Certification Process for TONISCO Systems Oy

Drilling Device TONISCO B30

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ABSTRACT

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TONISCO System Oy is a producer of hot tapping drilling devices. In order to increase sales of the product TONISCO B30 to the German market it was necessary to get a certificate to satisfy the grown demands of the customers. Part of this work was to find out the right German authority for the certification process, make contact, find out the requirements and fulfill some of these given requirements.

Therefor several authorities were contacted by phone, electronic mail or even in personal meetings and the requirements were discussed. After that first phase the durability of the pressurized parts were checked by written calculations and the parts were modeled in the CAD software Autodesk Inventor Professional 2012. The geometries were imported to FEM software ANSYS® Workbench to analyze the structure. Later the results from the hand calculation were compared to the simulated results.

Both, the hand calculation and the software simulation, are showing that the structure of the B30 is able to resist the loads in the testing environment of 60bar. Also in order to fulfill one of the given requirements, a manual for the drilling device has been created in German language. For the further certification process a detailed project plan has been generated.

The results of the work confirm the experiences of the daily usage of the device. With the already created material the certificate should be achieved.

This is the public version of the thesis and does not include confidential data or appendices.

Key words: certification process, fem simulation

CONTENTS

ABBREVIATIONS AND TERMS

A	Area
A_3	area of thread core
α_1	offset factor
α_n	screw mechanical classifications factor
A_N	nominal area of screw
A_{shaft}	Area of shaft
d	pitch diameter of thread
d_{min}	minimum diameter
δ_S	elastic misalignment (screw)
δ_T	elastic misalignment (part)
D_{shaft}	diameter of shaft
e_1	screw offset
E_S	young's modulus of screw materials
E_T	young's modulus of part materials
F	Force
f_z	setting value
H	thread overlap
k	Kilo (10^3)
k_A	tightening factor
l	length of thread
l_k	grip length in connection
mm	millimeter
N	Newton
n	load introduction factor
p	pressure
P	pitch of thread
R_m	ultimate strength
$R_{p\ 0,2}$	yield strength
s_f	safety factor
s_m	minimum safety factor
σ_l	bearing stress in hole
$\tau_{zy\ max}$	shear stress
t_{min}	smallest thickness

AGFW	German Heat and Power association
BGR	Professional Association Rules
CAD	Computer aided design
DEKRA	German Motor Vehicle Inspection Association
DGR	Pressure Equipment Directive
DIN	German Institute for Standardization
DN	Diametre Nominal, the European equivalent of Nominal Pipe Size
DVGW	German Technical and Scientific Association for Gas and Water
EN	European Committee for Standardization
FEM	Finite Element Method
FH	University of applied science (German)
ISO	International Organization for Standardization
OY	"Osakeyhtiö" equivalent of a limited company (Ltd.)
TAMK	Tampere University of applied science
TÜV	Technical Inspection Association
VdTÜV	Association of TÜV

1 INTRODUCTION

Piping networks are essential for our daily life. They are used to convey several types of fluids, like fresh water, waste water, gas or steam from one point to another.

There are all over the world different standards and norms which help the designer of networks to make decisions which material to use, where to place a valve, which kind of pump is necessary.

But what to do if later some changes in an existing pipeline network are needed?

For example in a factory a new machine needs a connection to the gas pipeline for heating. Or in a municipal area a new built house uses district heating for warming up the building. Should you each times stop the whole system to change the pipelines and integrate a new valve to realize these connections?

At this moment the method of Hot Tapping is an interesting alternative to prevent inconveniences for all involved parties. With Hot Tapping it's not necessary to shut down the fluid flow although you are drilling into the line.

The family owned enterprise TONISCO Systems OY is a worldwide operating company in developing and producing hot tapping drilling devices and other tools in the area for pipeline network operators.

Since in Germany it is required to have permission for every tool and working method you are using and TONISCO tries to improve their sales to the German market they need a certification from an authority.

Because there are many different authorities existing in Germany the topic of this research is to find the right authority, working out the requirements and fulfill them in order to get the permission of the drilling device.

2 TONISCO Systems OY

TONISCO System Oy is a Tampere based family company founded in 1969. During the first years the main business was to develop and supply pipeline maintenance tools. The production of the tools was carried out from different suppliers and subcontractors. But after a few years the production was moved to the company to ensure and improve the quality of the products.

In the beginning most of the customers were from the nearby surroundings, but more and more it was noticed that the products are also interesting for other pipeline network operators in other countries or even in other continents. In the year 2010 over 90% of the business value was generated abroad.

Today TONISCO Systems Oy has customers in over 20 countries in the world which are managed either directly or by agents.

The oldest and most important products of TONISCO are the hot tapping drilling devices. There are 5 different products to build connections to pipelines under pressure.

For branching to district heating networks TONISCO developed its own valve which is the top-selling product beside the drilling equipment like pilot driller and hole saws.

Also TONISCO sells pipeline squeezers with which one can stop defective pipes.

Furthermore TONSICO is not only selling their products but they also offer a branching service where employees of the company are coming to the construction site and do the drilling for the network operator. /1/

3 Description of hot tapping drilling device TONISCO B30

3.1 Hot tapping

In general the method of hot tapping, or pressure tapping, is to make connections to existing pipeline networks or vessels. The main point in this method is that there is no need to interrupt the flow in the pipeline. A hole-saw is used to make the actual opening, so this method is applicable on weldable and non-weldable pipeline networks. The whole procedure is very fast and clean.
Hot tapping is also used to open the pipeline for line stopping plugging heads. /2/

3.1.1 Example of Hot Tapping (non-weldable line)

The method of Hot Tapping has 7 steps. Each step is now detailed described.
In FIGURE 1 you see in the first step where the spit collar needed for non-weldable lines is added around the mainline in the wished direction. The screws are mounted properly so that it’s impossible for the collar to move. /3/

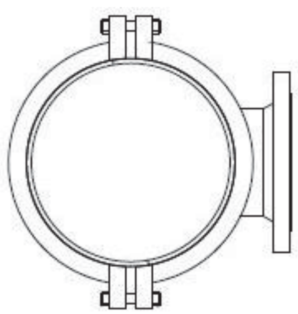

FIGURE 1: Spit collar to mainline /3/

After that the proper valve is mounted to the flange of the spit collar as you see in FIGURE 2. The valve mechanism is set on a completely opened position. /3/

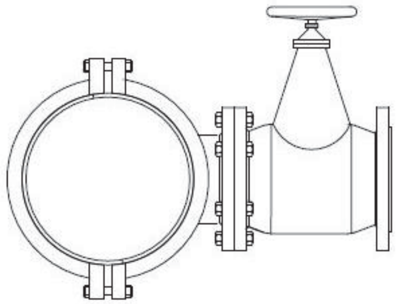

FIGURE 2: Valve to spit collar /3/

FIGURE 3 shows how the drilling device is assembled to the free end of the open valve. With the pressure test the tightness of the drilling device, the valve and the spit collar is tested. If there are any leakages the device and the valve have to be removed and checked. /3/

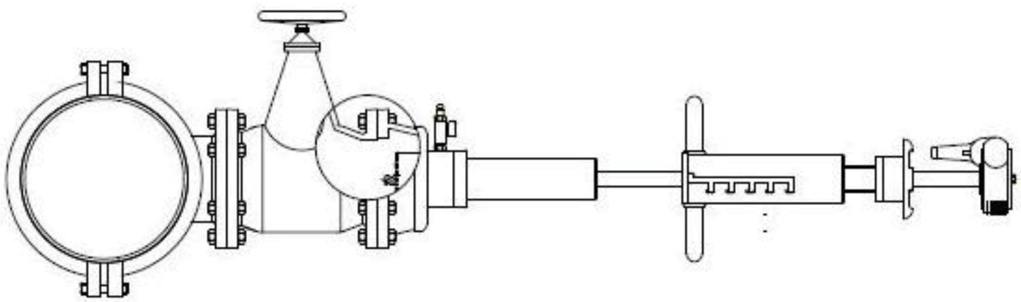

FIGURE 3: Drilling device to valve /3/

The fourth step is illustrated in FIGURE 4, the drilling with the pilot drill starts. After the successful penetration of the drill the pressure of the main pipe is in the whole structure. You can observe the penetration on the pressure gauge that is installed before.
For the drilling with the pilot drill a high rotation speed of the drive unit is chosen. Feed is given carefully in the beginning so that the pilot drill can centered on the pipewall. /3/

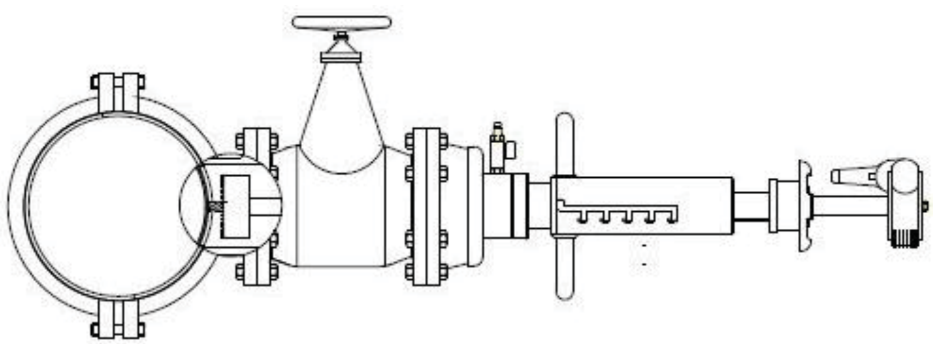

FIGURE 4: Drilling with pilot drill /3/

The actual drilling with the hole saw starts. The feed is given carefully and the rotation speed is low. The wings of the pilot drill are spreading out for some extent to grip the shoulder of the core while drilling through the pipe wall seen in FIGURE 5. /3/

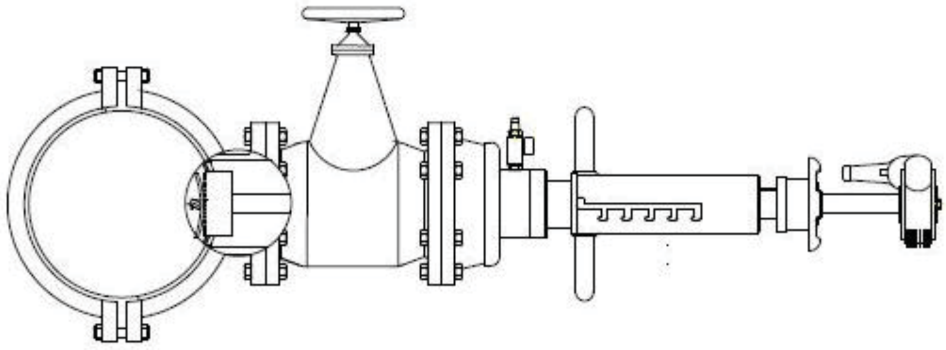

FIGURE 5: Drilling with hole saw /3/

FIGURE 6 shows the second last step when the drilling is completed. The shaft with hole saw and pilot driller is pulled back and the valve is closed. The pressure is discharged through the control cock. The drilling device can be removed from the valve. /3/

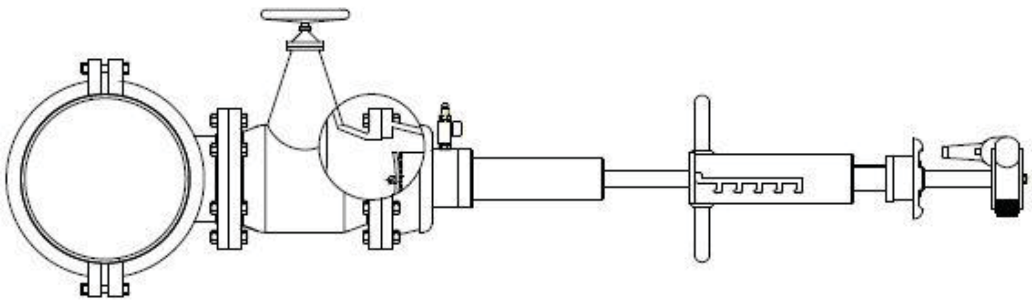

FIGURE 6: Dismantling of the drilling device /3/

The end of the branching line is mounted to the free end of the valve. The valve can be opened and the connection to the main line is produced. The finished branch can be seen in FIGURE 7. /3/

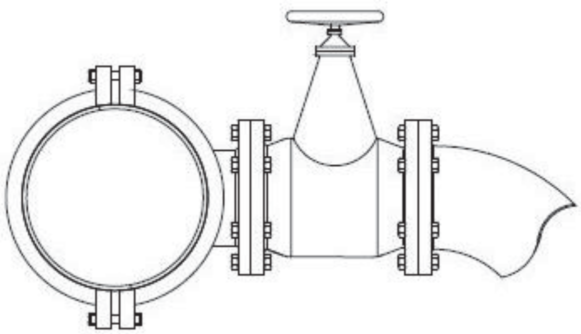

FIGURE 7: Finishing step /3/

3.2 TONISCO B30

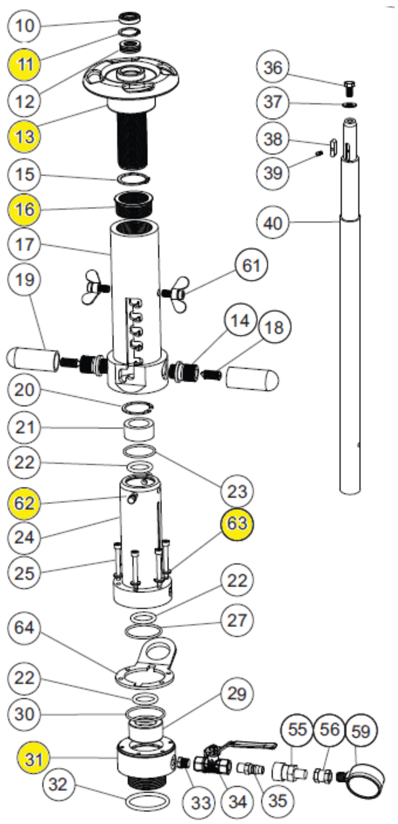

FIGURE 8: Exploded view drawing TONISCO B30 /48/

The TONISCO B30 is a medium sized hot tapping drilling device. It is possible to drill holes from DN 40 to DN 200, which is one of the most used ranges of branches.

The drive unit works either electrical or pneumatically so that the device can be used in different environments. The maximum working-pressure is 40bar and because of the polymeric sealing material the maximum temperature is 200°C.

In the certification process and especially during the testing the B30 is exposed to a pressure of 60bar so that all the calculations are made with this value.

Within the calculations the main focus was on the weakest parts of the structure. /4/

In FIGURE 8 you can see an exploded view drawing of the TONISCO B30. The yellow marked parts are these mentioned weakest parts.

These were the following:

- Retaining ring (11)
- Thread between Feed Wheel(13) and Feed Nut(16)
- Power/Force screw (62)
- Jointing screws (63)
- Thread between the lower body (31) and adapter

In the next chapters I will describe these parts more precisely.

3.2.1 Retaining Ring

Retaining rings are axially mounted and fit resiliently into the groove. There are two types, for shafts (DIN 471) and for boreholes (DIN 472).
The part that is protruding out of the groove forms an axially loadable shoulder and is used to fix components.

In the TONISCO B30 the retaining ring (FIGURE 8 part 11) is the most loaded part of the whole drilling device. The whole force for the pressure and the drilling is going through the shaft and the bearing directly into the ring. Therefore it has to withstand the main part of the stress.
A very big advantage of using a proper retaining ring is the opportunity of easy disassembly and so in case of replacing some parts a fast und uncomplicated possibility.

3.2.2 Thread between Feed Wheel and Feed Nut

The threads of the Feed Wheel and the Feed Nut are parts of a translation screw, which means the screw is designed to transform the turning motion from the Feed Wheel into a linear motion. Further the thread has to be all the time able to move while the parts are under stress. This is possible because of the materials. The material combination in this case is bronze and steel, bronze is softer than steel and has the property of self-lubrication so the probability of jamming is minimized.
But the material composition is not the only special property in a translation screw also the thread itself is a special feature.
There are usually three types of threads in a translation screw, buttress, round and square thread. All of them have high stiction, the screw is self-locking. For example when the operator stops turning the Feed Wheel during the drilling, the linear force of the pressure will not apply a torque to the Feed Wheel so the Wheel is not turning backwards and coming up when interrupting the drilling process.

3.2.3 Force screws

The Force Screws are an own development of TONISCO. FIGURE 9 shows a self-made modeling with the CAD-Software Autodesk Inventor Professional 2012. The Feed Socket hooks with its inlets into the screws and force is transmitted to the Upper Body. The main load of the screws is shear which has its highest value to the point where the thread ends.

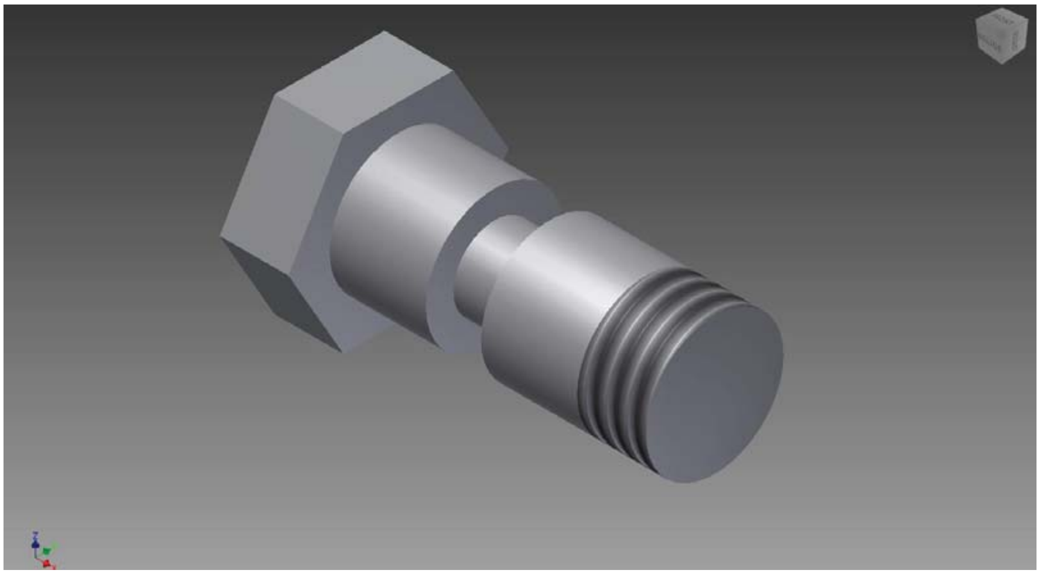

FIGURE 9: Power Screw (Autodesk Inventor modeling, screenshot)

The Material of the Screws is Aisi 316, better known as X5CrNiMo17 /6/; this is stainless steel with a minimum tensile strength of $500 N/mm^2$.

3.2.4 Jointing screws

The jointing screws connect the upper Body with the lower Body. They are normal Allen screws according to ISO 4762 /7/.
They are loaded only axially.

3.2.5 Thread Lower Body and adapter

The connection from the lower Body to the adapter or the valve is a normal M55x4-thread. Which part in this connection is the weaker one depends on the used adapter or valve.

4 Expiration of the certification process

In the beginning of the research the main task was to find out the right authority in order to get the certification. The first assumption was that the DVGW (German Technical and Scientific Association for Gas and Water) would be the right contact partner. As an independent technical-scientific association the DVGW is responsible for the self-regulation of the gas and water supply industry in Germany and also in Europe. The working areas of the DVGW are Regulation and Standardization, Research and Development, Professional Development and Communication. Founded in 1859 it seemed that the DVGW with its huge history and experiences in the water and gas sector would be the right partner for the certification. /8/

But during the phone calls and mail conversations to Germany it turned out that they are only responsible for the pipeline itself and the fittings (e.g. valves) in the pipeline, not for the drilling device.

So the search for other authorities continued. The general technical associations came into consideration like the DEKRA or the TÜV.

The DEKRA (German Motor Vehicle Inspection Association) is nowadays not only focusing on vehicle inspection but also in general technical systems so it was also a possible partner for the certification. /9/

Then during a demonstration of the drilling device for customers in the factory of TONISCO a discussion with an employee from the German TÜV (Technical Inspection Association) started.

The TÜV as an independent organization validate the safety of products in Germany. The history of the TÜV started back in the period of the industrialization when regional steam boiler supervision and revision associations were founded to increase the safety of steam engines and boilers.

Today the TÜV is separated in several groups like TÜV NORD Group or TÜV RHEINLAND Group and responsible for many safety certifications in all kind of technical areas. In the discussion with the member of the TÜV it became clear that the TÜV is the right partner for the certification of the TONISCO B30. /10/

The contact person from the TÜV is/was Mr. Dipl.-Ing. (FH) Stefan Luckmann.

4.1 Requirements

During the discussions with Mr. Luckmann of the TÜV the requirements that are to fulfill were defined.

First of all the bases of all requirements are the following Regulations:

- VdTÜV Merkblatt Armatur 100
- VdTÜV Merblatt Allgemeines 002
- Druckgeräterichtlinie 97/23/EG Artikel 3 Absatz 3 (DGR)(Pressure Equipment Directive)
- AGFW Arbeitsblatt FW 432
- BGR 119

Before the actual certification process the following points have to be clarified or ful-filled:

- the maximum of operation pressure
- the maximum of temperature
- the mediums which are in the pipelines (liquid/ gas and hazardous/ non-hazardous)
- the materials of all pressurized parts
- a detailed specification of the whole device
- calculation of all pressurized parts
- hazardous analysis
- current manual
- blueprints of all parts
- part list
- building description

For the audit/ certification at the company the following points have to be ensured:

- material certificate after DIN EN 10204 3.1
- calibration of all measuring equipment

During the actual certification testing the drilling device and the equipment is loaded with the 1.5 times of the maximum operating pressure and the tightness of the device is tested in different operating conditions.

To point out the certain steps in this certification process you can see in appendix 1 a project plan which was created in cooperation with Mr. Tobias Schengber.
There you can see which requirements and documents have to be fulfilled and which phases the certification process includes.

4.2 Materials

With the documents provided by TONSICO it was possible to create the following Table 1:

Table 1: Materials of pressurized parts

Part	*Blueprint No.*	*Material*
Feed wheel		
Feed nut		
Feed sleeve body		
Lower body		
Upper body		
Shaft		
Power screw		

You can see here the materials of all pressurized parts which are also need to be checked in a calculation. Noteworthy in this table is the material of the Feed Nut. This alloy of copper and tin is widely known as bronze. The main characteristics of Bronze alloys are their high strength, ductility, work hardening and corrosion resistance. Also they have very good slide properties, which is in the case of the Feed Nut in contact with the Feed Wheel very useful. These two parts are a so called motion connection or motion screw, means these two parts have to be able to move although the machine is working/drilling or not. The very good sliding properties ensure this requirement.

4.3 Manual

Although the manual of the drilling device have to be in German language the main parts are here briefly described in English. The complete German manual can be found in the Appendix 2.
The instructions consist of following parts:

Overview of main parts of TONISCO B30

The main and important parts of the drilling device are described and visualized in the final version that will be provided by TONISCO. The reader will get a brief impression of the device and gets to know the most important components for the actual drilling.

Service of TONSICO B30 + Daily service

Here are the maintenance intervals listed and the work that has to be done for the service is written down.
The difference between the daily and the annual service is the intensity of the work. It is described how to do the service (how to separate the parts of the device) and which tools and lubricants are to be used.

Preparing work

Before the actual drilling starts the operator has to fulfill some steps to prepare him, the mainline and the device for the drilling. These steps are like cleaning carefully all surfaces or lubricating the turning parts of the device. Furthermore the order of the assembly is described so that the operator can easily see how to handle the parts of the device.

Pressure testing

The pressure testing is of cause a part of the preparing work, but it is so important that it gets its own chapter.
Duration and sequence of the testing is described precisely and has to be respected.
To ensure to safety of the TONISCO B30 it is important to carry out the pressure testing before every use of the drilling device otherwise undiscovered damages in the structure or the sealing can cause leaks up to failure of the whole equipment.
The testing can be done with water or pressurized air. The testing with air is preferable.

Assembly of drilling device

This chapter explains how to finish the assembly of the device and which special features have to be considered.
Since it is possible to run the device with an electric or pneumatic unit there have to be some differences in the assembly and the working environment to be considered.

Branching under pressure

In this chapter the actual drilling is described. First it’s pointed out how to perform the drilling with the pilot driller. Which operating speed of the drive unit should be used and that the feed should be given in the beginning very carefully to ensure that the pilot drill get a centering. Thereafter the drilling with the hole saw is explained. The operating speed of the drive unit is even slower than in the drilling with the pilot drill.
Furthermore you can read in this chapter how to use the control cock to monitor the drilling process.

Use of shaft disassemble line

The disassemble line or also called shaft brake is to be used after the actual drilling. The drilling device should be removed but the pressure from the mainline would push out the shaft if the Feed Socket is unlocked from the force screws. To prevent this unwanted fact the disassemble line holds the shaft in its position and allows a slow push back of the shaft.

5 Calculation of pressurized parts

The proof that the device can withhold the forces of pressure and drilling it is necessary to have a look on the weakest parts of the unit.
These parts are:

- The retaining ring
- thread between Feed Wheel and Feed Nut
- thread between Feed Nut and Feed Socket
- the force screws
- the six jointing screws
- thread between Lower Body and the used Valve

These are the first parts that would brake if the forces and stresses are getting too high.

All the information concerning the threads and screws were taken out of the machine element book Roloff/Matek. /11/

5.1 Overview

In Table 2 you can find an overview about the calculations. The allowable loads, the occurring loads and the resulting safety values are listed.

Table 2: Overview strength calculation

No.	Connection	allowable pressure/force	occurring pressure/force	Safety
1	Retaining Ring	28,4 kN	-	3,922
2	Feed Wheel- Feed Nut	10 N/mm^2	-	1,783
3	Feed Nut- Feed Socket	130 N/mm^2	-	23,025
4	Power/Force Screw	176,532 N/mm^2 234,4 N/mm^2	- -	2.438 5,085
5	Jointing screw	11 kN	-	1.574
6	Lower Body- Valve	355 N/mm^2	-	76,67

5.2 Working Force

Since the pressure during the certification testing is 1.5 times as large as the maximum operation pressure I was calculating also with the higher pressure value (60bar).
The only part of the pressure that affects the structure of the device is the force that goes through the shaft.

$$A_{shaft} = \frac{\pi * {D_{shaft}}^2}{4} \qquad [mm^2] \qquad (5.1)$$

$$F_{pressure} = p * A_{shaft} \qquad [N] \qquad (5.2)$$

$$F_{drilling} = 3000N \quad (estimated)$$

$$F_{Work} = F_{pressure} + F_{drilling} \qquad [N] \qquad (5.3)$$

Where:

$F_{pressure}$	part of the force coming from the pressure in the pipeline.
$F_{drilling}$	estimated force that arises from the actual drilling process. The value is a really worst case estimation and will probably never occur.
$p = 60bar$	pressure in main line
D_{shaft}	diameter of shaft /12/

The Area of the Shaft is solved in equation 5.1, it is $\boxed{}$. According to this area the force coming from the pressure is (5.2). Finally in equation 5.3 the Force is calculated which occurs in the actual drilling process. This force must be endured by the whole structure. The working force is **7241N**.

5.3 The retaining ring

Due to the fact that the retaining ring is a standardized part the calculation consists only of the comparison from force value and the table value which shows what load the ring can resist.
In Table 9-7 in the Machine elements table book of Rolof/Matek you find the allowable force that ring can hold.

$$F_{ring.all} = 28.4kN > 7.241kN = F_{work} \tag{5.4}$$

$$s_F = \frac{F_{ring.all}}{F_{work}} \tag{5.5}$$

Where:

$F_{ring.all}$	allowable force that the retaining ring can withhold	[N]
s_F	safety factor of retaining ring /12/ /13/	

In comparison 5.4 you come to the conclusion that the ring is strong enough to withhold the load from pressure and drilling.

In calculation 5.5 the used retaining ring can withhold **3.922** times the load of the working force.

5.4 Thread between Feed Wheel and Feed Nut

This connection is as already mentioned a translation screw. The value of the allowable pressure in the thread is for this reason lower that the material could resist.
The capital "P" is the pitch of the thread, the "p" is the pressure in the connection.

$$p_{FeedNutInner} = \frac{F_{Work} * P_{FeedNutInner}}{l_1 * d_2 * \pi * H_1} \quad [N/mm^2] \tag{5.6}$$

$$s_F = \frac{p_{allFeedNut}}{p_{FeedNutInner}} \tag{5.7}$$

Where:

$P_{FeedNutInner}$	pitch of the inner Feed Nut thread
l_1	length of the thread
d_2	pitch diameter of thread
H_1	thread overlap
$p_{allFeedNut} = 10\ N/mm^2$	allowable pressure in translation thread of this material combination /12/ /13/

Equation 5.6 shows that the pressure in the thread is .

According to equation 5.7 the safety factor in this translation screw connection is **1.783**.

5.5 Thread between Feed Nut and Feed Socket

Since this connection is not a translation screw only the pressure in the thread is calculated and compared to the yield strength value.

$$p_{FeedNutOuter} = \frac{F_{Work} * P_{FeedNutOuter}}{l_2 * d_2 * \pi * H_2} \qquad [{}^{N}/_{mm^2}] \qquad (5.8)$$

$$s_F = \frac{R_{p\ 0,2}}{p_{FeedNutOuter}} \qquad (5.9)$$

Where:

$P_{FeedNutOuter}$	Pitch of outer Feed Nut thread
l_2	length of outer thread
$H_2 = \frac{P_{FeedNutOuter}}{2}$	thread overlap
d_2	pitch diameter
$R_{p\ 0,2} = 130\ {}^{N}/_{mm^2}$	yield strength of Feed Nut material CuSn10 /12/ /13/

The pressure in the thread has according to equation 5.8 a value of .

So this means that there is a safety factor of **23.024** in equation 5.9.

5.6 The power/ force screws

Since there are two screws the working force is spread and each screw has only the half value to withhold.

Force per screw:

$$F_i = \frac{F_{Work}}{2} \qquad [\mathrm{N}] \qquad (5.10)$$

First the holes of the Upper Body are checked for bearing stress according to the machine elements book Roloff/Matek Cap. 8 :

$$\sigma_l = \frac{F_i}{d * t_{min}} \qquad [\mathrm{N/mm^2}] \qquad (5.11)$$

$$\alpha_1 = \left(1{,}1 * \frac{e_1}{d}\right) - 0{,}3 \qquad (5.12)$$

$$\sigma_{l.all} = \alpha_1 * \frac{R_e}{s_m} \qquad [\mathrm{N/mm^2}] \qquad (5.13)$$

$$s_{F.\sigma} = \frac{\sigma_{l.all}}{\sigma_l} \qquad (5.14)$$

Where:

σ_l	bearing stress in hole
d	diameter of hole
t_{min}	smallest thickness in the direction of the bearing stress
α_1	offset factor
e_1	offset to edge of part
$S_m = 1{,}1$	minimum safety factor
$R_e = 355\ ^{N}/_{mm^2}$	yield strength of Upper Body material S355J2G5 /12/ /13/

The value of bearing stress in equation (5.11) is $\boxed{}$. Compared to the allowable stress calculated in (5.12) of $\boxed{}$, there is a safety of **<u>2.438</u>** in equation (5.14).

After checking the hole the screw is checked if it can withhold the Shear stress:

$$A_{ForceScrew} = \pi * \frac{d_{min}{}^2}{4} \qquad [mm^2] \qquad (5.15)$$

$$\tau_s = \frac{F_i}{A_{ForceScrew}} \qquad [N/mm^2] \qquad (5.16)$$

$$\tau_{s.all} = \alpha_a * \frac{R_m}{s_m} \qquad [N/mm^2] \qquad (5.17)$$

$$s_{F.\tau} = \frac{\tau_{s.all}}{\tau_s} \qquad (5.18)$$

Where:

$A_{ForceScrew}$	Area of the screw
d_{min}	minimum diameter
$\alpha_a = 0{,}44$	screw mechanical classifications factor
$R_m = 586\ {}^{N}/_{mm^2}$	ultimate strength of material X5CrNiMo17
s_m=1,1	minimum safety factor /12/ /13/

Equation 5.16 results in a shear stress of $\boxed{}$. The allowable shear stress of the screw is in 5.17 with $\boxed{}$ calculated. This is a safety factor of **5.085** (5.18).

5.7 The jointing screws

Each of the six jointing screws has to withstand a sixth of the working force. In this calculation the assembly prestressing force is calculated and afterwards compared to the allowable value according to the used screws. In FIGURE 10 you can get a very good overview which diameters and areas and meant in the following calculations.

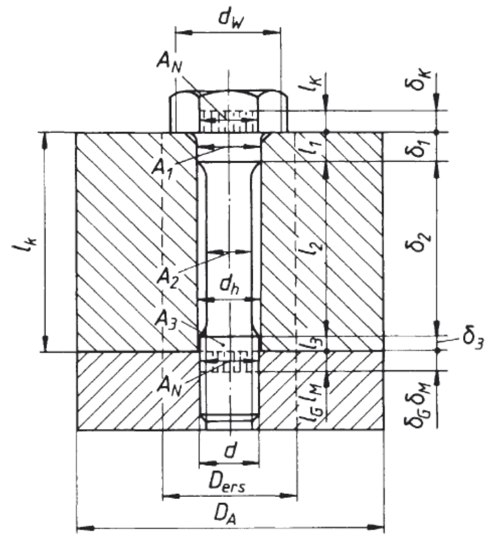

FIGURE 10: Overview screw connection /13, Chap.8 page 69/

The elastic compliance δ_T of the clamped parts is calculated:

$$A_{ers} = \frac{\pi}{4} * \left(D_A{}^2 - d_h{}^2\right) + \frac{\pi}{8} * d_w * (D_A - d_w) * [(x+1)^2 - 1] \qquad [\mathrm{mm}^2] \qquad (5.19)$$

$$x = \sqrt[3]{l_k * \frac{d_w}{D_A{}^2}} \qquad (5.20)$$

$$\delta_T = \frac{l_k}{A_{ers} * E_T} \qquad [\mathrm{mm/N}] \qquad (5.21)$$

Where:

l_k	grip length in the connection
$E_T = 210000\ {}^{N}/_{mm^2}$	young’s modulus of part materials
D_A	outer diameter of connection
d_h	diameter of hole
d_w	diameter of screw head area
A_{ers}	replacement area of a hollow cylinder with the same elastic compliance as the clamped parts
χ	additional factor for the replacement area
δ_T	elastic compliance of the clamped parts /12/ /13/

A_{ers}=

χ=

δ_T =

Then the elastic compliance δ_S of the screw is calculated:

$$\delta_S = \frac{1}{E_S}\left(0{,}4\frac{d}{A_N} + \frac{l_1}{A_1} + \frac{l_2}{A_2} + \frac{l_3}{A_3} + 0{,}5\frac{d}{A_3} + 0{,}4\frac{d}{A_N}\right) \qquad \text{[mm/N]} \qquad (5.22)$$

$$A_N = \pi * \frac{d^2}{4} \qquad [\text{mm}^2] \qquad (5.23)$$

Where:

δ_S	elastic compliance of the screw
$E_S = E_T$	young's modulus of screw
$d = 6\text{mm}$	diameter of thread
A_N	nominal diameter of screw
$A_1 = A_N$	according to FIGURE 10
$A_2 = A_N$	according to FIGURE 10
A_3	area of thread core
l_1, l_2, l_3	length of springy elements of screw; according to FIGURE 10 /12/ /13/

A_N =

δ_S =

Finally the assembly prestressing force can be calculated:

$$\phi_k = \frac{\delta_T}{\delta_T + \delta_S} = \qquad (5.24)$$

$$\Phi_{screw} = n * \Phi_k = \qquad (5.25)$$

$$F_z = \frac{f_z}{\delta_T} * \Phi_k \qquad \text{[N]} \qquad (5.26)$$

$$F_{VM} = k_A\left[F_p + F_d(1 - \Phi_{screw}) + F_z\right] \qquad \text{[N]} \qquad (5.27)$$

/12/ /13/

Where:

ϕ_k	ratio of force application on screw head and nut
$f_z = 8\mu m$	setting value Table TB 8-10a Roloff/Matek table book
$k_A = 4$	tightening factor
$n = 0{,}7$	load introduction factor
$F_p = \frac{F_{pressure}}{6}$	part of force from pressure for each screw
$F_d = \frac{F_{drilling}}{6}$	part of force from drilling for each screw /12/ /13/

In table 8-14 of Roloff/Matek table book you can find the value of maximum assembly prestressing force for a M6 10.9 screw:

$F_{sp.max} = 11kN$

According to equation 5.27 the assembly prestressing force is $\boxed{\quad\quad}$. Each screw can withhold a force of $\boxed{\quad}$.

$$F_{VM} < F_{sp.max} \tag{5.28}$$

$$s_F = \frac{F_{sp.max}}{F_{VM}} \tag{5.29}$$

Equation 5.29 results in a safety factor for the screws of **1.574**.

Additionally you have to say again that this calculation is a worst case scenario calculation. The tightening factor of 4 is very high and considers the assembly by hand with very high forces. If you would only compare the pressure in the thread with the allowable pressure then the safety factor would be much higher. /12/ /13/

5.8 The Thread between the Lower Body and the used valve

Here is the value of pressure stress in the thread compared with the maximum allowable.

$$p_{th} = \frac{F_{Work} * P_7}{l_{th} * d_{2.7} * \pi * H_{1.7}} \qquad [\mathrm{N/mm^2}] \qquad (5.30)$$

$$s_F = \frac{R_{p\ 0,2}}{p_{th.7}} \qquad (5.31)$$

Where:

P_7	Pitch of Lower Body thread
l_{th}	length of thread
$d_{2.7}$	pitch diameter of thread
$H_{1.7} = \frac{P_7}{2}$	thread overlap
$R_{p\ 0,2} = 355\ ^{N}/_{mm^2}$	yield strength of Lower Body material S355J2G5 /12/ /13/

The occurring pressure in the thread is $\boxed{}$ (5.30), so in equation 5.31 the safety factor for the lower body thread can be calculated as **76.67**.

6 Simulation with ANSYS and comparison with hand calculation

After doing all the hand calculations the important pressurized parts were modeled in the FEM-Software ANSYS to simulate the stress conditions and to compare the solutions of ANSYS to my hand calculations.
I worked hereby with the ANSYS-version that was provided by TAMK. This was an education-license with only the basic functions.

6.1 Background of FEM-Simulation

6.1.1 The Method of finite elements in software

During the last century the development in the engineering sector has made huge steps. Especially the computer based applications have a large share on this progress and opened many new possibilities. Before the computer were powerful enough everything had to be tested in real live, that means a prototype had to be built and several tests had to be carried out. This was expensive and time-consuming.

Still it is not possible to reproduce complex relationships with the theory of classic technical mechanics. Normally you create a simplified model of the problem, which is in most cases easier to solve. However, the solutions of these basic models should be considered carefully. To get more descriptive solutions the method of finite elements was invented.
Basically while watching the developmental history of FEM you notice that it is a relative young method. Essentially it was developed in the last 60 years equally of mathematicians and engineers.
In 1941 developed Hrennikoff a stick-model to solve 2-Dimentional problems. He used a matrix-version that is similar to today's finite elements method. Two years later Courant had the idea to divide the approaches for differential equations into a sequence of local approximations which is the basic idea of FEM. During the late 1950th and early 1960th the title "Finite Element Method" made his debut after Turner published a number of papers. The main idea in the finite elements method is to discretize or "break up" the domains or areas of interest into a collection of points and subdomains/ -areas called

nodes and elements. With the increasing number of powerful computers the FEM became more and more interesting for technical problems. /14/ /15/

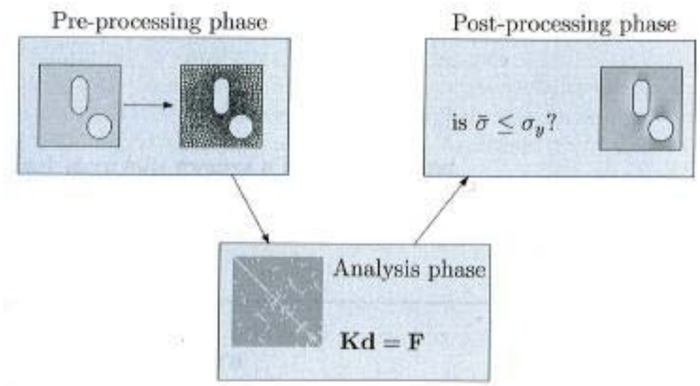

FIGURE 11: Flow chart of the finite element procedure /14/

As you can see in FIGURE 11 the procedure of the method can be divided into three steps: a pre-processing phase, an analysis phase and a post-processing phase. Nowadays you can say that there is a fourth step, the modeling of the component in a CAD-program (Computer aided design).

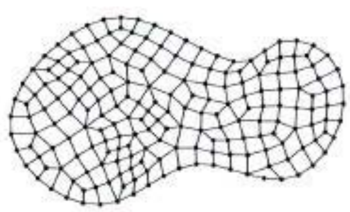

FIGURE 12: Finite elements mesh /14/

So in the phase of pre-processing the FEM-software generates a calculable FE-Model; a suitable mesh has to be created (FIGURE 12), assigning the element data, material properties and the boundary conditions and forces. In the second phase, the analysis, the system defined by the pre-processor can be numerically solved. An equation system of the form

$$K \times d = F \qquad (6.1)$$

$$Stiffness \times displacement = Force \qquad (6.2) \quad /14/$$

is then solved to the displacements.

Furthermore with the Hooke's law it is possible to calculate the stress. The last phase is for the output. The post-processor puts the solutions into colorful pictures so that the operator gets a quick overview about the conditions in the structure.

Although the computer is doing all the calculation, the operating engineer has to interpret the solutions. Because the computer is only a tools that does exactly what it is told to do, so if there is a mistake in the definition of the boundary conditions the whole calculating process can be wrong and the solutions are useless. /16/

6.1.2 Von Mises yield criterion

The term of equivalent stress is widely used in the strength theory in the engineering sector. It's a uniaxial stress and means that the equivalent stress has the same properties and effect to the material than the real, polyaxial stress.

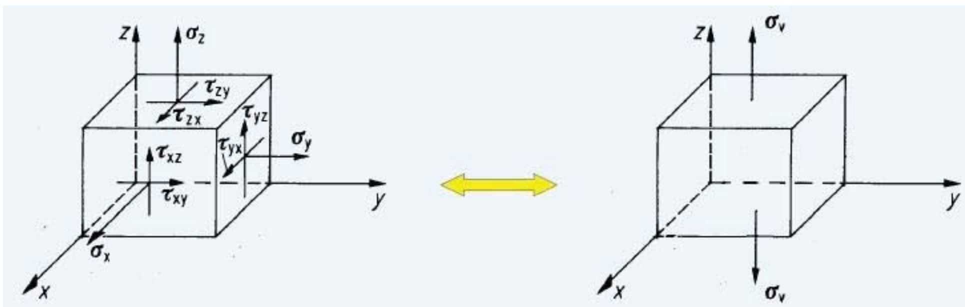

FIGURE 13: Real stress vs. equivalent stress /17/ (modified)

As you see in FIGURE 13 real stress consists of normal- and shear stress in all three dimensions, so it is impossible to compare the values with the material data. The equivalent stress transforms now these 6 stresses into only one stress which is comparable to the data from the tensile testing that is illustrated in FIGURE 14. /17/

FIGURE 14: Tensile testing /18/

There are several different equivalent stresses like the Von-Mises-yield-Criterion or the Tresca-criterion. The von-Mises-yield-criterion or von Mises stress is the most used one. One of the main assumptions in the von Mises stress is that the material behavior is isotropic. This means that the behavior due to the loads is in each direction of the structure identical. When this is fulfilled the stress value that is computed can be used to predict the behavior of the material under loads.

$$\sigma_v = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - \sigma_x\sigma_y - \sigma_x\sigma_z - \sigma_y\sigma_z + 3(\tau_{xy}^2 + \tau_{xz}^2 + \tau_{yz}^2)} \quad (6.3)$$

Where:

σ_v	equivalent stress
σ_x	normal stress in x-direction
σ_y	normal stress in y-direction
σ_z	normal stress in z-direction
τ_{xy}	shear stress in xy-plane
τ_{xz}	shear stress in xz-plane
τ_{yz}	shear stress in yz-plane

Equation 6.3 shows the actual formula of the von-Mises-yield-criterion for a three dimensional load case. According to FIGURE 13 you see the directions of the different stresses for a better understanding. /17/

When the value of the von Mises stress reaches the critically value of yield strength the material starts yielding. /18/

In my work with the FEM-Software ANSYS® Workbench I mostly used the von-Mises-yield-criterion to compare the simulations with my hand calculations.

6.1.3 Work with ANSYS®

ANSYS® is a computer-based software for engineering simulations invented in 1970 by Dr. John Swanson. It is possible to solve linear and non-linear problems out of several areas for example: structure-, fluid mechanics or electromagnetism. The software is in two versions available, the Classic version and the version called Workbench. The main difference between the versions is that the version Classic is mostly operated by keyboard commands and Workbench has a graphical user interface which makes it possible to set most of the commands by mouse.

In the following only the work with the version Workbench is described. /19/

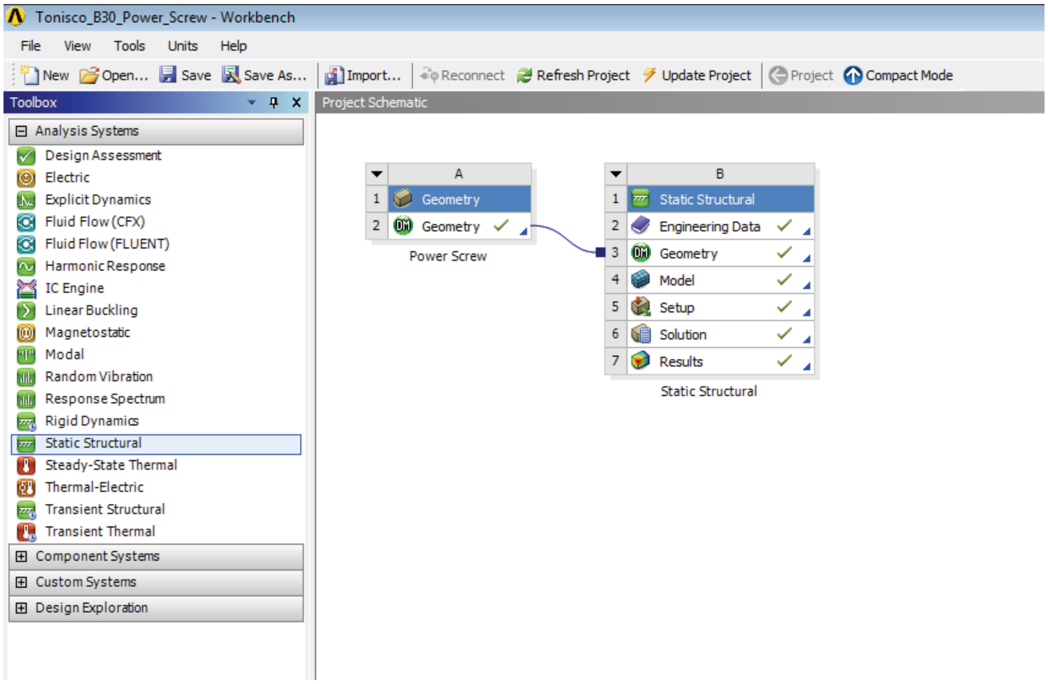

FIGURE 15: ANSYS® User Interface (screenshot)

Before the simulation with ANSYS® Workbench started all of the analyzed TONISCO B30 parts were modeled in the CAD-software Autodesk Inventor Professional 2012. After that the part geometries were imported into a static-structural analysis in Workbench, as you can see in FIGURE 15: ANSYS® User Interface. Now the engineering data could be adjusted, for instance the material data or the Young's modulus. If necessary in the Design Modeler some extra properties for the geometry could be added.

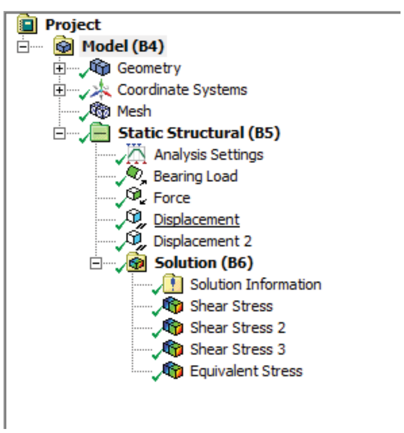

FIGURE 16: ANSYS® Mechanics interface (screenshot)

In the analyzing sub program called Mechanics the work continues. In FIGURE 16 you see the navigation structure of the program Mechanics. First the allowable displacements or support definitions are to define. Then the loads will be allocated to the geometry.

A big step is thereafter to generate the mesh. The nodes in the mesh represent the edges of the elements for the finite elements method. The most important setting for the mesh is to define the size of the elements. The amount of elements is linked with the size of elements. The smaller the elements are the higher is the amount of them. Also the solution will be better with a higher amount of elements but the computing time will increase. The operator has to balance this fact in order to get a qualitative result.

The software in calculating between these nodes the displacements and from that as described in equation 6.1 the stress in the structure.

Are all the predefinitions done the operator have to set the different solution-options. The above-described Von-Mises solution is a possibility but also shear or only normal stress.

When the solution is done the operator has to interpret it and decide if the solutions are reasonable or not. If it’s not the predefinitions have to be checked and change.

6.2 Simulations

Although for the certification process only a written calculation of the pressurized parts was necessary all these parts were simulated in ANSYS® and compared to the hand calculations.

Because threads are not shown in ANSYS® they are missing in the pictures.

6.2.1 Feed Wheel

Since the retaining ring is the weaker part in the connection of Feed Wheel and ring there is no hand calculation of the Feed Wheel. But to be sure the Feed Wheel was also simulated.

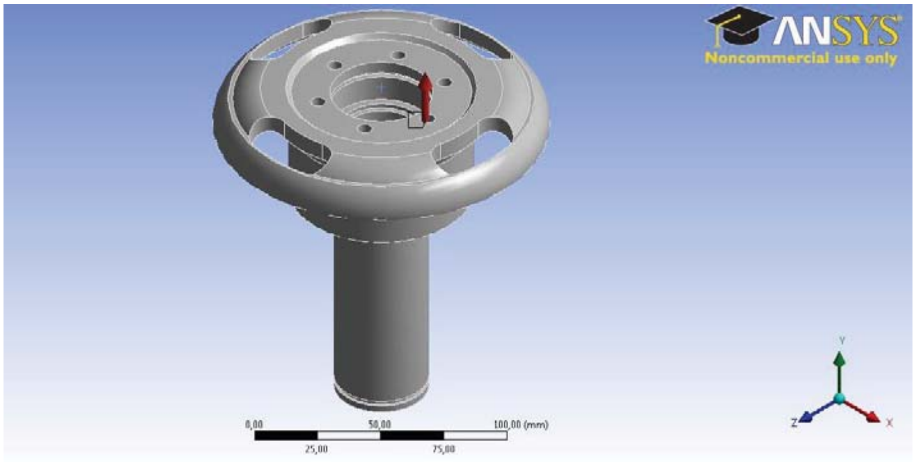

FIGURE 17: Force on Feed Wheel

The working force from pressure and the drilling is affecting to the upper wall of the groove (not visible in FIGURE 17). The displacement seen in FIGURE 18 is set to the thread of the Feed Wheel.

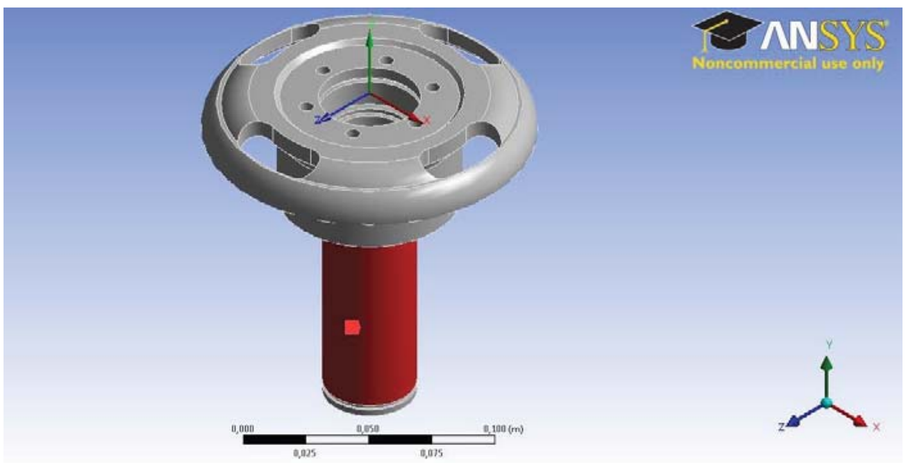

FIGURE 18: Displacement Feed Wheel

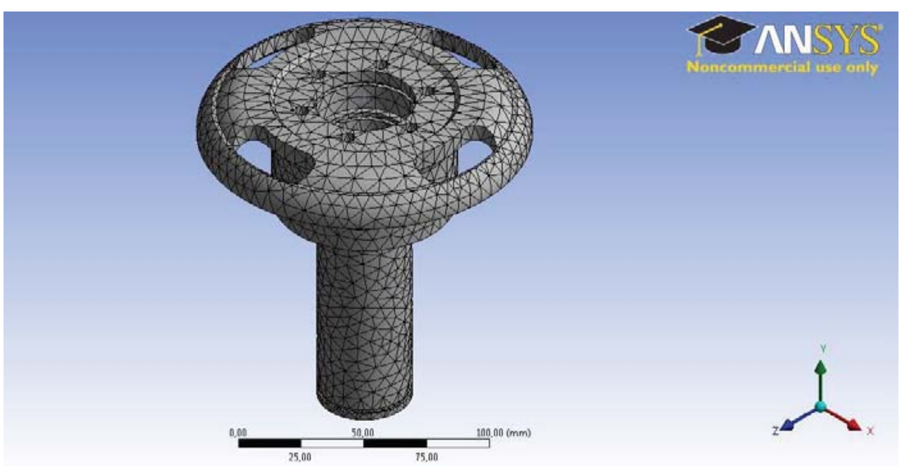

FIGURE 19: Feed Wheel Mesh

The Mesh for the calculation is shown in FIGURE 19. It was checked that the Mesh in and at the groove and also around the thread was complete and suitable.

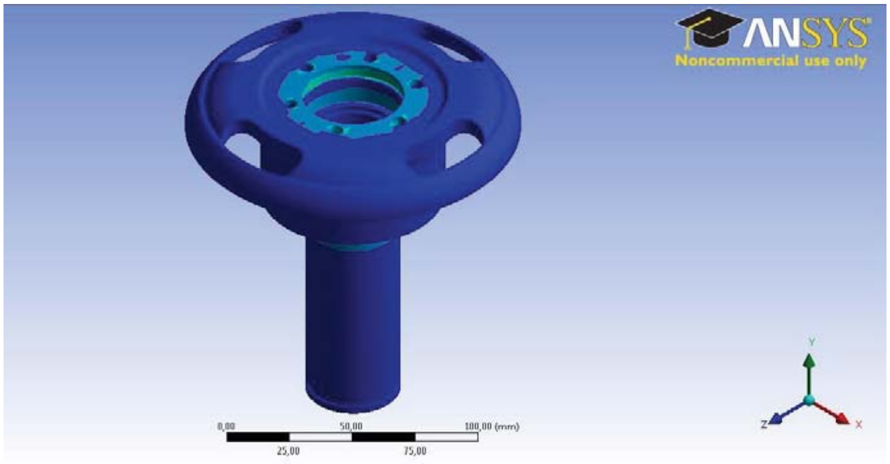

FIGURE 20: von-Mises stress Feed Wheel

The simulation calculates a maximum stress of seen in FIGURE 20. The yield strength of S355J2G3 is 355 N/mm^2/13/. Hence the structure is able to withhold more than 5 times the load of drilling.

6.2.2 Feed Nut

The Feed Nut is made of CuSn so the material data had to be changed.

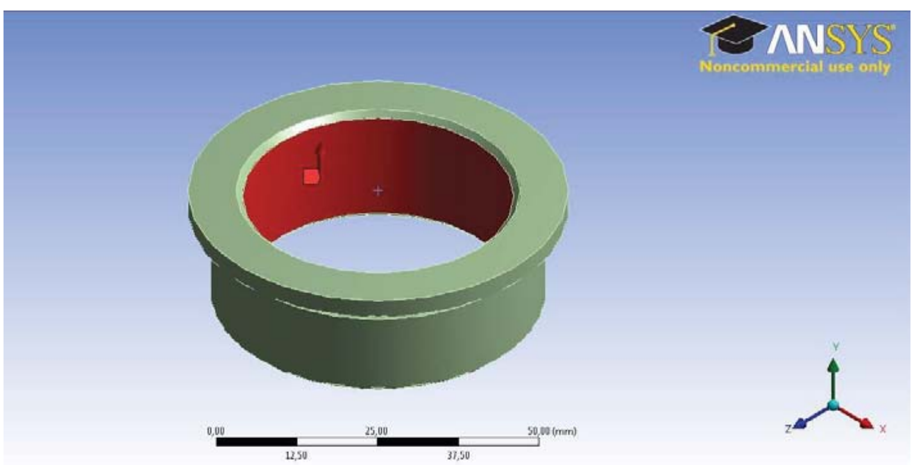

FIGURE 21: Force Feed Nut

FIGURE 21 shows how the force is affecting to the inner thread, the part of the translation screw.

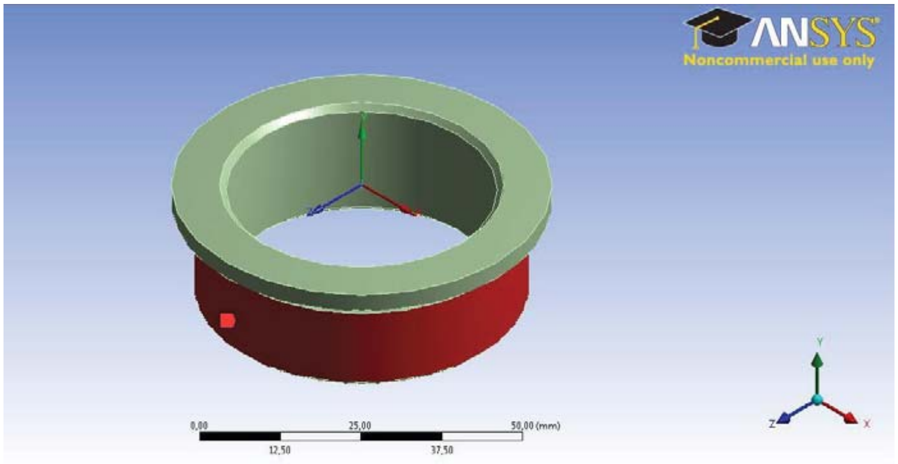

FIGURE 22: Displacement Feed Nut

With the outer thread the Nut is placed in the Feed socket because of that the displacement is on this thread (FIGURE 22). The movement to the y-direction is suppressed.

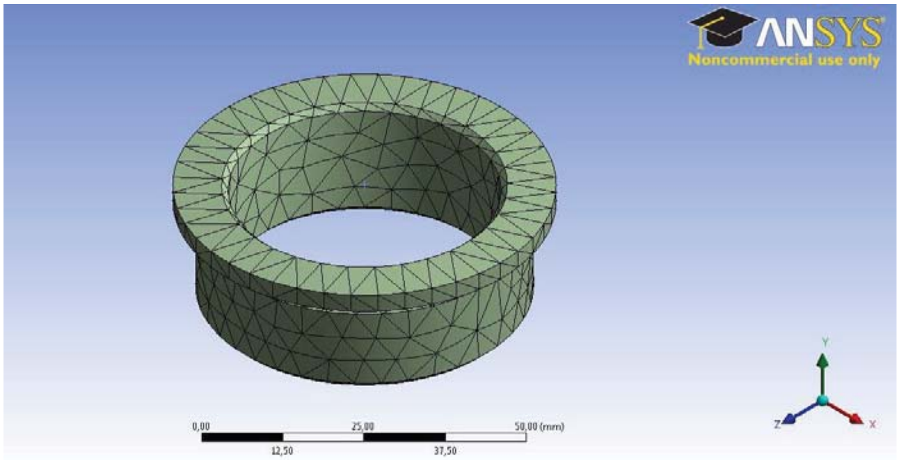

FIGURE 23: Mesh Feed Nut

The Mesh for the Feed Nut has no special properties as you can see in FIGURE 23. The connections of the elements in the area of the inner and outer thread are clear and regular which was necessary for a useable solution.

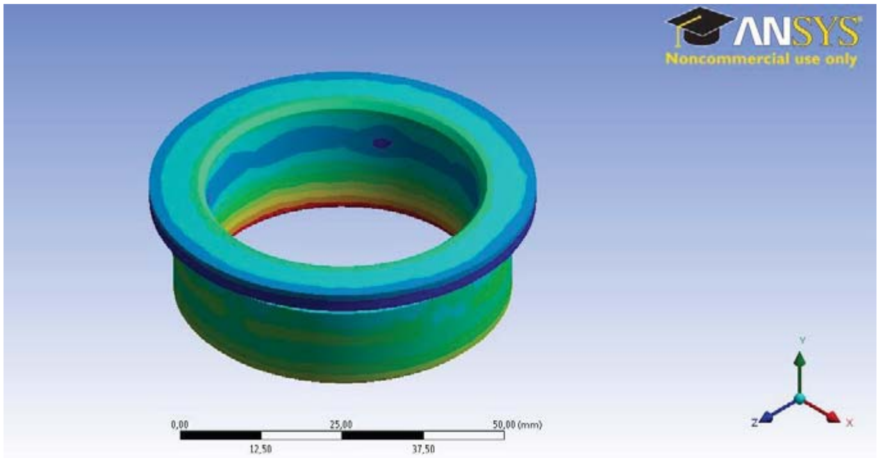

FIGURE 24: von-Mises stress Feed Nut

FIGURE 24 shows the distribution of the stress in the Feed Nut. The maximum stress is . To compare the distribution with the hand calculations you have to focus on the inner thread. Here is the average stress ; the written calculation showed a stress of , this seems plausible.

6.2.3 Upper Body

In the upper Body two different kinds of loads are occurring. On one hand there is the working force that is transmitted through the power screws into the body (FIGURE 25) and in the inner part of the body is also the pressure from the main line (FIGURE 26).

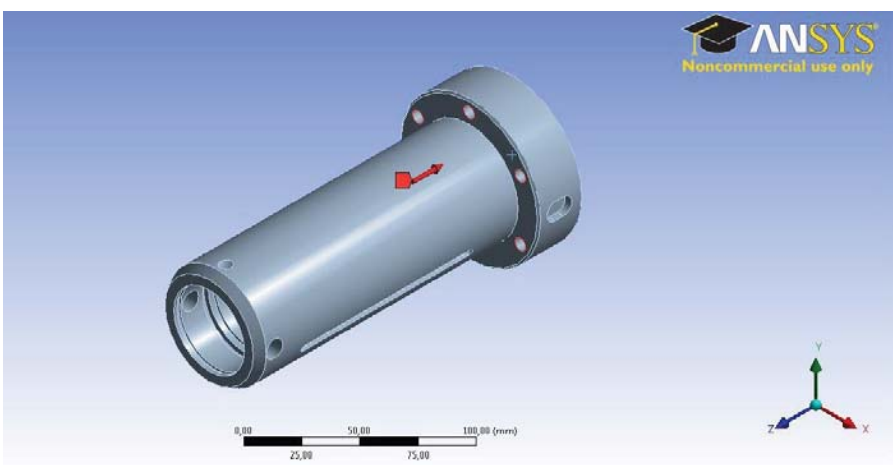

FIGURE 25: Force Upper Body

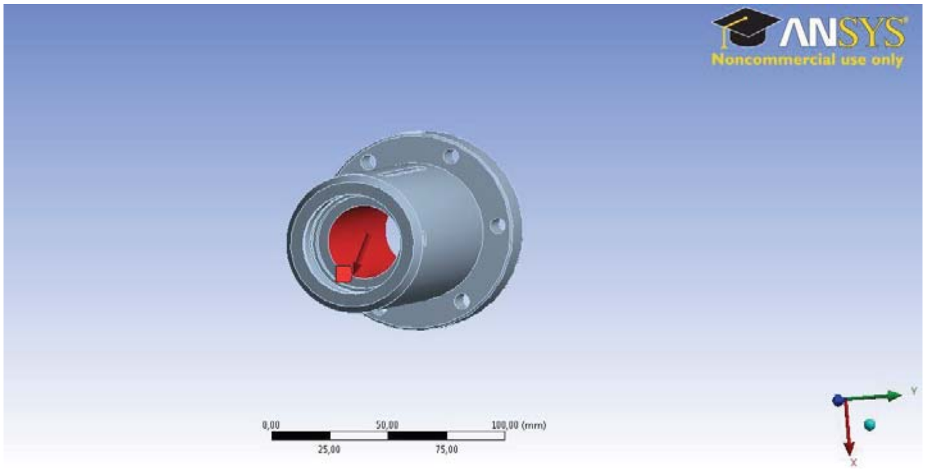

FIGURE 26: Pressure Upper Body

In the simulation the displacement is defined in the threads where the power screws are normally situated (FIGURE 27). The load is placed at the holes for the jointing screws. This is actually the other way around than in real load case but for the calculations in makes no difference.

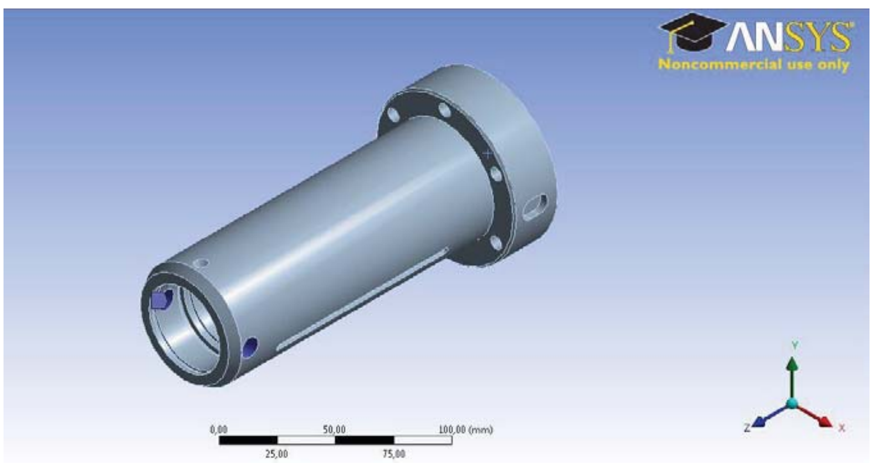

FIGURE 27: Displacement Upper Body

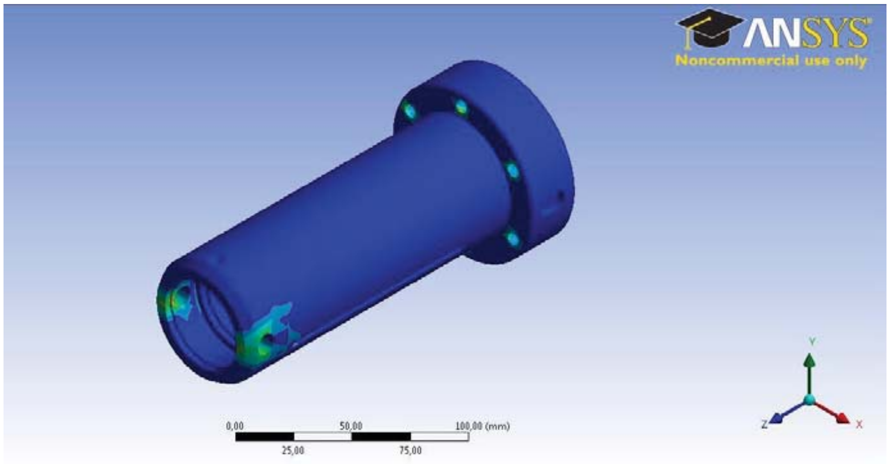

FIGURE 28: von-Mises stress Upper Body

In FIGURE 28 you can see the value of the van-Mises stress. In the hand calculations only the shear in the holes was calculated so there are no results to compare. But you can compare the simulated result to the yield strength of the Upper Body material, the yield strength value is 355N/mm^2 /13/ so the structure is able to withhold the stress according to the simulation.

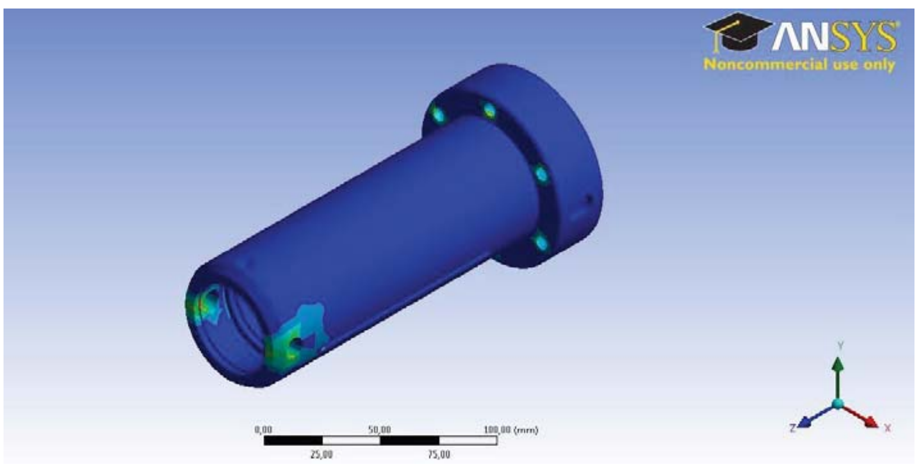

FIGURE 29: Shear stress Upper Body

The shear stress at the holes for the power screws is around a value of (FIGURE 29). The hand calculated stress value is . That seems also plausible.

6.2.4 Lower Body

The working force is affecting in the threads of the boreholes for the jointing screws (FIGURE 30). Additionally the pressure from the main line is applied on the inner wall of the part (FIGURE 31).

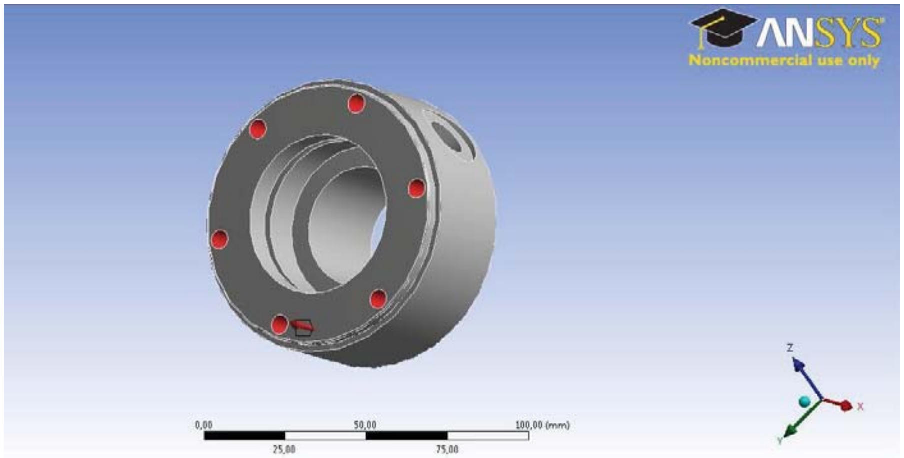

FIGURE 30: Force Lower Body

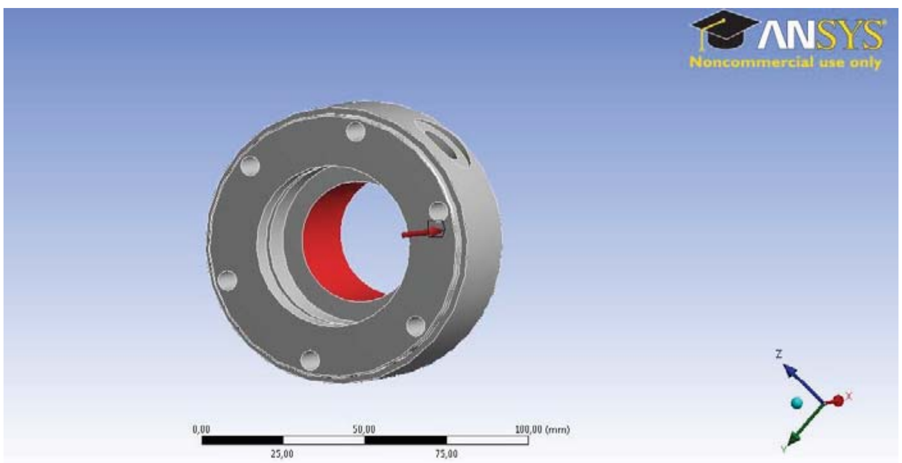

FIGURE 31: Pressure Lower Body

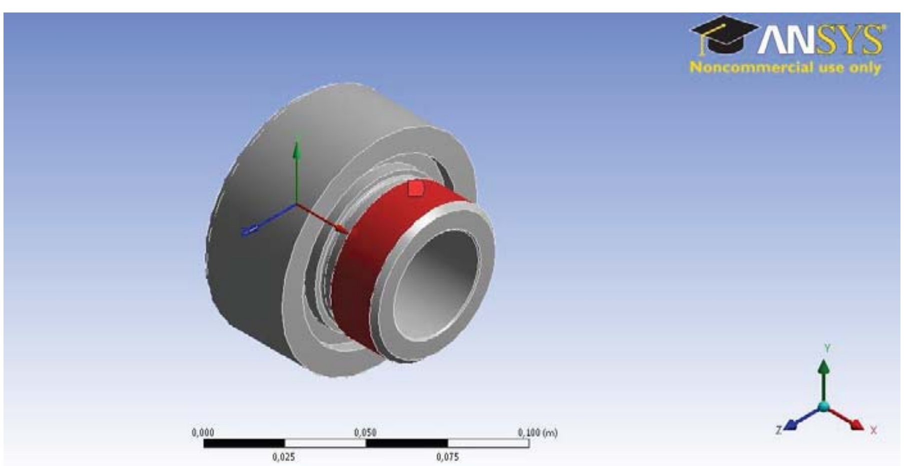

FIGURE 32: Displacement Lower Body

In the Lower Body part the displacement setting is on the thread of the part (FIGURE 32).

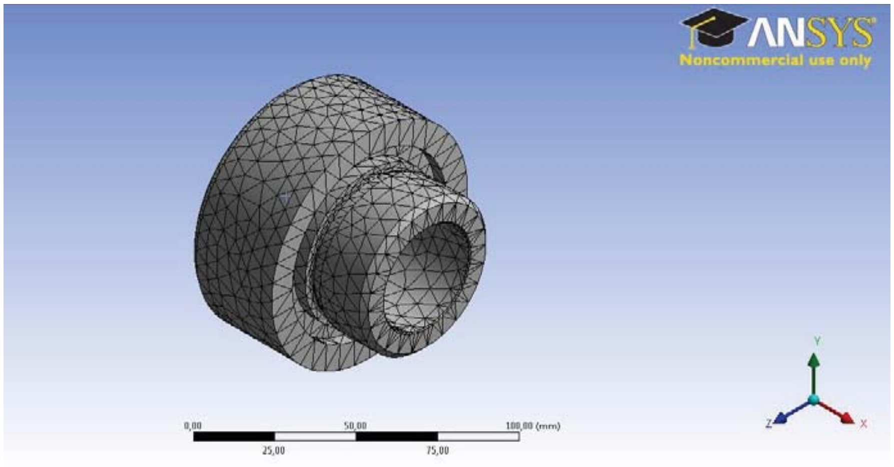

FIGURE 33: Mesh Lower Body

In FIGURE 33 the mesh on the Lower Body is illustrated. It is uniformly so you can expect a good solution.

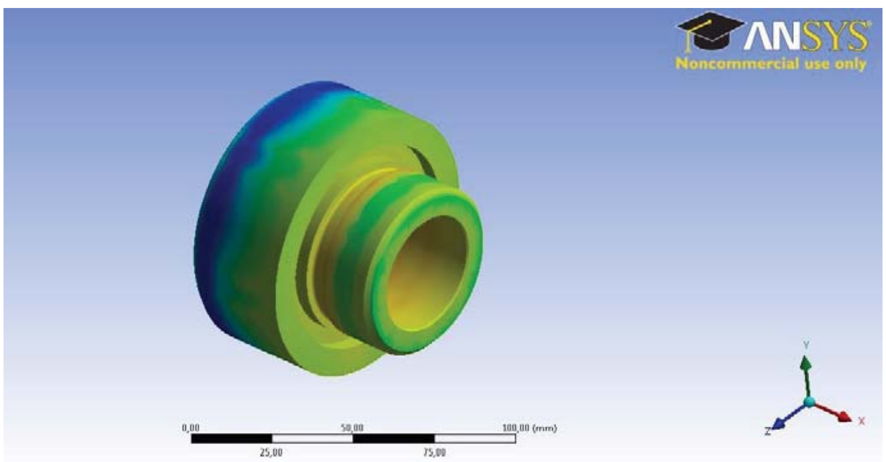

FIGURE 34: von-Mises stress Lower Body with pressure

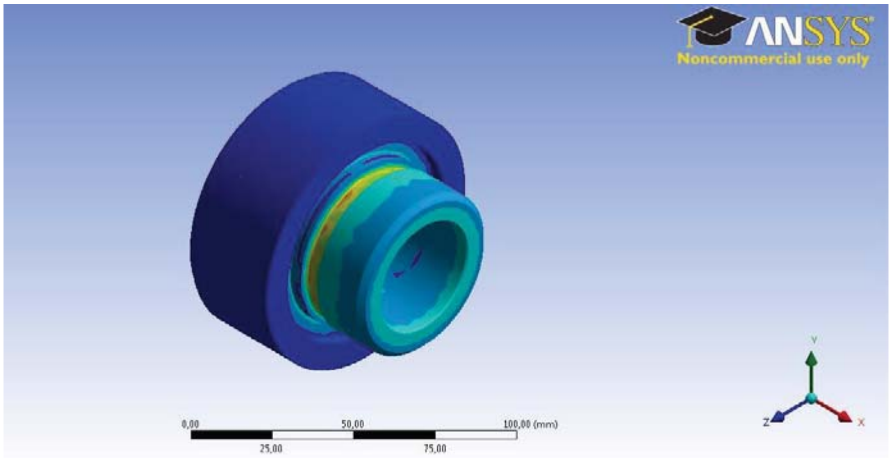

FIGURE 35: von-Mises stress Lower Body without pressure

FIGURE 34 shows the solution of the simulation with all loads, working force and pressure. The average stress value in the outer thread is around [redacted]; however this result is not comparable to the written calculation because the written calculation only considers the working force. Hence FIGURE 35 is a simulation only with the working force. The stress in the thread is from [redacted]. The calculated value is 4.6N/mm^2. Here the values are very similar and illustrate that the calculations are plausible.

But also the stress value from both loads would be resisted by the structure because the yield strength value of 355N/mm^2 /13/.

6.2.5 Power Screw

Because of the design of the Power Screw-Feed socket connection the screw is calculated only on pure shear stress. A special feature you can see in FIGURE 37. Since ANSYS® is not able to recognize threads from the CAD-Software it was necessary to define in the Design Modeler the length of the thread and so separate the front part from the rear part of the cylinder without removing any material.

FIGURE 36 shows where the working force is defined.

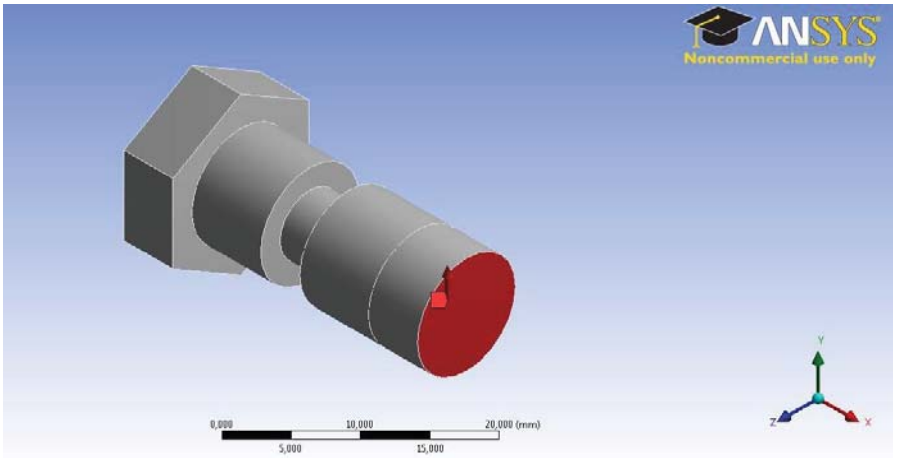

FIGURE 36: Force Power Screw

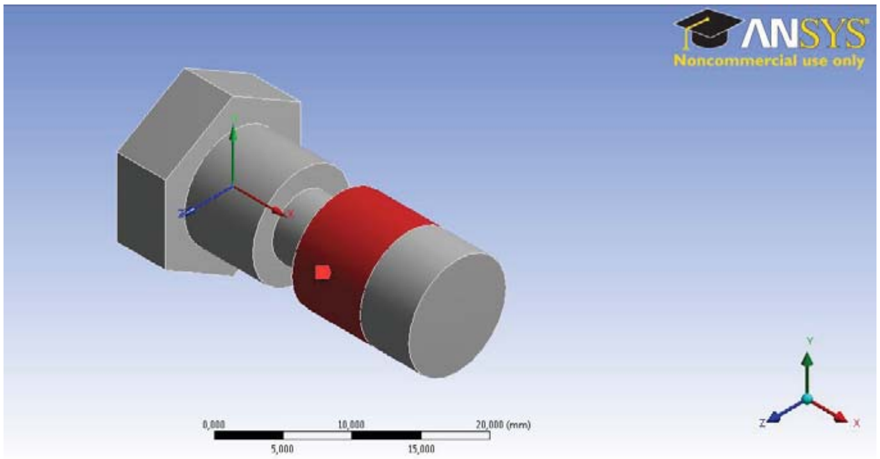

FIGURE 37: Displacement Power Screw

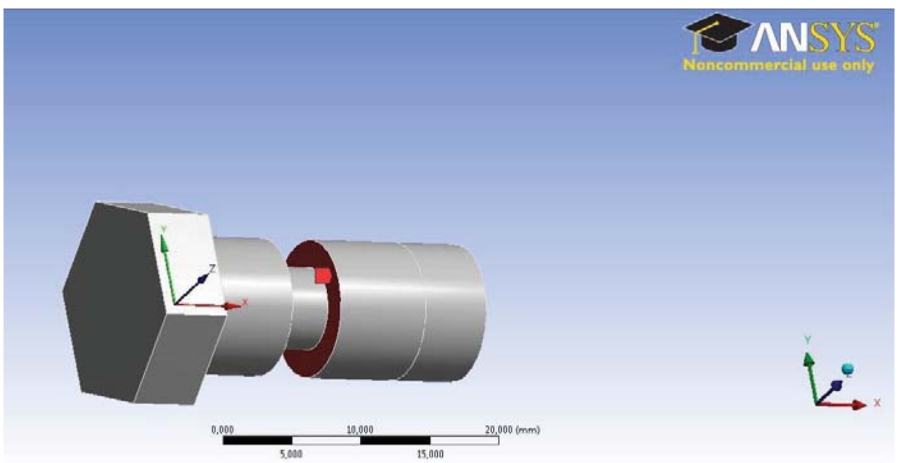

FIGURE 38: Displacement 2 Power Screw

FIGURE 37&38 show the displacement settings for the screw. The cylindrical displacement suppresses the movements in y- and z-direction and the displacement 2 prohibits movements in x-direction.

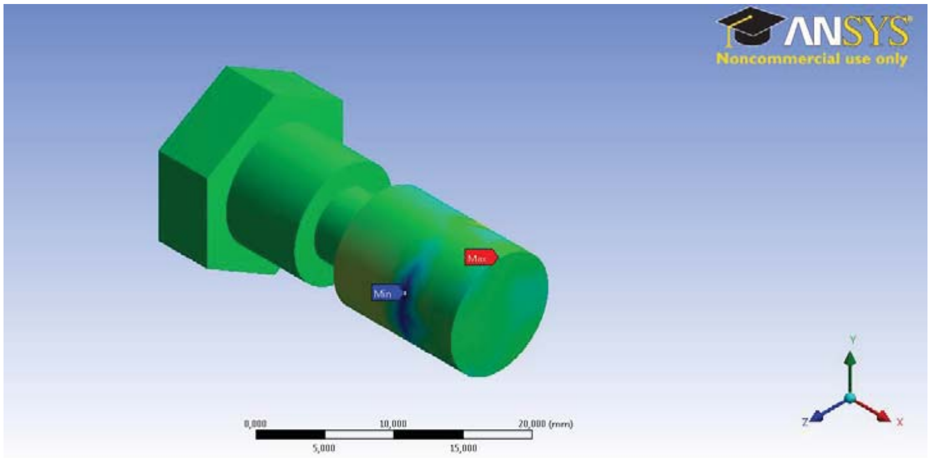

FIGURE 39: Shear stress YZ-Plane Power Screw

The highest value of shear is in the YZ-Plane (FIGURE 39) because it's parallel to the loads.

When comparing this value to the hand calculations it is too high. The reason is that in the written calculation only the average τ_{sm} over the whole area was calculated. In FIGURE 40 you can see the real course of the shear τ_{zy}.

In a round rod the maximum shear tension is:

$$\tau_{zy\,max} = {}^{4}/_{3} * \tau_{sm} \qquad (6.4) \quad /17/$$

So if you multiplying the calculated value with factor of 4/3, the calculated and simulated value are getting similar and plausible.

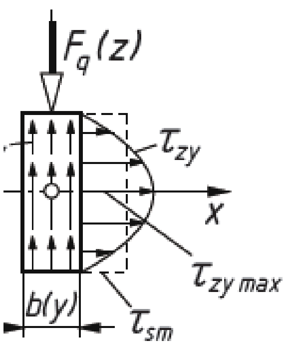

FIGURE 40: Shear Stress course /17/

7 DISCUSSION

The very first aim of this thesis was to get a certificate for the German market to increase the sales. During the discussions with all members of this thesis project, Kaarlo Nisso as representative from TONISCO System Oy, Mikko Ukonaho as my supervisor from TAMK and me, the topic was limited to find out the right authority, creating the contact, discuss the requirements in order to get the certificate and fulfill some of these requirements; therefor mainly the calculations and a simulation with the program ANSYS® Workbench.

The first part to find the right partner for the certification was more or less the time demanding task in this project. Because there were no experiences with some of the authorities it was necessary to contact all of them and ask for the responsible employee to discuss with. Waiting for some response or even not getting any answer was a little bit frustrating but after finding the TÜV as the right certification partner the whole process was going on really fluently and straight to the set target.

In the work with the calculation and simulation it was very interesting to connect the learned theory with a practical project. Furthermore it was very fascinating to see which problems can occur and how to find possibilities to solve them. Especially during the simulations it was very informative to see how small mistakes in the defining of boundary conditions are leading to big differences in the solutions comparing to the right results.

Very exciting was the creation of the German manual for the drilling device. Describing in precise and proper German language the features and functions of the device was also a challenging task in this thesis.

Finally I would say that the process of achieving the certificate is on a good way. All the given requirements are fixed and it shouldn’t be a problem to fulfill them step by step according to the project plan in the appendix 1. The structure of the TONISCO B30 is more than able to withstand the loads in the normal work environment and also during the certification testing, what the calculations and the simulations have both confirmed.

In the end I would like to thank Kaarlo Nisso and Mikko Ukonaho for all the support during this project.

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FIGURES

APPENDICES

Appendix 1. Project Plan Certification

Confidential

Herstellung einer Abzweigung an Hochdruckleitungen

-Verwendung der TONISCO B30 ausgestattet mit TONISCO Anbohrsperre