\tau_T + \tau_S)^2)^{1/2}$

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Edition, 2012)

Summary

Shaft 1

Material	34 CrNiMo 6 (2)
Material type	Through hardened steel
Material treatment	flame/ind. hardened
Surface treatment	No

Calculation of endurance limit and the static strength

Rolled steel, through hardening steel

Overload case F2 (chapter 2.4.2): Constant stress ratio

Safety number according Chapter 2.5	[jF]	1.20
Safety number according Chapter 1.5	[jm]	1.60
Safety number according Chapter 1.5	[jp]	1.20
Safety number according Chapter 1.5	[jmt]	1.20
Safety number according Chapter 1.5	[jpt]	1.00

Safety number according Chapter 1.5	[jG]	1.00
-------------------------------------	------	------

Cross section	Pos (Y co-ord) (mm)	
Limited cross section	47.00	Interference fit
Limited cross section	77.00	Interference fit

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
Limited cross section	2.57	1.00	0.98	1.22	2.54	3.11
Limited cross section	2.57	1.00	0.98	1.22	2.54	3.11

Nominal safety:	1.20	1.20	1.60
-----------------	------	------	------

Abbreviations:

Kfb: Notch factor bending

KRs: Surface factor

ALGmax: Highest utilization

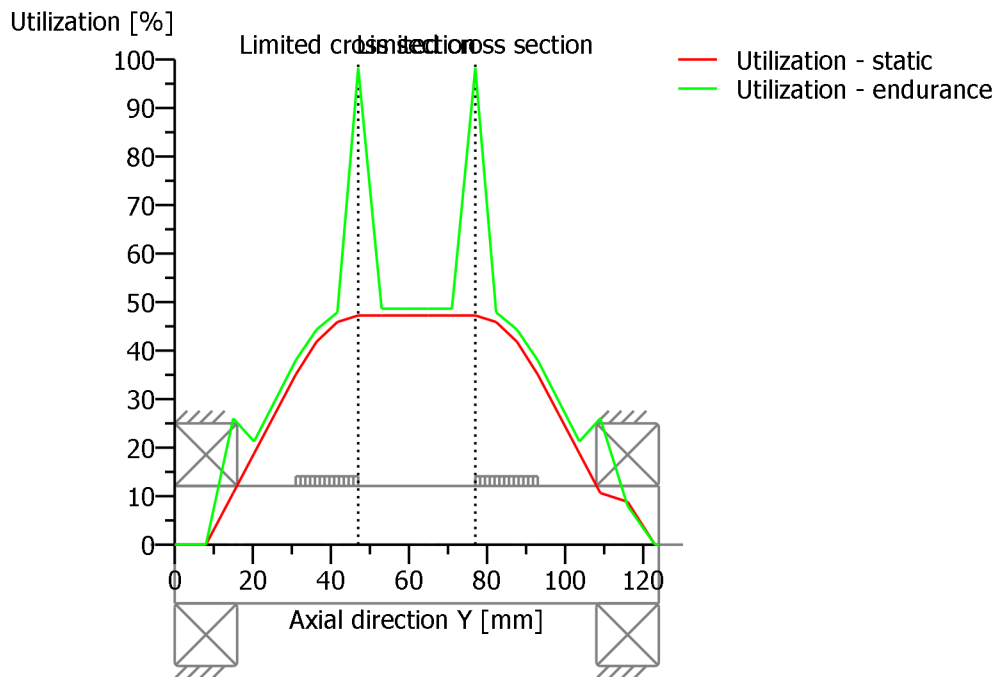
SD: Safety endurance limit

SS: Safety against yield point

SB: Safety against tensile stress

Utilization (%) [Smin/S]

Cross section	static	endurance
Limited cross section	47.240	98.219
Limited cross section	47.240	98.219
Maximum utilization (%)	[A]	98.219



Utilization = S_{min}/S (%)

Figure: Strength

Calculation details

General statements

Label	Shaft 1		
Drawing			
Length (mm)	[l]	124.00	
Speed (1/min)	[n]	1500.00	

Material	34 CrNiMo 6 (2)
Material type	Through hardened steel
Material treatment	flame/ind. hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.000	1.700	1.000	1.000
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, through hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm ²)	[R _m ,N]	1200.00
Yield point (N/mm ²)	[R _p ,N]	1000.00
Fatigue limit (N/mm ²)	[σ _d WN]	540.00
Fatigue limit (N/mm ²)	[σ _b WN]	570.00

Fatigue limit (N/mm ²)	[τ_t WN]	340.00		
Fatigue limit (N/mm ²)	[τ_s WN]	310.00		
Breaking elongation (%)	[A]	9.00		
Reference diameter (mm)	[deffNm, deffNp]	16.00	16.00	
Required life time	[H]	20000.00		
Number of load cycles	[NL]	1800000000		
Endurance limit for single stage use				
Temperature (°C)	[Temperatur]	20.000		
Temperature duration (h)	[TemperaturD]	10000.000		
Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[σ_Z , σ_D , f_r , R_{pmax}]	1.000	1.000	0.577	1150.0
[fWt, fWs]	0.577	0.450		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsigs, RmNmin]	0.500	2700.0	0.220	400.0
[MS, MT]	0.2805	0.1619		
[$k\sigma$, $k\tau$]	15	25		
[kD σ , kD τ]	0	0		
[ND σ , ND τ]	1e+006	1e+006		
[ND σ II, ND τ II]	0	0		

Thickness of raw material (mm) [d.eff] 35.00

Material data calculated acc. FKM directive with Kdm, Kdp

Geometric size factors (Kdm, Kdp) calculated from raw diameter

Material strength calculated from size of raw material

Constants	[adm, adp]	0.330	0.390
Size factors	[Kdm, Kdp]	0.906	0.887
Tensile strength (N/mm ²)	[Rm]	1087.11	
Yield point (N/mm ²)	[Rp]	886.95	
σ_{zdW} (N/mm ²)	[σ_{zdW}]	489.20	
σ_{bW} (N/mm ²)	[σ_{bW}]	516.38	
τ_tW (N/mm ²)	[τ_tW]	308.01	
τ_sW (N/mm ²)	[τ_sW]	280.84	

Overload case F2 (chapter 2.4.2): Constant stress ratio

Safety number according Chapter 1.5	[jm]	1.60
Safety number according Chapter 1.5	[jp]	1.20
Safety number according Chapter 1.5	[jmt]	1.20
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 2.5	[jF]	1.20
Safety number according Chapter 1.5	[jG]	1.00

Cross section 'Limited cross section'

Interference fit

Comment		
Position (Y-Coordinate) (mm)	[y]	47.000
External diameter (mm)	[da]	30.000
Inner diameter (mm)	[di]	0.000

Notch effect		Interference fit			
Characteristics:		Slight interference fit			
Mean roughness (μm)	[Rz]	1.800			
		Tension/Compression Bending Torsion Shearing			
Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]		0.0	0.0	0.0	0.4
Amplitude [Fzda, Mba, Ta, Fqa]		0.0	619.9	0.0	0.4
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]		0.0	1053.9	0.0	0.8
Cross section, moment of resistance: (mm^2)					
[A, Wb, Wt, A]		706.9	2650.7	5301.4	706.9

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	0.000	0.000	0.000	0.001
[σaz, σab, τat, τas]	0.000	233.872	0.000	0.001
[σzmax, σbmax, τtmax, τsmax]	0.000	397.582	0.000	0.002

FATIGUE PROOF:

Total safety factor according chapter 2.5.3 (Formula: $jD = jF \cdot jG / KTD$)	[jD]	1.200
---	------	-------

				Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient			[β(dB)]	2.636	2.636	1.736	1.368
[dB] (mm)	30.0,	[rB] (mm)	1.8,				
			[r] (mm)	1.8			
Support number			[n(r)]	1.142	1.142	1.138	1.138
Support number			[n(rB)]	1.142	1.142	1.138	1.138
Support number			[n(d)]	1.026	1.026	1.039	1.039
Mechanical material support factor			[nwm]	1.030	1.030	1.030	1.030
The support factor is determined with the support factor as defined by Stieler.							
Notch effect coefficient beta			[Kf]	2.636	2.568	1.671	1.368
Roughness factor			[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient							
Coefficient of surface strengthening			[KV]	1.500	1.500	1.500	1.200
Design coefficient			[KWK]	1.757	1.712	1.114	1.140
Fatigue limit of part (N/mm²)			[SWK]	278.391	285.735	252.111	246.336

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.667	1.000	0.781	0.861
Permissible amplitude (N/mm ²)	[SAK]	185.661	285.735	196.814	212.004
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm ²)	[SBK]	185.661	285.735	196.814	212.004
Rate of utilization	[aBK]	0.000	0.982	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.982

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_2]	0.001
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.982
---------------------	----------	-------

Safety endurance limit assessment	[S.Dauer]	1.222
Required safety	[jD]	1.200
Result (%)	[S/jD]	101.8

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3	[jges]	1.305
(Formula: $jges = jG \cdot \max(jm/KTm \cdot Rp/Rm, jp/KTp, jmt/KTm \cdot Rp/Rm, jpt/KTtp)$)		

		Tension/Compression Bending Torsion Shearing			
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.1387	1.1387	1.0000
Strength of part (N/mm ²)	[SSK]	886.95	1009.95	583.09	512.08
Rate of utilization	[aSK]	0.000	0.514	0.000	0.000

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm ²)	[SvBn]	397.582
Rate of utilization	[aSKvBn]	0.514

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm ²)	[SvQn]	0.003
Rate of utilization	[aSKvQn]	0.000

Highest utilization	[aSKmax]	0.514
---------------------	----------	-------

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	3.113
Required safety	[jm/Ktm]	1.600
Result (%)	[S/jm]	194.6

Safety against yield point	[S.Rp]	2.540
Required safety	[jp/KTp]	1.200
Result (%)	[S/jp]	211.7

Cross section 'Limited cross section'

Interference fit

Comment		
Position (Y-Coordinate) (mm)	[y]	77.000
External diameter (mm)	[da]	30.000
Inner diameter (mm)	[di]	0.000
Notch effect		Interference fit
Characteristics:	Slight interference fit	
Mean roughness (μm)	[Rz]	1.800

		Tension/Compression Bending Torsion Shearing			
Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]		0.0	0.0	0.0	0.4
Amplitude [Fzda, Mba, Ta, Fqa]		0.0	619.9	0.0	0.4
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]			0.0	1053.9	0.0
Cross section, moment of resistance: (mm ²)					0.8
[A, Wb, Wt, A]		706.9	2650.7	5301.4	706.9

Stresses: (N/mm ²)					
[σmz, σmb, τmt, τms]		0.000	0.000	0.000	0.001
[σaz, σab, τat, τas]		0.000	233.872	0.000	0.001

[σ_{zmax}, σ_{bmax}, τ_{tmax}, τ_{smax}] 0.000 397.582 0.000 0.002

FATIGUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.200

(Formula: $jD = jF \cdot jG / KTD$)

		Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient	[β(dB)]	2.636	2.636	1.736	1.368
[dB] (mm) 30.0,	[rB] (mm) 1.8,				
	[r] (mm) 1.8				
Support number	[n(r)]	1.142	1.142	1.138	1.138
Support number	[n(rB)]	1.142	1.142	1.138	1.138
Support number	[n(d)]	1.026	1.026	1.039	1.039
Mechanical material support factor	[nwm]	1.030	1.030	1.030	1.030
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.636	2.568	1.671	1.368
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Coefficient of surface strengthening	[KV]	1.500	1.500	1.500	1.200
Design coefficient	[KWK]	1.757	1.712	1.114	1.140
Fatigue limit of part (N/mm ²)	[SWK]	278.391	285.735	252.111	246.336

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.667	1.000	0.781	0.861
Permissible amplitude (N/mm ²)	[SAK]	185.661	285.735	196.814	212.004
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm ²)	[SBK]	185.661	285.735	196.814	212.004
Rate of utilization	[aBK]	0.000	0.982	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.982

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_2]	0.001
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.982
Safety endurance limit assessment	[S.Dauer]	1.222
Required safety	[jD]	1.200
Result (%)	[S/jD]	101.8

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.305

(Formula: $jges = jG \cdot \text{Max}(j_m / K T m \cdot R_p / R_m, j_p / K T p, j_{mT} / K T m \cdot R_p / R_m, j_{pT} / K T p)$)

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]	1.700	1.330		
Plastic support number	[npl]	1.0000	1.1387	1.0000	
Strength of part (N/mm ²)	[SSK]	886.95	1009.95	583.09	512.08
Rate of utilization	[aSK]	0.000	0.514	0.000	0.000

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm ²)	[SvBn]	397.582
--	--------	---------

Rate of utilization	[aSKvBn]	0.514
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm ²)	[SvQn]	0.003
Rate of utilization	[aSKvQn]	0.000
Highest utilization	[aSKmax]	0.514
Safety for fracture and yield stresses:		
Safety against fracture	[S.Rm]	3.113
Required safety	[jm/Ktm]	1.600
Result (%)	[S/jm]	194.6
Safety against yield point	[S.Rp]	2.540
Required safety	[jp/KTp]	1.200
Result (%)	[S/jp]	211.7

Important remarks concerning strength calculation according to FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization ≤ 1
- Currently the following restrictions still apply::
Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
 $\beta_S = 1.0 + (\beta_T - 1.0) / 2.0$ (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with $p = 20\text{MPa}$
- Proven safety: Effective safety according to special formula,
condition: safety > required safety or result > 100%

End of Report

lines: 578

E.2 KISSsoft Reports - Alternative Shaft

Name : veio_ressalto_novo

Changed by: up201106920

on: 21.06.2017

at: 21:11:15

Important hint: At least one warning has occurred during the calculation:

1-> The required service life of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' is not achieved!
The static safety is low (in range 0.5 - 2.0).

Please check whether these values are acceptable or not.

2-> The thermally safe operating speed of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' could be critical.
Please run the calculation for the 'Thermally safe operating speed'
(in the special tab of menu 'Calculation').

3-> The required service life of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' is not achieved!
The static safety is low (in range 0.5 - 2.0).

Please check whether these values are acceptable or not.

4-> The thermally safe operating speed of bearing 'Shaft 'Shaft 1', Rolling bearing 'Rolling bearing' could be critical.
Please run the calculation for the 'Thermally safe operating speed'
(in the special tab of menu 'Calculation').

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	Shaft 1
Drawing	
Initial position (mm)	0.000
Length (mm)	118.000
Speed (1/min)	1500.00
Sense of rotation: clockwise	
Material	34 CrNiMo 6 (2)
Young's modulus (N/mm ²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m ³)	7830.000
Coefficient of thermal expansion (10 ⁻⁶ /K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	0.790
(Notice: Weight stands for the shaft only without considering the gears)	
Weight of shaft, including additional masses (kg)	0.790
Mass moment of inertia (kg*mm ²)	117.223
Momentum of mass GD ² (Nm ²)	0.005
Position in space (°)	0.000

Gears mounted with stiffness according to ISO
Consider deformations due to shearing
Shear correction coefficient 1.100
Contact angle of rolling bearings is considered
Tolerance field: Mean value
Reference temperature (°C) 20.000

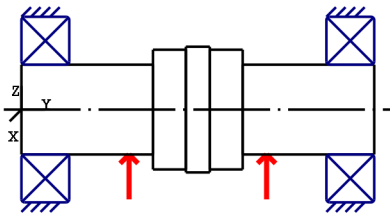
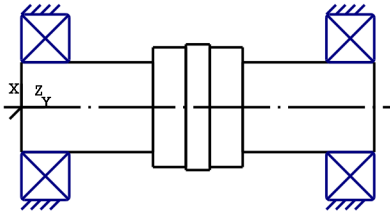


Figure: Load applications

Shaft definition (Shaft 1)

Outer contour

Cylinder (Cylinder)			0.000mm ...	44.000mm
Diameter (mm)	[d]	30.0000		
Length (mm)	[l]	44.0000		
Surface roughness (µm)	[Rz]	2.8000		

Relief groove right (Relief groove right)

$r=1.20$ (mm), $t=0.20$ (mm), $l=2.50$ (mm), $Rz=2.8$, Turned ($Ra=3.2\mu\text{m}/125\mu\text{in}$)

Form E (DIN 509), Series 1, with raised fatigue limit

Cylinder (Cylinder)			44.000mm ...	55.000mm
Diameter (mm)	[d]	40.0000		
Length (mm)	[l]	11.0000		
Surface roughness (µm)	[Rz]	8.0000		

Radius right (Radius right)

r=2.00 (mm), Rz=2.8, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder)		55.000mm ...	63.000mm
Diameter (mm)	[d]	42.0000	
Length (mm)	[l]	8.0000	
Surface roughness (µm)	[Rz]	2.8000	

Cylinder (Cylinder)		63.000mm ...	74.000mm
Diameter (mm)	[d]	40.0000	
Length (mm)	[l]	11.0000	
Surface roughness (µm)	[Rz]	8.0000	

Radius left (Radius left)

r=2.00 (mm), Rz=2.8, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder)		74.000mm ...	118.000mm
Diameter (mm)	[d]	30.0000	
Length (mm)	[l]	44.0000	
Surface roughness (µm)	[Rz]	2.8000	

Relief groove left (Relief groove left)

r=1.20 (mm), t=0.20 (mm), l=2.50 (mm), Rz=2.8, Turned (Ra=3.2µm/125µin)

Form E (DIN 509), Series 1, with raised fatigue limit

Forces

Type of force element	Centric force	
Label in the model	Centrical load	
Position on shaft (mm)	[y _{local}]	36.0000
Position in global system (mm)	[y _{global}]	36.0000
Length of load application (mm)		15.9990
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		20000.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

Type of force element	Centric force	
Label in the model	Centrical load	
Position on shaft (mm)	[y _{local}]	82.0000
Position in global system (mm)	[y _{global}]	82.0000
Length of load application (mm)		15.9990
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		20000.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

Bearing

Label in the model		Rolling bearing
Bearing type		SKF *NJ 206 ECJ
Bearing type		Cylindrical roller bearing (single row)
Bearing position (mm)	[y _{local}]	8.000
Bearing position (mm)	[y _{global}]	8.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	62.000
Width (mm)	[b]	16.000
Corner radius (mm)	[r]	1.000
Basic static load rating	[C ₀]	36.500
Basic dynamic load rating	[C]	44.000
Fatigue load rating	[C _U]	4.600
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Label in the model		Rolling bearing
Bearing type		SKF *NJ 206 ECJ
Bearing type		Cylindrical roller bearing (single row)
Bearing position (mm)	[y _{local}]	110.000
Bearing position (mm)	[y _{global}]	110.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	30.000
External diameter (mm)	[D]	62.000
Width (mm)	[b]	16.000
Corner radius (mm)	[r]	1.000
Basic static load rating	[C ₀]	36.500
Basic dynamic load rating	[C]	44.000
Fatigue load rating	[C _U]	4.600
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C _{theo}]	0.000
Basic static load rating (kN)	[C _{0theo}]	0.000

Results

Shaft

Maximum deflection (μm)	74.854
Position of the maximum (mm)	59.000
Mass center of gravity (mm)	59.000
Total axial load (N)	0.000
Torsion under torque (°)	0.000

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	[u _A]	10.00	μm
Rolling bearings, classical calculation (contact angle considered)			

Shaft 'Shaft 1' Rolling bearing 'Rolling bearing'

Position (Y-coordinate)	[y]	8.00	mm
Equivalent load	[P]	20.00	kN
Equivalent load	[P ₀]	20.00	kN
Life modification factor for reliability[a ₁]		1.000	
Nominal bearing service life	[L _{nh}]	153.97	h
Static safety factor	[S ₀]	1.83	
Bearing reaction force	[F _x]	0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	-19.996	kN
Bearing reaction force	[F _r]	19.996	kN (-90°)
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _{rr}]	0.322	Nm
Sliding moment of friction	[M _{sl}]	0.028	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 10000/1 EN:2013			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.350	Nm
Power loss	[P _{loss}]	54.956	W

The moment of friction is calculated according to the details in SKF Catalog 2013.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

Displacement of bearing	[u _x]	0.000	μm
Displacement of bearing	[u _y]	0.000	μm
Displacement of bearing	[u _z]	16.250	μm
Displacement of bearing	[u _r]	16.250	μm (90°)
Misalignment of bearing	[r _x]	1.781	mrad (6.12')
Misalignment of bearing	[r _y]	-0.000	mrad (0')
Misalignment of bearing	[r _z]	-0.000	mrad (0')
Misalignment of bearing	[r _r]	1.781	mrad (6.12')

Shaft 'Shaft 1' Rolling bearing 'Rolling bearing'

Position (Y-coordinate)	[y]	110.00	mm
Equivalent load	[P]	20.00	kN
Equivalent load	[P ₀]	20.00	kN
Life modification factor for reliability[a ₁]		1.000	
Nominal bearing service life	[L _{nh}]	153.97	h
Static safety factor	[S ₀]	1.83	
Bearing reaction force	[F _x]	-0.000	kN
Bearing reaction force	[F _y]	0.000	kN
Bearing reaction force	[F _z]	-19.996	kN
Bearing reaction force	[F _r]	19.996	kN (-90°)
Oil level	[H]	0.000	mm
Rolling moment of friction	[M _{rr}]	0.322	Nm
Sliding moment of friction	[M _{sl}]	0.028	Nm
Moment of friction, seals	[M _{seal}]	0.000	Nm
Moment of friction for seals determined according to SKF main catalog 10000/1 EN:2013			
Moment of friction flow losses	[M _{drag}]	0.000	Nm
Torque of friction	[M _{loss}]	0.350	Nm
Power loss	[P _{loss}]	54.956	W

The moment of friction is calculated according to the details in SKF Catalog 2013.

The calculation is always performed with a coefficient for additives in the lubricant $\mu_{bl}=0.15$.

Displacement of bearing	[u _x]	0.000	μm
Displacement of bearing	[u _y]	0.000	μm
Displacement of bearing	[u _z]	16.250	μm

Displacement of bearing	$[u_r]$	16.250	μm (90°)
Misalignment of bearing	$[r_x]$	-1.781	mrad (-6.12')
Misalignment of bearing	$[r_y]$	-0.000	mrad (0')
Misalignment of bearing	$[r_z]$	0.000	mrad (0')
Misalignment of bearing	$[r_r]$	1.781	mrad (6.12')

Damage (%) [H] (20000.000)

No.	B1	B2
19999.99	9999.99	

Σ9999.99 9999.99

Utilization (%) [H] (20000.000)

B1	B2
430.60	430.60

B1: Rolling bearing

B2: Rolling bearing

Displacement [mm]

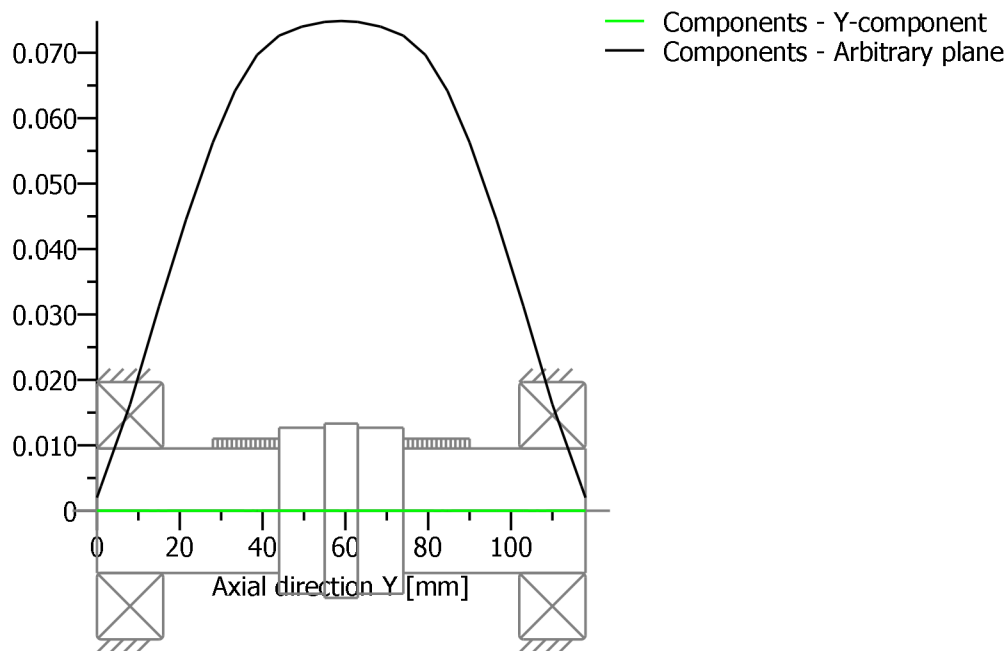
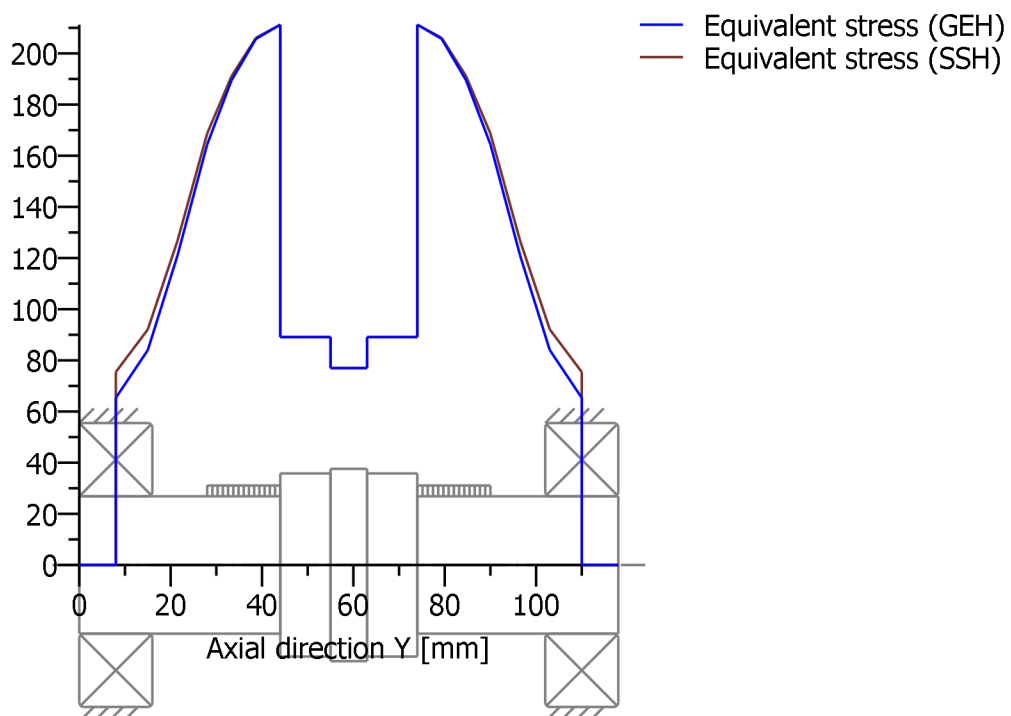


Figure: Deformation (bending etc.) (Arbitrary plane 90 120)

Stress [N/mm²]



Nominal stresses, without taking into account stress concentrations

GEH(von Mises): $\sigma_V = ((\sigma_B + \sigma_{Z,D})^2 + 3 \cdot (\tau_T + \tau_S)^2)^{1/2}$

SSH(Tresca): $\sigma_V = ((\sigma_B - \sigma_{Z,D})^2 + 4 \cdot (\tau_T + \tau_S)^2)^{1/2}$

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Edition, 2012)

Summary

Shaft 1

Material	34 CrNiMo 6 (2)
Material type	Through hardened steel
Material treatment	flame/ind. hardened
Surface treatment	No

Calculation of endurance limit and the static strength

Rolled steel, through hardening steel

Overload case F2 (chapter 2.4.2): Constant stress ratio

Safety number according Chapter 2.5	[jF]	1.20
Safety number according Chapter 1.5	[jm]	1.60
Safety number according Chapter 1.5	[jp]	1.20
Safety number according Chapter 1.5	[jmt]	1.20
Safety number according Chapter 1.5	[jpt]	1.00

Safety number according Chapter 1.5	[jG]	1.00
-------------------------------------	------	------

Cross section	Pos (Y co-ord) (mm)	
C-C	74.00	Shoulder with relief groove
B-B	44.00	Shoulder with relief groove
D-D	79.33	Smooth shaft
A-A	38.67	Smooth shaft

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
C-C	1.85	0.93	0.72	1.67	3.00	3.70
B-B	1.85	0.93	0.72	1.67	3.00	3.70
D-D	0.97	0.93	0.47	2.57	3.22	3.98
A-A	0.97	0.93	0.47	2.57	3.22	3.98

Nominal safety:	1.20	1.20	1.60
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Abbreviations:

Kfb: Notch factor bending

KRs: Surface factor

ALGmax: Highest utilization

SD: Safety endurance limit

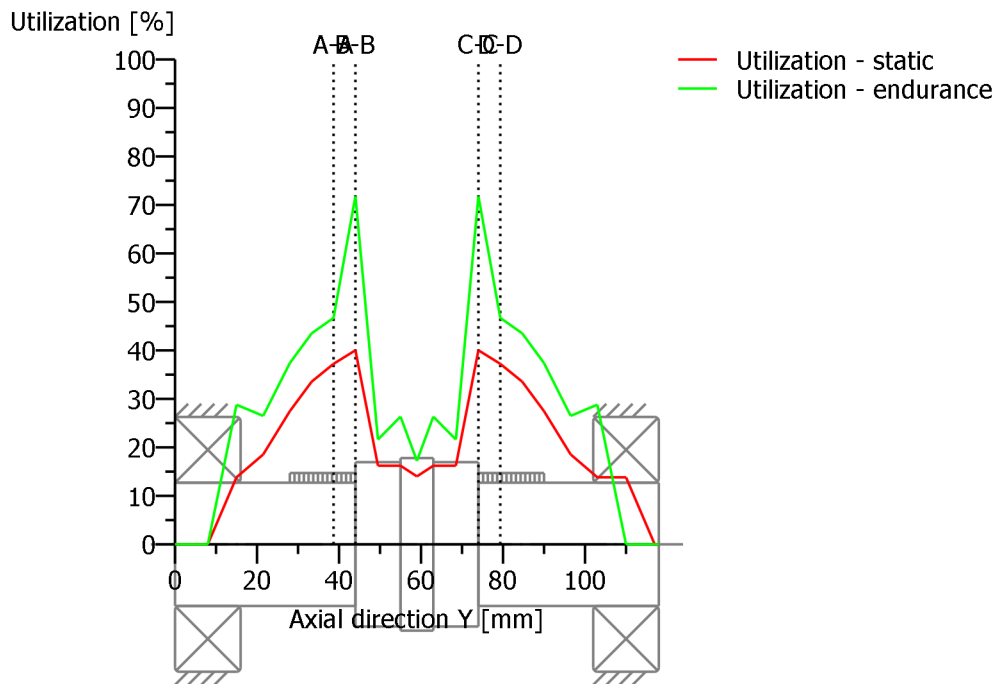
SS: Safety against yield point

SB: Safety against tensile stress

Utilization (%) [Smin/S]

Cross section	static	endurance
C-C	40.021	71.798
B-B	40.020	71.795
D-D	37.222	46.699
A-A	37.220	46.697

Maximum utilization (%)	[A]	71.798
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Utilization = S_{min}/S (%)

Figure: Strength

Calculation details

General statements

Label	Shaft 1		
Drawing			
Length (mm)	[l]	118.00	
Speed (1/min)	[n]	1500.00	

Material	34 CrNiMo 6 (2)
Material type	Through hardened steel
Material treatment	flame/ind. hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.500	1.500	1.500	1.500
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, through hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm ²)	[R _{m,N}]	1200.00
Yield point (N/mm ²)	[R _{p,N}]	1000.00
Fatigue limit (N/mm ²)	[σ _{zdWN}]	540.00
Fatigue limit (N/mm ²)	[σ _{bWN}]	570.00

Fatigue limit (N/mm ²)	[σ_{WN}]	340.00		
Fatigue limit (N/mm ²)	[τ_{sWN}]	310.00		
Breaking elongation (%)	[A]	9.00		
Reference diameter (mm)	[d_{effNm} , d_{effNp}]	16.00	16.00	
Required life time	[H]	20000.00		
Number of load cycles	[NL]	1800000000		
Endurance limit for single stage use				
Temperature (°C)	[Temperatur]	20.000		
Temperature duration (h)	[TemperaturD]	1000.000		
Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[σ_Z , σ_D , f_r , R_{pmax}]	1.000	1.000	0.577	1150.0
[f_{Wt} , f_{Ws}]	0.577	0.450		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsig, R_{mNmin}]	0.500	2700.0	0.220	400.0
[MS, MT]	0.2678	0.1546		
[$k\sigma$, $k\tau$]	15	25		
[$kD\sigma$, $kD\tau$]	0	0		
[ND σ , ND τ]	1e+006	1e+006		
[ND σ_{II} , ND τ_{II}]	0	0		

Thickness of raw material (mm) [d.eff] 45.00

Material data calculated acc. FKM directive with K_{dm} , K_{dp}

Geometric size factors (K_{dm} , K_{dp}) calculated from raw diameter

Material strength calculated from size of raw material

Constants	[adm, adp]	0.330	0.390
Size factors	[K_{dm} , K_{dp}]	0.876	0.851
Tensile strength (N/mm ²)	[R_m]	1050.86	
Yield point (N/mm ²)	[R_p]	850.65	
σ_{zdW} (N/mm ²)	[σ_{zdW}]	472.89	
σ_{bW} (N/mm ²)	[σ_{bW}]	499.16	
τ_{tW} (N/mm ²)	[τ_{tW}]	297.74	
τ_{sW} (N/mm ²)	[τ_{sW}]	271.47	

Overload case F2 (chapter 2.4.2): Constant stress ratio

Safety number according Chapter 1.5	[j _m]	1.60
Safety number according Chapter 1.5	[j _p]	1.20
Safety number according Chapter 1.5	[j _{mt}]	1.20
Safety number according Chapter 1.5	[j _{pt}]	1.00
Safety number according Chapter 2.5	[j _F]	1.20
Safety number according Chapter 1.5	[j _G]	1.00

Cross section 'C-C' Shoulder with relief groove

Comment	Y= 74.00mm	
Position (Y-Coordinate) (mm)	[y]	74.000
External diameter (mm)	[da]	30.000
Inner diameter (mm)	[di]	0.000

Notch effect	Shoulder with relief groove				
[D, d, D1, r, t1] (mm)	40.000	29.600	30.000	1.200	Qu[0].Geo.t
Shape B					
Mean roughness (µm)	[Rz]		2.800		

Tension/Compression Bending Torsion Shearing					
Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]	0.0	0.0	0.0	2.1	
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	559.9	0.0	0.0	
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	0.0	839.9	0.0	3.2	
Cross section, moment of resistance: (mm²)					
[A, Wb, Wt, A]	688.1	2546.1	5092.2	688.1	

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	0.000	0.000	0.000	0.004
[σaz, σab, τat, τas]	0.000	219.910	0.000	0.000
[σzmax, σbmax, τtmax, τsmax]	0.000	329.865	0.000	0.006

FATIGUE PROOF:

Total safety factor according chapter 2.5.3	[jD]	1.200
(Formula: $jD = jF \cdot jG / KTD$)		

Tension/Compression Bending Torsion Shearing					
Stress concentration factor	[a]	2.451	2.200	1.617	1.309
References stress slope	[G]	2.102	2.102	0.958	0.958
Support number	[n(r)]	1.155	1.155	1.185	1.185
Support number	[n(d)]	1.027	1.027	1.040	1.040
Mechanical material support factor	[nwm]	1.032	1.032	1.032	1.032
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.122	1.854	1.313	1.105
Roughness factor	[KR]	0.929	0.929	0.959	0.959
Coefficient of surface strengthening	[KV]	1.500	1.500	1.200	1.200
Design coefficient	[KWK]	1.465	1.287	1.130	0.956
Fatigue limit of part (N/mm²)	[SWK]	322.730	367.550	240.273	283.906

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.678	1.000	0.789	0.789
Permissible amplitude (N/mm²)	[SAK]	218.711	367.547	189.520	223.937
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	218.711	367.547	189.520	223.937
Rate of utilization	[aBK]	0.000	0.718	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.718

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	0.007
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.718
Safety endurance limit assessment	[S.Dauer]	1.671

Required safety [jD] 1.200
Result (%) [S/jD] 139.3

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.295
(Formula: $jges = jG \cdot \max(jm/KTm \cdot Rp/Rm, jp/KTp, jmt/KTm \cdot Rp/Rm, jpt/KTtp)$)

		Tension/Compression Bending Torsion Shearing			
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.1627	1.1627	1.0000
Strength of part (N/mm ²)	[SSK]	850.65	989.06	571.04	491.12
Rate of utilization	[aSK]	0.000	0.432	0.000	0.000

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm²) [SvBn] 329.865
Rate of utilization [aSKvBn] 0.432

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²) [SvQn] 0.011
Rate of utilization [aSKvQn] 0.000

Highest utilization [aSKmax] 0.432

Safety for fracture and yield stresses:

Safety against fracture [S.Rm] 3.704
Required safety [jm/Ktm] 1.600
Result (%) [S/jm] 231.5

Safety against yield point [S.Rp] 2.998
Required safety [jp/KTp] 1.200
Result (%) [S/jp] 249.9

Cross section 'B-B' Shoulder with relief groove

Comment Y= 44.00mm
Position (Y-Coordinate) (mm) [y] 44.000
External diameter (mm) [da] 30.000
Inner diameter (mm) [di] 0.000
Notch effect Shoulder with relief groove
[D, d, D1, r, t1] (mm) 40.000 29.600 30.000 1.200 Qu[1].Geo.t
Shape B
Mean roughness (µm) [Rz] 2.800

		Tension/Compression Bending Torsion Shearing			
Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]		0.0	0.0	0.0	0.9
Amplitude [Fzda, Mba, Ta, Fqa]		0.0	559.9	0.0	0.0
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]			0.0	839.8	0.0
Cross section, moment of resistance: (mm ²)					1.3
[A, Wb, Wt, A]		688.1	2546.1	5092.2	688.1

Stresses: (N/mm²)
[σmz, σmb, τmt, τms] 0.000 0.000 0.000 0.002
[σaz, σab, τat, τas] 0.000 219.903 0.000 0.000

[σ_{zmax}, σ_{bmax}, τ_{tmax}, τ_{smax}] 0.000 329.854 0.000 0.003

FATIGUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.200

(Formula: $jD = jF \cdot jG / KTD$)

		Tension/Compression Bending Torsion Shearing			
Stress concentration factor	[a]	2.451	2.200	1.617	1.309
References stress slope	[G']	2.102	2.102	0.958	0.958
Support number	[n(r)]	1.155	1.155	1.185	1.185
Support number	[n(d)]	1.027	1.027	1.040	1.040
Mechanical material support factor	[nwm]	1.032	1.032	1.032	1.032
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.122	1.854	1.313	1.105
Roughness factor	[KR]	0.929	0.929	0.959	0.959
Coefficient of surface strengthening	[KV]	1.500	1.500	1.200	1.200
Design coefficient	[KWK]	1.465	1.287	1.130	0.956
Fatigue limit of part (N/mm ²)	[SWK]	322.730	367.550	240.273	283.906

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.678	1.000	0.789	0.789
Permissible amplitude (N/mm ²)	[SAK]	218.711	367.549	189.520	223.937
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm ²)	[SBK]	218.711	367.549	189.520	223.937
Rate of utilization	[aBK]	0.000	0.718	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.718

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_2]	0.003
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.718
Safety endurance limit assessment	[S.Dauer]	1.671
Required safety	[jD]	1.200
Result (%)	[S/jD]	139.3

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.295

(Formula: $jges = jG \cdot \text{Max}(j_m / K T_m \cdot R_p / R_m, j_p / K T_p, j_{mT} / K T_m \cdot R_p / R_m, j_{pT} / K T_p)$)

		Tension/Compression Bending Torsion Shearing			
Plastic notch factor	[Kpb, Kpt]	1.700	1.330		
Plastic support number	[npl]	1.0000	1.1627	1.1627	1.0000
Strength of part (N/mm ²)	[SSK]	850.65	989.06	571.04	491.12
Rate of utilization	[aSK]	0.000	0.432	0.000	0.000

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm ²)	[SvBn]	329.854
Rate of utilization	[aSKvBn]	0.432

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm ²)	[SvQn]	0.004
Rate of utilization	[aSKvQn]	0.000
Highest utilization	[aSKmax]	0.432
Safety for fracture and yield stresses:		
Safety against fracture	[S.Rm]	3.704
Required safety	[jm/Ktm]	1.600
Result (%)	[S/jm]	231.5
Safety against yield point	[S.Rp]	2.998
Required safety	[jp/KTp]	1.200
Result (%)	[S/jp]	249.9

Cross section 'D-D' Smooth shaft

Comment		
Position (Y-Coordinate) (mm)	[y]	79.333
External diameter (mm)	[da]	30.000
Inner diameter (mm)	[di]	0.000
Notch effect	Smooth shaft	
Mean roughness (µm)	[Rz]	2.800

Tension/Compression Bending Torsion Shearing

Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]	0.0	0.0	0.0	6664.5	
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	542.1	0.0	0.0	
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	0.0	813.2	0.0	9996.7	
Cross section, moment of resistance: (mm ²)					
[A, Wb, Wt, A]	706.9	2650.7	5301.4	706.9	

Stresses: (N/mm ²)				
[σmz, σmb, τmt, rms]	0.000	0.000	0.000	12.571
[σaz, σab, τat, τas]	0.000	204.528	0.000	0.000
[σzmax, σbmax, τtmax, τsmax]	0.000	306.793	0.000	18.857

FATIGUE PROOF:

Total safety factor according chapter 2.5.3	[jD]	1.200
(Formula: $jD = jF \cdot jG / KTD$)		

Tension/Compression Bending Torsion Shearing

Stress concentration factor	[a]	1.000	1.000	1.000	1.000
References stress slope	[G']	0.000	0.000	0.000	0.000
Support number	[n(r)]	1.000	1.000	1.000	1.000
Support number	[n(d)]	1.027	1.027	1.040	1.040
Mechanical material support factor	[nwm]	1.032	1.032	1.032	1.032
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	1.000	0.974	0.962	1.000
Roughness factor	[KR]	0.929	0.929	0.959	0.959
Coefficient of surface strengthening	[KV]	1.000	1.200	1.200	1.200
Design coefficient	[KWK]	1.076	0.875	0.837	0.869
Fatigue limit of part (N/mm ²)	[SWK]	439.370	540.547	324.325	312.437

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.678	0.972	0.789	0.789
Permissible amplitude (N/mm ²)	[SAK]	297.757	525.564	255.817	246.441
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm ²)	[SBK]	297.757	525.564	255.817	246.441
Rate of utilization	[aBK]	0.000	0.467	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.467

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm ²)	[SmV_2]	21.774
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.467
Safety endurance limit assessment	[S.Dauer]	2.570
Required safety	[jD]	1.200
Result (%)	[S/jD]	214.1

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.295

(Formula: $jges = jG \cdot \max(jm/KTm \cdot Rp/Rm, jp/KTp, jmt/KTm \cdot Rp/Rm, jpt/KTtp)$)

		Tension/Compression Bending Torsion Shearing			
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.1627	1.1627	1.0000
Strength of part (N/mm ²)	[SSK]	850.65	989.06	571.04	491.12
Rate of utilization	[aSK]	0.000	0.402	0.000	0.050

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm ²)	[SvBn]	306.793
Rate of utilization	[aSKvBn]	0.402

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm ²)	[SvQn]	32.661
Rate of utilization	[aSKvQn]	0.050

Highest utilization	[aSKmax]	0.402
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Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	3.983
Required safety	[jm/Ktm]	1.600
Result (%)	[S/jm]	248.9

Safety against yield point	[S.Rp]	3.224
Required safety	[jp/KTp]	1.200
Result (%)	[S/jp]	268.7

Cross section 'A-A' Smooth shaft

Comment		
Position (Y-Coordinate) (mm)	[y]	38.667
External diameter (mm)	[da]	30.000

Inner diameter (mm)	[di]	0.000
Notch effect		Smooth shaft
Mean roughness (µm)	[Rz]	2.800

Tension/Compression Bending Torsion Shearing

Load: (N) (Nm)				
Mean value [Fzdm, Mbm, Tm, Fqm]	0.0	0.0	0.0	6665.7
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	542.1	0.0	0.0
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	0.0	813.2	0.0	9998.6
Cross section, moment of resistance: (mm²)				
[A, Wb, Wt, A]	706.9	2650.7	5301.4	706.9

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	0.000	0.000	0.000	12.573
[σaz, σab, τat, τas]	0.000	204.519	0.000	0.000
[σzmax, σbmax, τtmax, τsmax]	0.000	306.778	0.000	18.860

FATIGUE PROOF:

Total safety factor according chapter 2.5.3	[jD]	1.200
(Formula: $jD = jF \cdot jG / KTD$)		

Tension/Compression Bending Torsion Shearing

Stress concentration factor	[a]	1.000	1.000	1.000	1.000
References stress slope	[G]	0.000	0.000	0.000	0.000
Support number	[n(r)]	1.000	1.000	1.000	1.000
Support number	[n(d)]	1.027	1.027	1.040	1.040
Mechanical material support factor	[nwm]	1.032	1.032	1.032	1.032
The support factor is determined with the support factor as defined by Stielor.					
Notch effect coefficient beta	[Kf]	1.000	0.974	0.962	1.000
Roughness factor	[KR]	0.929	0.929	0.959	0.959
Coefficient of surface strengthening	[KV]	1.000	1.200	1.200	1.200
Design coefficient	[KWK]	1.076	0.875	0.837	0.869
Fatigue limit of part (N/mm²)	[SWK]	439.370	540.547	324.325	312.437

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.678	0.972	0.789	0.789
Permissible amplitude (N/mm²)	[SAK]	297.757	525.560	255.817	246.441
Effective Miner sum	[DM]	0.3	0.3	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	297.757	525.560	255.817	246.441
Rate of utilization	[aBK]	0.000	0.467	0.000	0.000

Calculation of the combined stress types:

Rate of utilization for the combined load components

a) For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.467

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	21.778
Rate of utilization	[aBKv_2]	0.000

Highest utilization	[aBKmax]	0.467
Safety endurance limit assessment	[S.Dauer]	2.570
Required safety	[jD]	1.200

Result (%) [S/jD] 214.1

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.295

(Formula: $jges = jG \cdot \text{Max}(j_m/KT_m \cdot R_p/R_m, j_p/KT_p, j_{mt}/KT_m \cdot R_p/R_m, j_{pt}/KT_p)$)

		Tension/Compression Bending Torsion Shearing			
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.1627	1.1627	1.0000
Strength of part (N/mm ²)	[SSK]	850.65	989.06	571.04	491.12
Rate of utilization	[aSK]	0.000	0.402	0.000	0.050

Rate of utilization for the combined load components:

a) For outer surface (shear stress = 0)

Equivalent stress (N/mm²) [SvBn] 306.778
Rate of utilization [aSKvBn] 0.402

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²) [SvQn] 32.667
Rate of utilization [aSKvQn] 0.050

Highest utilization [aSKmax] 0.402

Safety for fracture and yield stresses:

Safety against fracture [S.Rm] 3.983
Required safety [j_m/K_{tm}] 1.600
Result (%) [S/j_m] 248.9

Safety against yield point [S.Rp] 3.224
Required safety [j_p/K_{tp}] 1.200
Result (%) [S/j_p] 268.7

Important remarks concerning strength calculation according to FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization ≤ 1
- Currently the following restrictions still apply::
Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
 $\beta_S = 1.0 + (\beta_T - 1.0) / 2.0$ (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with p = 20MPa
- Proven safety: Effective safety according to special formula,
condition: safety > required safety or result > 100%

End of Report

lines: 799

E.3 KISSsoft Report - Interference Fit Hub

Name : Interference fit

Changed by: up201106920

on: 11.06.2017

at: 22:57:25

Important hint: At least one warning has occurred during the calculation:

1-> The working deviation is too small at the lower boundary of the tolerance range !

2-> Safety against sliding:

The required safety is not achieved!

3-> The minimal pressure is too small for the given load.

($p_{min} = 0.0000 \text{ N/mm}^2 < p_{b+pr} = 93.3333 \text{ N/mm}^2$)

Cylindrical interference fit [M01a]

Calculation method: According DIN 7190:2001 (elastic range)
with additions (Centrifugal force, Micro sliding, Assembly, etc.)
for shrunken or pressed fits

Diameter shaft (mm) [DiA]/[DiI] 30.00/ 0.00
Diameter hub (mm) [DaA]/[DaI] 37.35/ 30.00

Entered outer-diameter of hub and the corresponding lengths in mm:

Part	A:	D =	36.000	l =	11.000
Part	B:	D =	42.000	l =	8.000
Part	C:	D =	36.000	l =	11.000

Equivalent outer-diameter (mm)	[DaA]	37.35
Length of Interference fit (mm)	[l]	30.00

Diameter of joint (mm)	[DF]	30.00
Tolerance Shaft		p6
Upper allowance Shaft (µm)	[AoI]	35.0
Lower allowance Shaft (µm)	[AuI]	22.0
Tolerance measure Shaft (µm)	[TI]	13.0
Tolerance hub		H7
Upper allowance Hub (µm)	[AoA]	21.0
Lower allowance Hub (µm)	[AuA]	0.0
Tolerance measure Hub (µm)	[TA]	21.0
Largest interference (µm)	[Po]	35.0
Smallest interference (µm)	[Pu]	1.0

Nominal torque (Nm)	[T]	0.00
Application factor	[KA]	1.00
Service torque (Nm)	[Tb]	0.00
Axial force (N)	[FA]	10000.00
Bending moment (Nm)	[Mb]	560.00
Radial force (N)	[Fr]	0.00
Circumferential force (N)	[Fu]	0.00

Speed (1/min)	[n]	1500.00		
Interference (µm)	[P]	18.0 (1.0.. 35.0)		
Embedding (µm)	[s]	4.48		
Effective interference at 20°C(68°F) (µm)	[Pw]	13.52 (-3.5.. 30.5)		
Effective interference at working temp.. (µm)	[PwTh]	13.52 (-3.5.. 30.5)		
Service temperature shaft (°C)	[ThB]	100.0		
Service temperature hub (°C)	[ThB]	100.0		
Pressure stress by				
- Interference (after mounting) (N/mm²)	[pM]	16.46 (0.00.. 37.17)		
- Interference (working) (N/mm²)	[p]	16.45 (0.00.. 37.16)		
- Bending moment (N/mm²)	[pb]	93.33		
- Radial force (N/mm²)	[pr]	0.00		
Coefficient of friction, axial	[µa]	0.200		
Coefficient of friction, circumferential	[µu]	0.200		
Safety against sliding	[Sr]	0.93 (0.00.. 2.10)		
Required safety against sliding	[SSr]	1.20		

Equivalent stress according to von Mises
 $\sigma_V = (\sigma_R^2 + \tau^2 - \sigma_R \cdot \tau)^{0.5}$

Shaft

Material	34 CrNiMo 6 (2)			
Young's modulus (N/mm²)	[E]	206000.00		
Poisson's ratio (-)	[ν]	0.30		
Density (kg/m³)	[ρ]	7830.00		
Coefficient of thermal expansion (10 ⁻⁶ /K)	[α]	11.50		
Tensile strength (N/mm²)	[Rm]	1100.00		
Diameter range (mm)=	16- 40)			
Yield point (N/mm²)	[Rp]	900.00		
Diameter range (mm)=	16- 40)			
Surface class of roughness	N5 Rz=2.8 (Honing)			
Surface roughness (µm)	[Rz]	2.80		
External diameter (mm)	[DiA]	30.00		
Diameter increase (µm)	[ΔD]	25.92 (27.60.. 23.81)		
Diameter increase (o/oo)	[ΔD]	0.86 (0.92.. 0.79)		
Equivalent stress outsideø (N/mm²)	[σVa]	16.45 (0.01.. 37.15)		
- Radial stress (N/mm²)	[σRa]	-16.45 (0.00.. -37.16)		
- Tangential stress (N/mm²)	[σTa]	-16.45 (0.01.. -37.15)		
- With outside load (N/mm²)	[σVaMF]	109.78 (93.33.. 130.49)		
Inner diameter (mm)	[DiI]	0.00		
Equivalent stress insideø (N/mm²)	[σVi]	16.44 (0.02.. 37.14)		
- Radial stress (N/mm²)	[σRi]	-16.44 (0.02.. -37.14)		
- Tangential stress (N/mm²)	[σTi]	-16.44 (0.02.. -37.14)		
Safety against fracture	[SiRm]	10.02 (11.79.. 8.43)		
Required safety against fracture	[SSi.Rm]	1.50		
Safety against yield point	[SiRp]	8.20 (9.64.. 6.90)		
Required safety against yield point	[SSi.Rp]	1.00		

Hub

Material	34 CrNiMo 6 (2)		
Young's modulus (N/mm ²)	[EA]	206000.00	
Poisson's ratio (-)	[vA]	0.30	
Density (kg/m ³)	[ρ]	7830.00	
Coefficient of thermal expansion (10 ⁻⁶ /K)	[α]	11.50	
Tensile strength (N/mm ²)	[Rm]	1200.00	
Diameter range (mm)=	16- 40)		
Yield point (N/mm ²)	[Rp]	1000.00	
Diameter range (mm)=	16- 40)		
Surface class of roughness	N5 Rz=2.8 (Honing)		
Surface roughness (μm)	[Rz]	2.80	
Equivalent outer-diameter (mm)	[Da]	37.35	
Diameter increase (μm)	[ΔD]	45.22 (34.37.. 58.88)	
Diameter increase (o/oo)	[ΔD]	1.21 (0.92.. 1.58)	
Equivalent stress outsideø (N/mm ²)	[σVa]	59.92 (0.05.. 135.25)	
- Radial stress (N/mm ²)	[σRa]	-0.00 (0.00.. -0.00)	
- Tangential stress (N/mm ²)	[σTa]	59.92 (0.05.. 135.25)	
Inner diameter (mm)	[Dal]	30.00	
Diameter increase (μm)	[ΔD]	39.44 (27.61.. 54.33)	
Diameter increase (o/oo)	[ΔD]	1.31 (0.92.. 1.81)	
Equivalent stress insideø (N/mm ²)	[σVi]	85.81 (0.06.. 193.69)	
- Radial stress (N/mm ²)	[σRi]	-16.45 (0.00.. -37.16)	
- Tangential stress (N/mm ²)	[σTi]	76.39 (0.06.. 172.42)	
- Radial stress with outside load (N/mm ²)	[σRai]	-109.79 (-93.33.. -130.49)	
- Tangential stress with outside load (N/mm ²)	[σTai]	509.34 (433.02..605.38)	
- With outside load (N/mm ²)	[σViMF]	572.19 (486.45.. 680.08)	
Safety against fracture	[Si.Rm]	2.10 (2.47.. 1.76)	
Required safety against fracture	[SSi.Rm]	1.50	
Safety against yield point	[Si.Re]	1.75 (2.06.. 1.47)	
Required safety against yield point	[SSi.Re]	1.00	

Mere elastic stress, no verification of elastic plastic interference fit according to DIN 7190.

Service / Mounting / Dismounting

Transverse-interference-fit:			
Mounting clearance (mm)	[PsTh]	0.030	
Temperature difference for mounting:			
Shaft temperature: (°C)	Hub temperature:	[ThA] (°C)	
20	208		
-150	83		
(calculated using coefficient of thermal expansion)			
Sub-cooling of the shaft according to DIN 7190	(10 ⁻⁶ /K)	[αw]	8.50
Coefficient of thermal expansion for hub	(10 ⁻⁶ /K)	[αn]	11.50
Longitudinal pressure fit:			
Assembly temperature shaft (°C)	[ThM]	20.00	
Assembly temperature hub (°C)	[ThM]	20.00	
Coefficient. of friction (longitudinal)	[μe=μa*1.300]	0.26	

Press on (force) (kN)	[Fpress]	12.10 (0.00.. 27.32)
Coefficient. of friction (longitudinal)	[μ ll= μ a*1.600]	0.32
Press out (force) (kN)	[Fpress]	14.90 (0.00.. 33.63)
Coefficient. of friction	[μ]	0.32
Max. torque to avoid Micro sliding (Nm)	[Tlimit]	23.51 (0.00.. 53.09)

Notice concerning the display: Number-1 (Number-2.. Number-3):

Number-1: Value calculated with the mean allowance

Number-2: Value of the smallest possible allowance

Number-3: Value of the largest possible allowance

Notice: All strains and stresses are calculated for the purely elastic case.

End of Report

lines: 172

EES Results

Model based on general correlations

Equations

\$UnitSystem SI K Pa J

$$A_v = 0,135 \cdot 0,136 \cdot 2 + 0,135 \cdot 0,225 \cdot 2 \quad (1)$$

$$Per_v = (0,0675 \cdot 2 + 0,086 \cdot 2) \cdot 2 + (0,0675 \cdot 2 + 0,175 \cdot 2) \cdot 2 \quad (2)$$

$$Charac_{lenthgh,v} = 0,135 \quad (3)$$

$$W1_v = 0,175 \quad (4)$$

$$W2_v = 0,086 \quad (5)$$

$$A_{hb} = 0,086 \cdot 0,175 \quad (6)$$

$$Per_{hb} = (0,086 + 0,175) \cdot 2 \quad (7)$$

$$Charac_{lenthgh,hb} = A_{hb}/Per_{hb} \quad (8)$$

$$\text{call } fc_{plate,vertical}('Engine_Oil_unused' ; T_{v,int}; T_{oil}; P_{oil}; Charac_{lenthgh,v}) \quad (9)$$

$$h_{conv,v,int}; ;) \quad (10)$$

$$\text{call } fc_{plate,horizontal2}('Engine_Oil_unused' ; T_{hb,int}; T_{oil}; P_{oil}; Charac_{lenthgh,hb}) \quad (11)$$

$$h_{conv,hb,int}; ;) \quad (12)$$

Thermal conductivity in carter walls

$$T_{v,average} = \frac{T_{v,ext} + T_{v,int}}{2} \quad (13)$$

$$t_{ca} = 0,025 \quad (14)$$

$$k_{carter} = k(Stainless_{AISI302}; T = T_{v,average}) \quad (15)$$

Internal convection heat coefficient

$$T_{room} = (20 + 273,15) \quad (16)$$

$$P_{air} = 101300 \quad (17)$$

$$v_{air} = 2 \quad (18)$$

$$\text{call } External_{Flow,VerticalPlate}('Air'; T_{room}; T_{v,ext}; P_{air}; v_{air}; W1, v) \quad (19)$$

$$; h_{conv,v,ext}; ; ;) \quad (20)$$

$$\text{call } fc_{plate,horizontal2}('Air'; T_{hb,ext}; T_{room}; P_{air}; Charac_{lengtgh,hb}) \quad (21)$$

$$h_{conv,hb,ext}; ;) \quad (22)$$

$\dot{Q}_v$ - Heat loss vertical walls

$$stefan_{boltz} = 5,67 \cdot 10^{-8} \quad (23)$$

$$\dot{Q}_v = (T_{oil} - T_{v,int}) \cdot (A_v) \cdot h_{conv,v,int} \quad (24)$$

$$\dot{Q}_v = (T_{v,int} - T_{v,ext}) \cdot k_{carter} \cdot A_v / t_{ca} \quad (25)$$

$$\dot{Q}_{v,convext} = (T_{v,ext} - T_{room}) \cdot (A_v) \cdot h_{conv,v,ext} \quad (26)$$

$$\dot{Q}_{v,radext} = (T_{v,ext}^4 - T_{room}^4) \cdot (A_v) \cdot stefan_{boltz} \quad (27)$$

$$\dot{Q}_v = \dot{Q}_{v,convext} + \dot{Q}_{v,radext} \quad (28)$$

$\dot{Q}_{hb}$ - Heat loss horizontal plate

$$\dot{Q}_{hb} = (T_{oil} - T_{hb,int}) \cdot (A_{hb}) \cdot h_{conv,hb,int} \quad (29)$$

$$\dot{Q}_{hb} = (T_{hb,int} - T_{hb,ext}) \cdot k_{carter} \cdot A_{hb} / t_{ca} \quad (30)$$

$$\dot{Q}_{hb,convext} = (T_{hb,ext} - T_{room}) \cdot (A_{hb}) \cdot h_{conv,hb,ext} \quad (31)$$

$$\dot{Q}_{hb,radext} = (T_{hb,ext}^4 - T_{room}^4) \cdot (A_{hb}) \cdot stefan_{boltz} \quad (32)$$

$$\dot{Q}_{hb} = \dot{Q}_{hb,convext} + \dot{Q}_{hb,radext} \quad (33)$$

$$\dot{Q}_{carter} = \dot{Q}_v + \dot{Q}_{hb} \quad (34)$$

$$T_{oil} = (100 + 273, 15) \quad (35)$$

$$P_{oil} = 101300 \quad (36)$$

Solution

$$\begin{aligned}A_{hb} &= 0,01505 \text{ [m}^2\text{]} \\Charac_{lenghtgh,hb} &= 0,02883 \text{ [m]} \\h_{conv,hb,ext} &= 4,915 \text{ [W/m}^2\text{-K]} \\h_{conv,v,ext} &= 44,07 \text{ [W/m}^2\text{-K]} \\k_{carter} &= 16,08 \text{ [W/m-K]} \\Per_v &= 1,584 \text{ [m]} \\P_{oil} &= 101300 \text{ [Pa]} \\\dot{Q}_{hb} &= 11,94 \text{ [W]} \\\dot{Q}_{hb,radext} &= 7,325 \text{ [W]} \\\dot{Q}_{v,convevt} &= 202,1 \text{ [W]} \\stefan_{boltz} &= 5,670 \times 10^{-8} \text{ [W/m}^2\text{-K}^4\text{]} \\t_{ca} &= 0,025 \text{ [m]} \\T_{hb,int} &= 356,7 \text{ [K]} \\T_{v,average} &= 342,1 \text{ [K]} \\T_{v,int} &= 344 \text{ [K]} \\W1_v &= 0,175 \text{ [m]}\end{aligned}$$

$$\begin{aligned}A_v &= 0,09747 \text{ [m}^2\text{]} \\Charac_{lenghtgh,v} &= 0,135 \text{ [m]} \\h_{conv,hb,int} &= 48,27 \text{ [W/m}^2\text{-K]} \\h_{conv,v,int} &= 82,68 \text{ [W/m}^2\text{-K]} \\Per_{hb} &= 0,522 \text{ [m]} \\P_{air} &= 101300 \text{ [Pa]} \\\dot{Q}_{carter} &= 247,3 \text{ [W]} \\\dot{Q}_{hb,convevt} &= 4,611 \text{ [W]} \\\dot{Q}_v &= 235,3 \text{ [W]} \\\dot{Q}_{v,radext} &= 33,21 \text{ [W]} \\T_{room} &= 293,2 \text{ [K]} \\T_{hb,ext} &= 355,5 \text{ [K]} \\T_{oil} &= 373,2 \text{ [K]} \\T_{v,ext} &= 340,2 \text{ [K]} \\v_{air} &= 2 \text{ [m/s]} \\W2_v &= 0,086 \text{ [m]}\end{aligned}$$