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DESIGNING AND MODELING OF PISTON IN COMBUSTION ENGINES

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A. - Acronyms

BDC: Bottom death centre

CAD: Computer aided program

CI: Combustion ignition

TDC: Top death centre

SI: Spark ignition

a: Coefficient of linear expansion

d: Pin diameter

d_0: Primary pin diameter

D: Diameter of the cylinder

D_p: Primary diameter of the cylinder

E: Elastic modulus

f: Deflection of the pin

F_t: Area of the top of the piston

g: Thickness of the crown

g_e: Specific fuel consumption

h_{c1}: Convection coefficient in the cylinder walls

J: Moment of inertia

k: Pressure in the pin

k_g: Maximum allowable stress

k_{dop}: Permissible grown pressure

l: Length of the connecting rod

l_0: Primary length of the pin

l_1: With of the connecting rod

l_2: Length of the pin

M: Bending moment

N: Normal pressure

N_e: Specific power per cylinder

p: Pressure

P: Gas forces

p_{max}: Maximum pressure

q: Heat transfer

r: Radius of the crank

T_{c1}: Temperature of the convection coefficient

T_{s1}: Temperature of the first sealing ring

T_{s2}: Temperature of the second sealing ring

T_{sc}: Temperature of the scrap ring

W_d: Calorific value

α: Connecting rod angle

σ_c: Thermal stress

σ_g: Load stress

σ_{gc}: Total bending stress of the pin

σ_{go}: Bending stress of the pin due to the ovalization of the pin

σ_{gu}: Bending stress of the pin due to the deflection of the pin

σ_x: Radial stress

σ_y: Shear stress

σ_c: Thermal stress

λ: Conductivity of the material

τ: Shear stress of the pin

1. Aim and scope of the work

1.1 Aim of the project

The purpose of this final project is to design and calculate a piston for a large two-stroke diesel engine. They are used in railroad, marine or stationary services. They are very big and heavy, all the parts have to be very well studied and calculated.

Modelling process has different parts but it has to be understood what we want to achieve, the optimal shape of the piston to obtain maximum performance. The piston itself has four different main parts that should be taken into account: the crown, piston rings, hubs and the skirt.

The crown takes a very important part in the whole process. It is the part that supports most of the loads and heat. As a consequence is the one that suffers the hardest deformation. The piston object of study is very large, it has to be thought a refrigeration system inside the crown to relief some of the heat stresses.

As the object is to see under certain conditions how the piston works, at first it has to be understood how work the different parts mentioned before. So that we can get the optimal and correct results we are looking for.

When working, the piston is subjected to a treatment of thermal stresses and force stresses. Both of them acting against the piston and causing a deformation on it. At first, the working conditions have to be very well defined. That way the results obtained will be the most accurate to reality as possible.

The materials used for the construction of it also play a very important role in the whole process. Given different materials the deformations and the heat flow will be, of course, different. In the modelling process has to be chosen the one that gives better results taking into account different factors.

1.2. Scope of the project

The way to achieve these purposes is using CAD programs for modelling. In this case Solid works will be used. Starting with basic initial shape to start working and then adding new features to see how piston reacts. Everything has to be implemented to get the best result.

As it is a large piston it has oil refrigeration. This is very important because will help cooling the piston and reducing the deformation. This has to be implemented in the CAD program to see the results afterwards.

Once the initial shape is done the next step is to simulate under certain conditions. That is one of the most important parts of the project. The heat transfer has to be very well defined. There is the heat flow inside the piston and then conduction in the piston rings. The performance of the rings will establish somehow the deformation of the piston.

Once the initial shape is done, is time to simulate it in a finite element program. There are a lot of programs in the market but in this case ANSYS will be mainly used. Two different simulations will take place. Loads and thermal stresses will give us the result of the total deformation. None of them can be simplified because both values are very high.

It will be an iterative process until the optimal shape is obtained. Every time something is changed it has to be analysed again to see if it works as expected or there are some issues that have not been taken into account. All constrains used and the different boundary conditions used for the simulation have to be revised, in order to make sure they are as similar to reality as possible.

The weight of the piston is very important. The finite element simulation will give us ideas of which are the parts that are not submitted to great stress. This will allow us to reduce the material in these areas. A lighter piston has lots of advantages. First of all and most important is the price. The less material used the less cost has. In terms of efficiency a lighter piston gives better performance.

Finite element simulation will give us is the critical parts. Another way to say is the parts that are submitted to the worst working conditions. These are the most important. They have to be very well studied and analysed. If they exceed the maximum stress that the material can support, it will be necessary to add more material or if it is not possible redesign the mentioned part.

The objective is, in the end, achieve a piston that is very well calculated and that responds as good as possible to reality. Making an especial effort in the two points mentioned before.

2. Recognizing of the problem in literature and technical solution

Main problem expected to be found in the design of the large piston is the deformation, due to pressure and temperature. The heat coming from the exhaust gases will be the main reason of deformation.

2.1. Data of the engine

The technical data is:

Specifications of the engine

Description	Value
Configuration	Two-stroke diesel engine cooled by air
Max pressure ‘p_{max}’	5 MPa
Temperature in combustion chamber ‘T’	900 K
Bore ‘D’	300 mm
Rotation speed ‘n’	750 rpm
Specific fuel consumption ‘g_e’	0.3 kg/kWh
Power ‘N_e’	300 kW per one cylinder
Calorific value (gasoline and oil) ‘W_d’	42000 kJ/kg
Piston cooling	By oil maximum temperature 500K

2.2. Cooling system

Piston engines, and, in particular, internal combustion engines, are often cooled using lubrication oil. This is traditionally achieved by spraying lubrication oil onto the piston, to facilitate cooling in the lower part of the piston crown surface.

In big engines this system becomes inefficient due to the huge amount of heat to be transferred. To correct this problem large amount of oil sprayed is required. This requires additional components such as larger than necessary oil storage tanks, reducing engine’s power to weight ratio and increasing manufacturing an operational costs of the engine.

Accordingly, heat transfer between a piston and the lubrication oil is not uniform across the engine. This causes thermal gradients and strains within the engine potentially leading to the formation of cracks.

To solve this problem a uniform cooling system is needed. Including a tortuous flow channel in the piston crown, uniform cooling is achieved. This is made to increase the contact surface between the lubrication oil flowing through the cooling chamber and the piston head and also, prolong the contact time period during which the lubrication oil contacts the piston head.

The lubrication oil is injected into the cooling chamber through a series of fluidly coupled channels embedded in a crankshaft and a rod connecting the piston head to the crankshaft. After heat exchange with the piston head in the cooling chamber, the lubrication oil is returned to the crankshaft.

As a result, heat transfer is conducted uniformly in every piston. This can significantly reduce the chance of engine failures caused by thermal gradients and strains within the engine.

There are multiple solutions to the description given before.

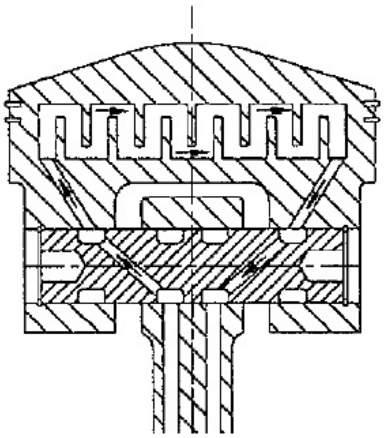


Fig.2.1. Sealed lubrication

In Fig. 2.1 it can be seen a first solution. It is a schematic flow diagram of an embedded cooling system used by a piston engine. This lubrication system is sealed. The oil goes in and out through the connecting rod.

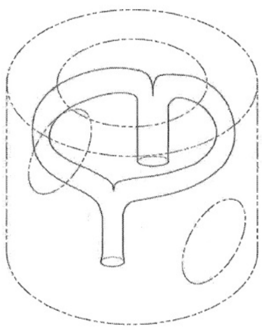


Fig. 2.2. Sealed cooling system surrounding the piston head

In Fig. 2.2 there is another sealed cooling system. The difference between this one and the one shown in Fig. 2.1 is that this one only goes by the exterior area of the piston crown. It is a good approach but it does not treat the interior part of the piston crown. This could lead to thermal gradients and strains in the inner part of the piston.

2.3. Piston rings

Generally, a compression ring and an oil ring are attached as a set of a piston ring to a piston that performs reciprocating motion. The compression ring possesses a function to prevent a phenomenon called blow-by. High-pressure combustion gas flows into the crankcase from the combustion room. On the other hand, the oil ring mainly possesses the function to suppress the excess of lubricant on the inner wall of the cylinder liner.

The main function of the piston rings is to seal the combustion chamber from the rest of the engine. There are some considerations that must be done before designing the piston rings.

The first piston ring should not be too far back the piston head. This increases the volume of the gap between the piston and the cylinder walls, this increases the secretion of hydrocarbon compounds into the exhaust gases.

Number of rings, from which depends on the height of the annular part is related primarily to the speed of the engine and the pressure of the combustion. At medium speeds, the first ring takes over the 75% of the entire pressure.

The choice of the number of rings should be the result of careful analysis, with one hand, depends on to the gas that passes into the crankcase should be the minimum, on the other, the number of rings determines the mass of the piston, engine height and friction losses.

The next figure shows some examples of some designs in piston rings and also the flowing of the oil while working.

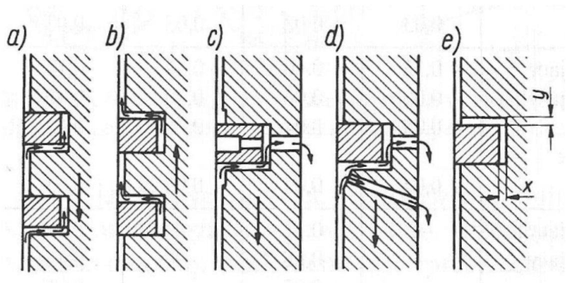


Fig. 2.3. Examples and designs of piston rings

a), b) pumping oil through the rings, c), d) scraping and oil flow towards the piston, e) clearance between the piston ring and the piston itself

During the movement of the piston towards the BDC friction rings occupy the position shown in fig. 2.3.a. with the craped oil filling the clearance existing between the rings and grooves in the piston. During the movement to the TDC oil moves over rings as fig. 2.3.b. Oil, which thus moves to the side of the TDC, is finally burned in the combustion chamber.

Burnt oil is not only an economical issue, but also can significantly increase the emissions of toxic components in the exhaust gases, particularly in the form of hydrocarbons. In addition, burning oil creates a sludge and carbon deposits in the combustion chamber, which easily lead to SI engines to self-igniting.

The intensity of the phenomenon of the oil pump may be decisive to decide the axial clearances. Approximate clearances are shown in the next table

Clearances depending on the piston diameter

Engine	Bore	Type of ring	Four-stroke engines		Two-stroke engines	
			Air cooled	Water cooled	Air cooled	Water cooled
Spark ignition engine	100-170	1. Sealing 2. Sealing Scrapper	0,08 0,07 0,04	0,06 0,06 0,04	0,12 0,08 0,04	0,10 0,08 0,04
Diesel engine	100-150	1. Sealing 2. Sealing Scrapper	0,13 0,09 0,04	0,10 0,08 0,04	0,17 0,10 0,05	0,14 0,10 0,05
Diesel engine	150-200	1. Sealing 2. Sealing Scrapper	-	0,12 0,08 0,04	0,20 0,12 0,05	0,16 0,10 0,05
Diesel engine	200-250	1. Sealing 2. Sealing Scrapper	-	0,14 0,10 0,04	0,23 0,14 0,05	0,16 0,10 0,05
Diesel engine	250-300	1. Sealing 2. Sealing Scrapper	-	-	0,25 0,15 0,05	0,18 0,12 0,05

2.4. Materials

Adverse working conditions make the requirements of the materials used in the piston very wide and diverse. Materials used in the piston manufacturing can be divided in the following groups:

- Cast iron (non-alloy and alloy steels)
- Aluminium alloys
- Special steel

Cast iron used in piston is usually perlite structure with separate laminas. When alloyed, more fine-grained structure is obtained and will improve the mechanical properties of the material.

The advantages of cast iron are: good sliding properties, high abrasion resistance, small decrease in strength and hardness in high temperatures, small coefficient of thermal expansion. Disadvantages are: high density and small coefficient of heat conduction. Hardness in cast iron should be in the range 180 – 240 HB and also, should be adapted to the hardness of the rings and cylinder walls.

Besides, for uniform pistons with larger speeds light alloys of aluminium are used. Advantages in aluminium alloys: low density (approximately three times less than cast iron), good thermal conductivity, ease in casting and good machinability. Disadvantages are: mean coefficient of linear expansion (2.5 times greater than cast iron), lower hardness, decrease in strength in high temperatures and finally slightly higher price.

Small density of aluminium allows the construction of lightweight pistons, which influences positively in fuel consumption and also reducing the stress and pressure from inertia forces.

Concerning the relative large coefficient of heat conduction attached to aluminium alloys results in lowering the temperature of the piston crown. It is very important especially in spark-ignition engines. Next figure is a comparison between both engines spark-ignition and diesel engines.

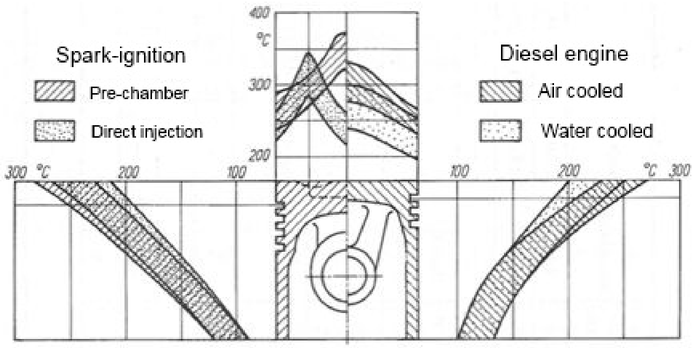


Fig. 2.4. Types of temperature on pistons SI and CI

To reduce a major drawback in the alloy is preferable to choose the lowest rate of coefficient of expansion, by the addition of Si (up to 20%) this effect is considerably lowered, so consequently, improved.

In order to increase resistance to abrasion, piston alloy is heat treated and artificially aged, so that hardness of 120 - 140 HB can be achieved.

Aluminium alloys can be divided into the following main groups:

- Copper aluminium alloys Al-Cu
- Eutectic aluminium alloys Al-Si
- Hypereutectic aluminium alloys Al-Si

Al-Cu alloys are characterized by high thermal conductivity, which is the basic advantage; further advantage is slightly higher strength at high temperatures. One disadvantage is a significant linear coefficient of thermal expansion.

Eutectic alloys Al-Si have a lower coefficient of linear expansion and thermal conductivity at the same time.

Hypereutectic aluminium alloys Al-Si have the lowest coefficient of linear expansion and the major resistance to abrasion of all aluminium alloys.

Al-Si alloys are now widely used. Particularly they are suitable for air-cooled engines, supercharged engines and two-stroke engines. Another benefit of Al-Si alloys is the ease of casting.

Cast steel has a greater strength compared to aluminium alloys which allows performing relatively thin bottoms, which not only allows maintaining a moderate weight piston, but if cooled the bottom reduces the chances of appearing thermal stresses by reducing the temperature gradient.

Good resistance to abrasion and small linear coefficient of thermal expansion are also important advantages to cast steel. The unfavourable point to cast steel is the conductivity of the material which is five times less than light alloys.

The primary method of producing pistons is casting. Casting in moulds (shells), used in lightweight alloys, result in a more fine-grained structure of the material and better mechanical properties.

In some types of engines forging is used. It evokes positive changes in the structure of the material but requires proper interior shape of the piston.

In order to improve the strength properties and hardness of the alloy pistons are applied appropriated machining heat. It has the objective of removing stress after casting of forging and ensuring the stabilization of dimensions.

2.4.1. Material properties

Different material properties

Property	Units	Aluminium alloys									Cast iron	Cast steel
		Al-Cu	Al-Cu-Ni		Al-Si		Al-Si		Al-Si			
		Permanent mould	Permanent mould	Forgings	Permanent mould	Forgings	Permanent mould	Forgings	Permanent mould	forgings		
Density	Kg/dm^3	2,9	2,9	2,8	2,8	2,75	2,75	2,7	2,7	2,68	7,3	7,5
Hardness	20ºC HB	100-120	100-120	100-130	80-110	100-130	100-130	100-130	90-130	100-140	180-240	200-250
	200ºC HB	90-110	90-110	90-110	75-85	80-100	70-90	70-90	65-85	65-90	180-220	150-210
Tensile strength	70ºC MPa	230	240	390	170-220	-	180-240	320-140	150-190	260-290	250	580-700
	300ºC MPa	100	110	160	100-120	-	90-120	130-150	80-110	110-130	200	-
Coefficient of linear expansion	$1/K \cdot 10^{-6}$	28	23-24	23-24	21-23,5	21-23,5	20,5-21,5	20,5-21,5	18,5-20	18,5-20	11-12,5	11-12
Conduction	$\frac{W}{m \cdot K}$	170	148-158	148-158	145-165	145-165	142-160	142-160	117-150	117-150	42-63	52
Elastic modulus	MPa	68000	68000	68000	75000	75000	75000	75000	82000	86000	95000	210000
Poisson Coefficient	-	0,26	0,26	0,26	0,26	0,26	0,26	0,26	0,26	0,26	0,3	0,3

2.4.2. Material composition

Type of material / Component	Al-Cu	Al-Cu-Si	Al-Si[1)]	Al-Si	Al-Si[2)]	Cast iron with lamellar graphite	Cast iron	Cast steel	Steel
C	-	-	-	-	-	2,4-2,8	2,4-2,8	0,2-0,38	0,35-0,44
Si	0,5	4,5-5,5	11-13	20-22	23-26	1,8-2,4	2,9-3,1	0,5-0,6	0,25-,05
Cu	2,5-4,5	6,0-7,0	0,8-1,5	1,4-1,8	0,8-1,5	5-7	-	-	-
Ni	1,75-2,25	-	0,8-1,3	1,4-1,8	0,8-1,3	13,5-17,0	19,5-20,5	-	-
Mg	1,25-1,75	0,2-0,35	0,8-1,3	1,4-1,6	0,8-1,3	-	0,03-0,05	-	-
Fe	≤0,6	≤0,8	≤0,7	≤0,7	≤0,7	Rest	Rest	-	-
Ti	≤0,3	-	≤0,2	≤0,2	≤0,2	-	-	-	-
Mn	≤0,2	0,3-0,45	≤0,3	1,6-0,8	≤0,20	1,0-1,4	0,6-0,8	0,8-1,4	0,8-1,1
Zn	≤0,2	-	≤0,3	≤0,2	≤0,2	-	-	-	-
Al	Rest	-	Rest	Rest	Rest	-	-	-	-
Co	-	-	-	≤0,7	-	-	-	-	-
Mo	-	-	-	-	-	-	0,9-1,1	-	-
Cr	-	-	-	-	≤0,6	-	-	-	-

1) Universal alloy with low expansion coefficient
2) Engines with heavy heat load, such as two-stroke air cooled engines

2.5. Piston pin

Piston pin is used to transfer the pressure from the piston to the connecting rod, also is in charge to provide the swinging motion to the connecting rod. Working conditions are hard and high temperature, tight spaces and small sizing make the calculations very challenging.

2.5.1. Piston pin materials

Working conditions determine the characteristics of the applied materials. Large pressure require very hard and abrasion-resistant surface, and because of the swinging movement resistance to fatigue. There requirements lead to the use of steel with low carbon contents C=0.12 – 0.18 %. After manufacturing pins are externally carburized to give them the hardness and an abrasion-resistant surface.

In large engines components used for carburized steels, e.g.:

$$C = 0{,}12 \div 0{,}18\%, \quad Cr = 0{,}6 \div 0{,}9\%, \quad Mn = 0{,}4 \div 0{,}6\%$$
$$Si = 0{,}15 \div 0{,}35\%, \quad P,\ S < 0{,}035\%,$$

Or nitrified steel, for example:

$$C = 0{,}26 \div 0{,}34\%, \quad Mn = 0{,}5 \div 0{,}7\%, \quad Mo = 0{,}15 \div 0{,}20\%$$
$$Si = 0{,}15 \div 0{,}35\%, \quad Cr = 2{,}2 \div 2{,}5\%, \quad V = 0{,}1 \div 0{,}15\%$$

Hardness in case of carburized piston pins is 58-64 HRC, while the nitride is approximately 64 HRC.

2.5.2. Piston pin construction

Some examples of piston pin construction are given in the next figure:

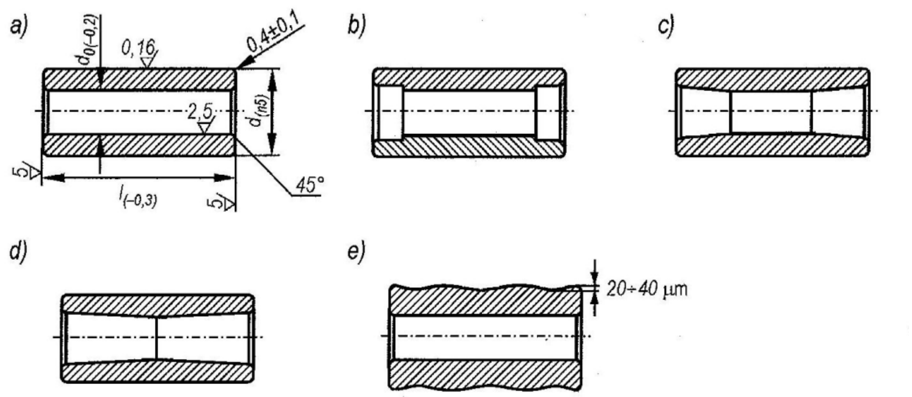


Fig. 2.5. Piston pin construction examples: a) b) cylindrical pins with openings, c) d) pins with cylindrical holes, e) pin with a profiled shape of the exterior surface.

There are several ways to determine the freedom of movement of the pin. Basic solution but most frequently used, particularly in high-speed engines is called carrier bolt mooring (Fig. 2.5.a.). In this embodiment, the carrier bolt is secured only against axial displacement. Can be mounted in the head of the rod and rotate with the hub of the piston (Fig. 2.5.b), can be mounted in the piston (Fig. 2.5.c).

The advantage of mounting solutions with bolt in the head of the connecting rod is reducing the width of the head of the connecting rod, without fear of blurring the pin at large pressures. It is very important to provide the pin the right lubrication otherwise the heat would destroy it. Some examples of carrier bolts are:

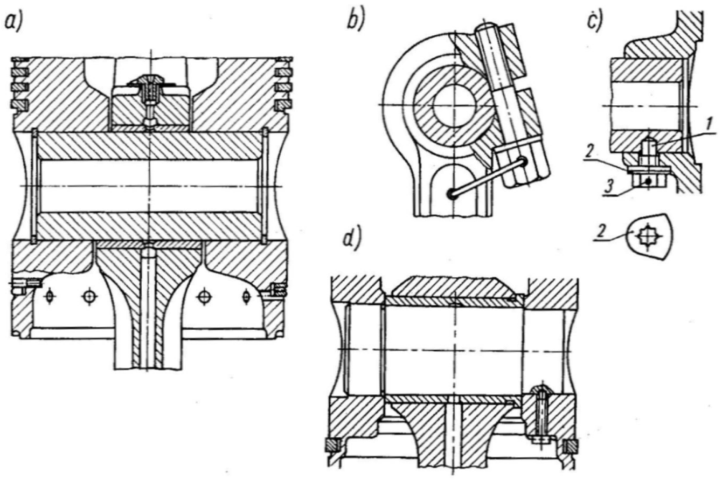


Fig. 2.6. Examples rear piston pins: a) the carrier bolt floating, b) the carrier bolt mounted in the head of the rod, c) the carrier bolt mounted in the hub Piston, d) stepped carrier bolt.

Another solution is to use snap rings, the principal advantage is the easiness in designing them. Some examples of different constructions using them are:

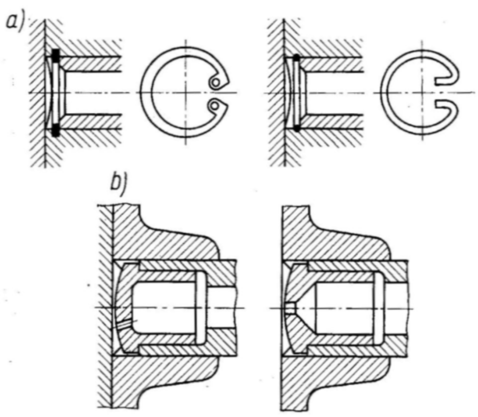


Fig 2.7 Variations of axial arrangements to help the engine, snap rings a) cylinder liner protection against scratches to help the nut b

3. Piston calculation

3.1. Thickness crown calculation

First calculation of the stresses applied in the piston based on the thickness of the piston crown. It is important to note that the thickness marks how the stresses work in the piston. A first approach is to calculate the thickness without taking into account the possible ribs that can be added afterwards.

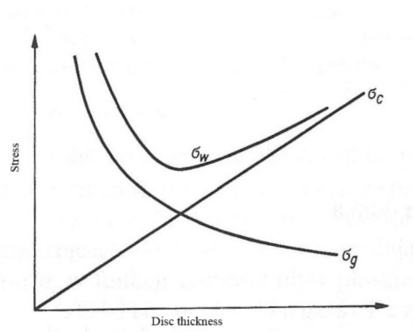


Fig. 3.1. Dependence of the stresses in function of the disc thickness.

It can be seen that there is an optimal thickness where the least stress in the crown is given. In this first calculation the objective is to find an indicative value of it. To start with, support diameter is needed. It should be: $D_p = [0{,}8 \div 0{,}86]D = 258mm$. It is taken 0.86 as it is the worst case.

Pressure acting on the piston crown:

$$p = \frac{\pi \cdot D_p}{8} \cdot p_{max} \tag{1}$$

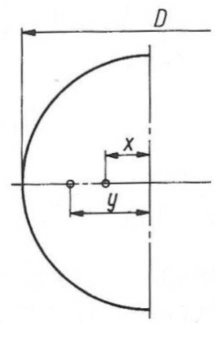


Fig 3.2. Application of the maximum pressure 'X' and reaction applied in 'Y'

$$x = \frac{2}{3}\frac{D_p}{\pi} = 54{,}75mm \tag{2}$$

$$y = \frac{D_p}{\pi} = 82{,}12mm \tag{3}$$

Bending moment value combining (1), (2) and (3):

$$M = P(y-x) = \frac{{D_p}^3 \cdot P_{max}}{24} = 0{,}0036Nm \tag{4}$$

The minimum thickness of the crown:

$$g = \frac{D_p}{2}\sqrt{\frac{P_{max}}{K_g}} = 32{,}25mm \tag{5}$$

Tanking $K_g$as the maximum an allowable bending stress of aluminium is equal [80 MPa].

It has to be defined W:

$$W = \frac{D_p g^2}{6} = 4{,}47 \cdot 10^{-5} m^3 \tag{6}$$

It is said minimum because pressure stress acts as following:

$$\sigma_g = \frac{M_g}{W} = \frac{D_p^2 p_{max}}{4g^2} \tag{7}$$

This thickness is only acceptable if there no thermal component. But as real piston there is the heat part that also affects the piston and causes stresses on it.

Radial stress on the bottom of the crown:

$$\sigma_x = \vartheta \frac{3}{4}\frac{r^2}{g^2} p_{max} = 36{,}06\ MPa \tag{8}$$

Shear stress at the bottom of the crown:

$$\sigma_y = \frac{3}{4}\nu \frac{r^2}{g^2} p_{max} = 9{,}37\ MPa \tag{9}$$

In large pistons appear stresses because of the difference in temperature between the two faces of the piston crown. Thermal stress is calculated by:

$$\sigma_c = \frac{qEag}{2\lambda(1-0.26)} \tag{10}$$

$$E = 6{,}8 \cdot 10^{10}\frac{N}{m^2};\ a = 28 \cdot 10^{-6}\frac{1}{K};\ \lambda = 170\frac{W}{mK}$$

Defining the heat transfer q as:

$$q = \frac{\mu N_e g_e W_d}{F_t \cdot 3600} = 892{,}35\frac{kW}{m^2} \tag{11}$$

$$\mu = 0{,}06;\ N_e = 300kW;\ g_e = 0.3\frac{kg}{kWh};\ W_d = \frac{42000KJ}{kg}$$

$F_t = \pi \cdot r^2 = \pi \cdot 0.15^2 = 0.0707 m^2$

Using equations (10) and (11) and fixing $\sigma_c = 85MPa$ as the maximum stress given by aluminium, this gives us a maximum thickness of the crown of g=0,0125 m.

It can be seen that both thickness obtained from pressure and thermal stresses are not compatible. With that conclusion it has to be thought of another source of cooling the piston crown so that we can achieve our working conditions.

3.2. Length of the piston carrier

The calculation takes into account the normal force pushing the piston to the cylinder liner. Maximum value takes place when the crank is deflected from the axis of the cylinder 35°.

Pressure corresponding to this position is 0.75 of the maximum pressure:

$$P = 0{,}75 p_{max} \frac{\pi D^2}{4} = 0{,}26 MPa \quad (12)$$

Normal pressure:

$$N = P \tan \beta \quad (13)$$

Where: angle of the connecting rod α=35°

Angle β can be found on the ABO sinus theorem.

$$\frac{\sin \alpha}{l} = \frac{\sin \beta}{r} \quad (14)$$

Tanking *l* as the length of the connecting rod and *r* as the radius of the crank.

l=0,8m and r=0,2m. Is obtained an angle $\beta = 8{,}24°$

Normal pressure is then using (13): N_{max}=0,037MPa

Length of the guiding part:

$$l = \frac{N_{max}}{K_{dop} D} = 0{,}251 m \quad (15)$$

Being K_{dop} permissible ground pressure [0.5÷0.8] MPa

3.3. Pressure in the pin

$$K = \frac{F_t p_{max}}{2dl_2} = 5{,}89MPa \qquad (16)$$

Being F_t: piston crown surface; d: pin diameter; l_2: length of the pin

3.4. Piston pin calculation

For the calculation of the piston pin some steps must be followed in order to obtain the current sizing of the pin.

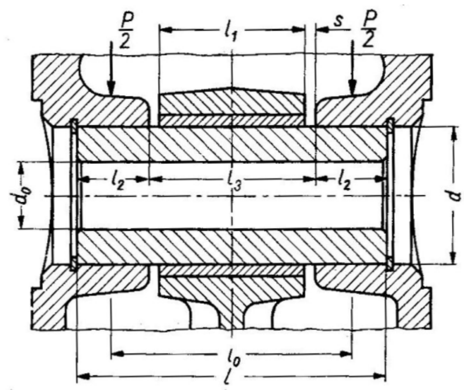


Fig. 3.3. Drawing of the different lengths to be calculated

Initial values should be between:

$$d = (0{,}3 \div 0{,}42)D$$
$$d_o = (0{,}62 \div 0{,}75)d$$
$$l = (0{,}8 \div 0{,}9)D$$

Values taken are:

d=0.09m; d_0=0.055m and l=0.25m

3.4.1. Deflection of the pin

Maximum deflection values should be:,

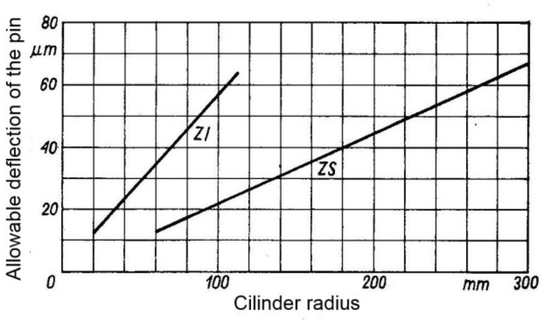


Fig. 3.5. Permissible deflection pin for SI engines and CI

Pin deflection is calculated according to the formula

$$f = \left(1 - \frac{l_1}{2l_0}\right)\frac{l_0^3 P}{48EJ} = 44{,}79 \cdot 10^{-6}m = 44{,}79\mu m \quad (16)$$

Where: P - the largest value of the gas forces, J - moment of inertia section, E - modulus of elasticity of steel; $2.1 \cdot 10^{11}$ N/m^2.

$$P = p_{max}F_t = 353429\,N \quad (17)$$

$$J = \frac{\pi}{64}(d^4 - d_0^4) = 2{,}77 \cdot 10^{-4}m \quad (18)$$

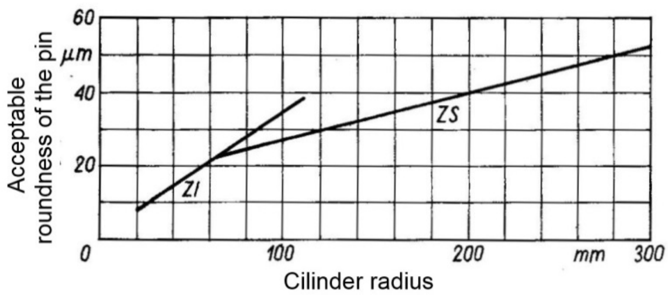


Fig. 3.6. Acceptable roundness of the pin for SI engines and CI

Deformation of the pin under P force is given by:

$$\Delta d_{max} = \frac{0{,}084Pr^3}{EJ_0} = 60{,}32 \cdot 10^{-6}m = 60{,}32\mu m \quad (19)$$

$$r = \frac{d + d_0}{4} = 36{,}25 \cdot 10^{-3}m \cdot \quad (20)$$

$$J_0 = l\frac{(d - d_0)^3}{96} = 111{,}65 \cdot 10^{-9}m^4 \quad (21)$$

It can be seen that it is a little bit higher than the value shown in fig. 3.6., in the simulation it will have to be checked if it is acceptable.

Pressure between the piston pin and the small end of the connecting rod is:

$$k = \frac{P}{dl_1} = 20{,}2 \cdot 10^6 Pa = 20{,}2MPa \quad (22)$$

k should not exceed 45MPa. It can be seen that it is accomplished successfully.

3.4.2. Bending stress

Bending stress due to the deflection of the pin:

$$\sigma_{gu} = \gamma \frac{Pl_0}{\frac{\pi}{8} \cdot \frac{d^4 - d_0^4}{d}} = 182MPa \tag{23}$$

$$l_0 = \frac{3}{4} l = 0{,}1875m \tag{24}$$

$\gamma = \frac{2}{3}$;

Bending stress due to the ovalization of the pin

$$\sigma_{go} = \frac{1}{8} \frac{P \cdot \frac{d+d_0}{4}}{\frac{l(d-d_0)^2}{24}} = 125{,}5 \cdot 10^6 Pa = 125{,}5MPa \tag{25}$$

Total bending stress

$$\sigma_{gc} = \sqrt{{\sigma_{gu}}^2 + {\sigma_{go}}^2 - \sigma_{go}\sigma_{gu}} = 161{,}64 \cdot 10^6 Pa = 161{,}64MPa \tag{26}$$

Allowable bending stresses amount

- For carbon steel k_{dop} = 120 - 150 MPa

- For alloy steels k_{dop} = 160 - 240 MPa

Greatest shear stress is between the piston hub and the small end of the connecting rod.

$$\tau = \frac{0{,}85P}{d^2} \cdot \frac{1+\delta+\delta^2}{1-\delta^4} = 85{,}63 \cdot 10^6 Pa = 85{,}63MPa \tag{27}$$

$$\delta = \frac{d_0}{d} = 611{,}11 \cdot 10^{-3} \tag{28}$$

It can be seen that none of the results exceeds the maximum allowable stress given by steel.

3.5. Material used in the piston

It was decided to use aluminium alloy. It is lighter that steel and with the correct design can stand great pressures and heat.

Material properties

Property	Units	Aluminium alloy
		Al-Si
Density	kg/dm^3	2,8
Hardness	20ºC HB	80-110
	200ºC HB	75-85
Tensile strength	70ºC MPa	170-220
	300ºC MPa	100-120
Coefficient of linear expansion	$1/K \cdot 10^{-6}$	21-23,5
Conduction	$\frac{W}{m \cdot K}$	145-165
Elastic modulus	MPa	75000
Poisson Coefficient	-	0,26

Technical denomination for the aluminium alloy is: PN-EN 573-2: 1997

Technical denomination for steel piston rings: EN-GLSP-150:2001

3.6. Initial scheme

Given the initial dimensions calculated above a first scheme of the piston can be seen. The thickness of the piston crown is 35 mm. It is a very big value but it is a starting point. The piston already has the refrigeration pipe but simulations will be done with and without it. The purpose is to see the effect of the pipe in the piston.

Each hub has three ribs that help to hold the pressure coming from the gases in the piston crown surface. The bottom of the piston crown has three radial ribs. The objective of this solution is to help the piston hold the pressure applied in the top of the piston crown.

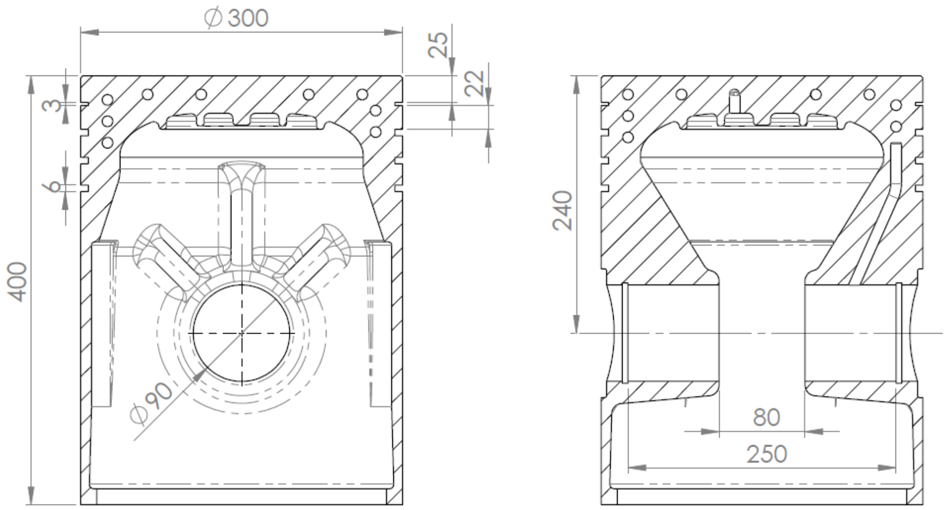
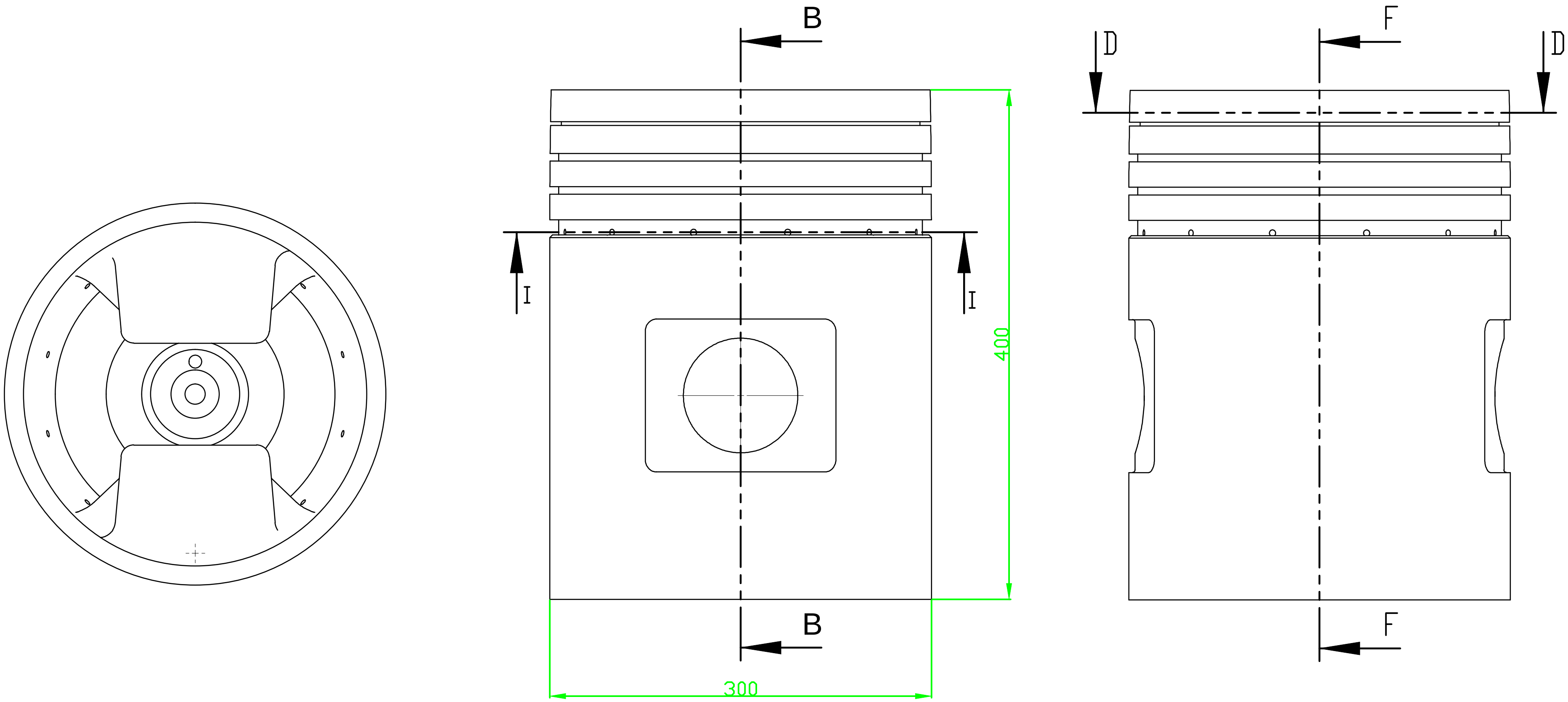


Fig. 3.7. Initial dimensions of the piston

4. Design of the piston

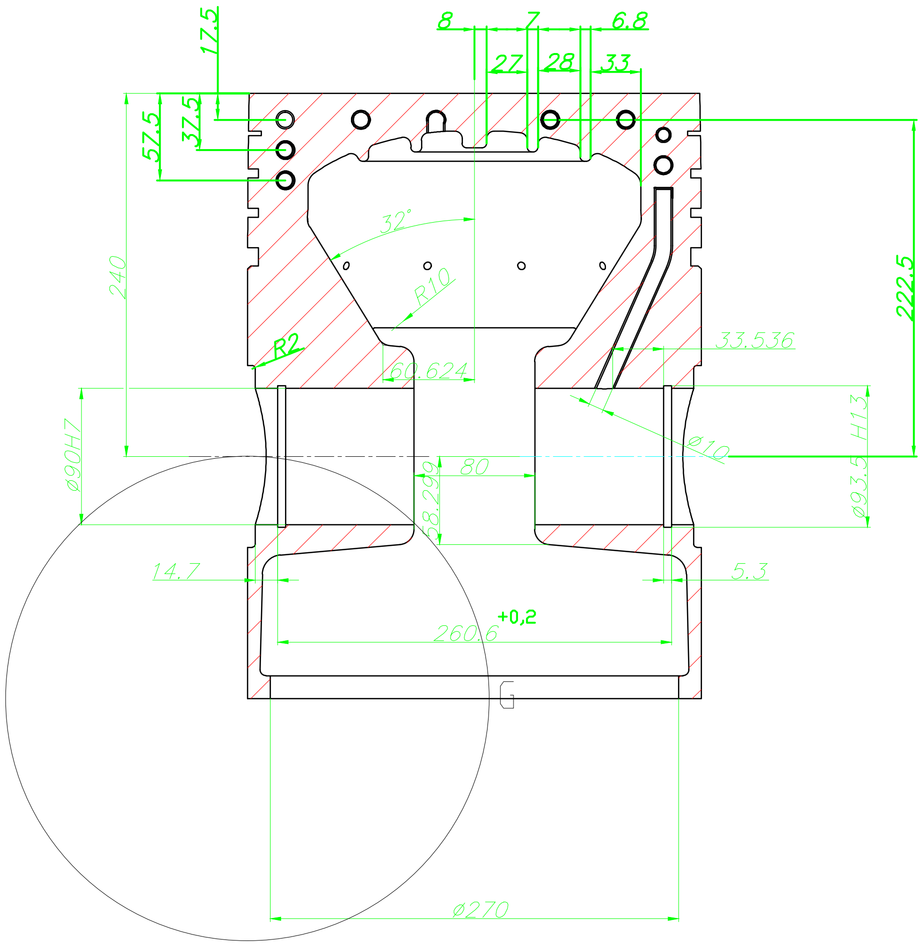
In the following pages the drawings of the piston will be exposed. They contain all the views necessary to understand all the features of the piston.

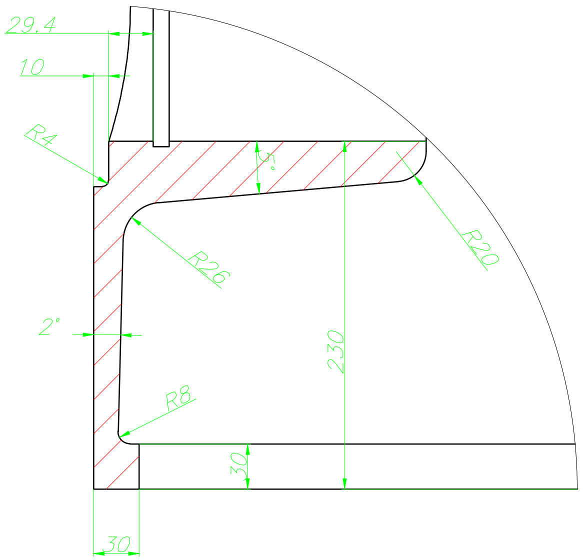


Remark:
All not dimensional radius in the alloy should be maximum 3–5mm.
All not dimensional angles in the alloy are 1,5°.
All non–specified tolerances follow the norm UNE 227/68–1m

Cracow University of technology	Name: J. Anguera	Material: PN–EN 573–2:1997	Object's name: Piston	Plane number: 1.01	Scale 1:5

Section B–B

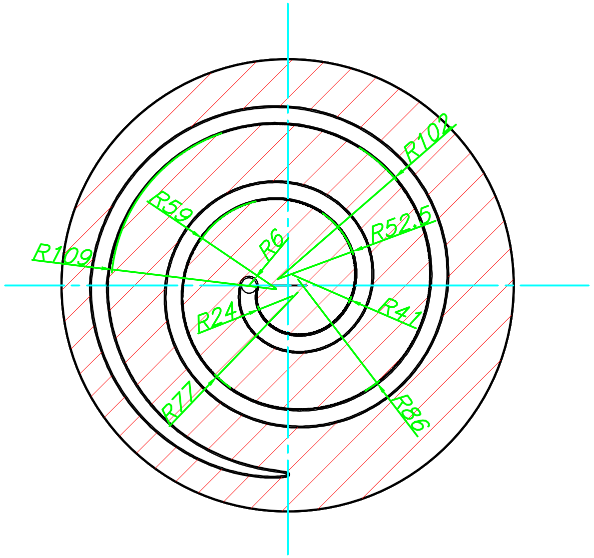


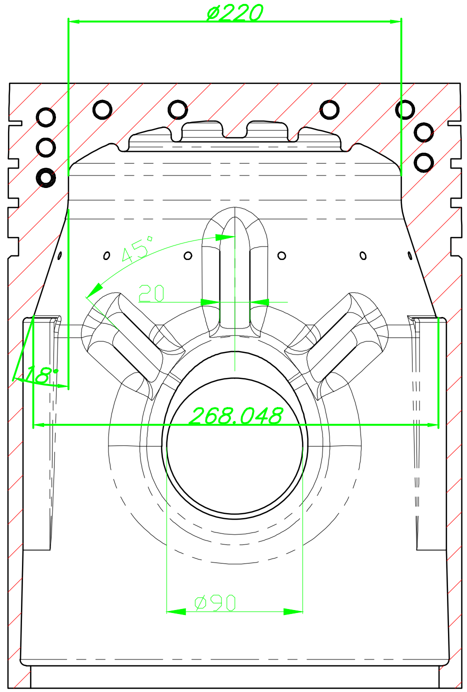


Detail G
Scale 1:2,5

Remark:
All not dimensional radius in the alloy should be maximum 3–5mm.
All not dimensional angles in the alloy are 1,5°.
All non–specified tolerances follow the norm UNE 227/68–1m

Cracow University of technology	Name: J. Anguera	Material: PN–EN 573–2:1997	Object's name: Piston	Plane number: 1.02	Scale 1:5

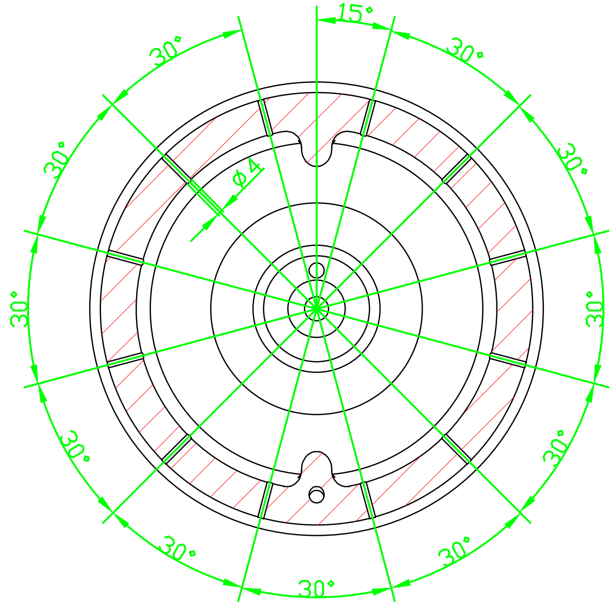




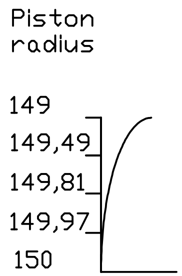
Section F–F

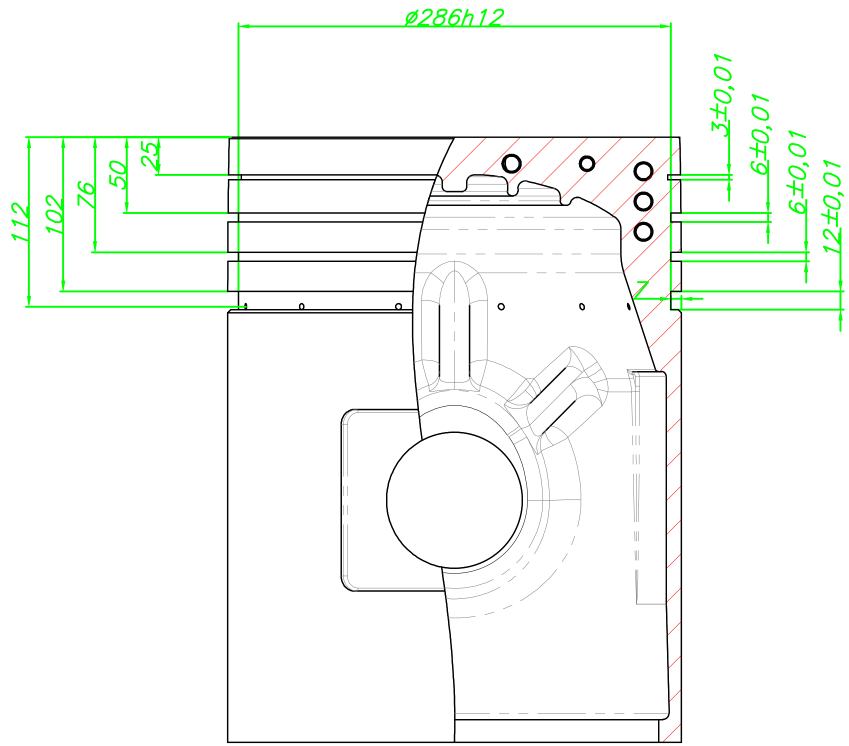
Remark:
All not dimensional radius in the alloy should be maximum 3–5mm.
All not dimensional angles in the alloy are 1,5°.
All non–specified tolerances follow the norm UNE 227/68–1m

Cracow University of technology	Name: J. Anguera	Material: PN–EN 573–2:1997	Object's name: Piston	Plane number: 1.03	Scale 1:5



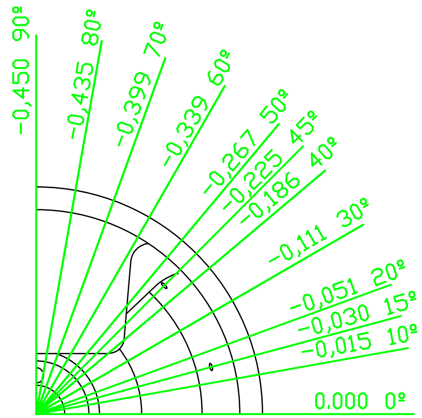
Section I–I





Remark:
All not dimensional radius in the alloy should be maximum 3–5mm.
All not dimensional angles in the alloy are 1,5°.
All non–specified tolerances follow the norm UNE 227/68–1m

Cracow University of technology	*Name:* *J. Anguera*	*Material:* *PN–EN 573–2:1997*	*Object's name:* *Piston*	*Plane number:* *1.04*	*Scale* *1:5*



Piston ovalization is constant and simetric over the entire piston

Remark:
All not dimensional radius in the alloy should be maximum 3-5mm.
All not dimensional angles in the alloy are 1,5°.
All non-specified tolerances follow the norm UNE 227/68-1m

Cracow University of technology	Name: J. Anguera	Material: PN-EN 573-2:1997	Object's name: Piston	Plane number: 1.05	Scale 1:5

5. Piston calculation model

Before simulating it has to be taken into account that some simplifications were made to make the simulation process simpler.

5.1. Convection coefficient

Convection between piston skirt and cylinder walls is not uniform. It changes along the axis of the piston. Understanding in the top is bigger due to the temperature gradient and in the bottom is lower.

In this simulation is assumed the convection coefficient is constant all along the piston. And the value is $h_{c1}=5\cdot10^{-5}$ W/m$^2\cdot{}^{o}$C and temperature $T_{c1}=200^{o}$C.

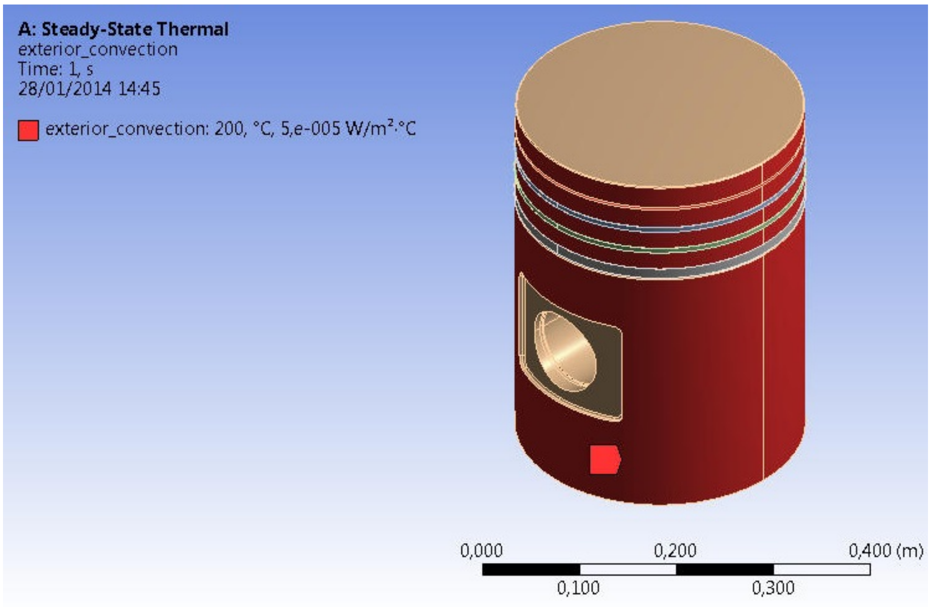


Fig. 5.1 Convection coefficient distribution

5.2. Piston rings

The three piston rings used in this piston they should have different convections values and conduction values to be well defined.

In order to simplify the process it is defined a temperature for each one. The temperatures are the following:

- First sealing ring: T_{s1}= 150ºC
- Second sealing ring: T_{s2}= 140ºC
- Scraping ring: T_{sc}= 130ºC

The following picture shows the different temperatures applied in the piston rings.

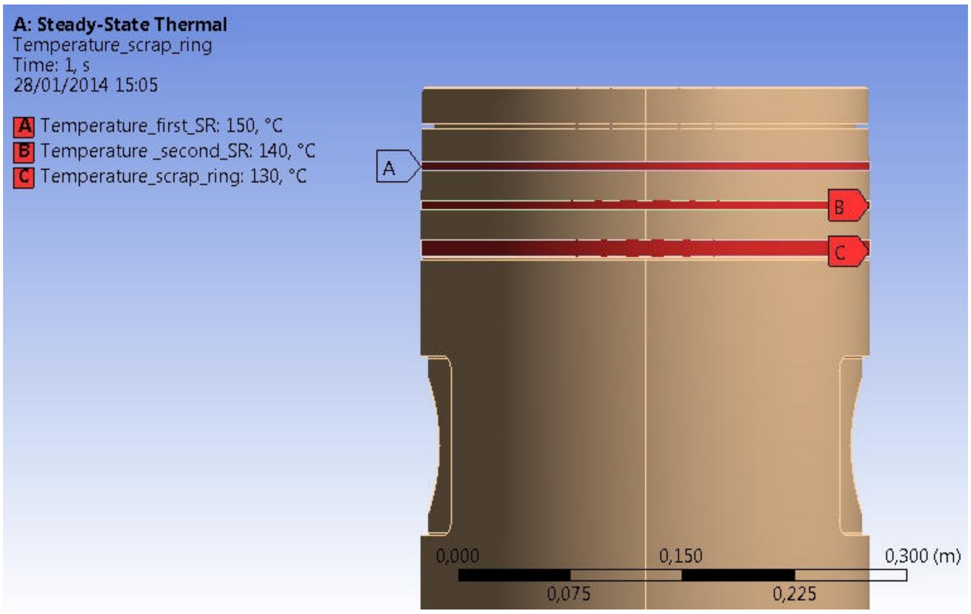


Fig. 5.2. Temperature definition in the piston rings

5.3. Refrigeration pipe

The refrigeration pipe is a basic element to maintain the piston in acceptable working conditions. The flow of oil is the key element to keep a good temperature through the piston crown and helping the piston rings to refrigerate de piston.

In this simulation it is fixed a temperature of the pipe instead of simulation the fluid. It is assumed that the mistake done is not major so the simplification can be done. It is fixed at the temperature of T_p=180^{o}C.

Refrigeration pipe is shown in figure 5.3.

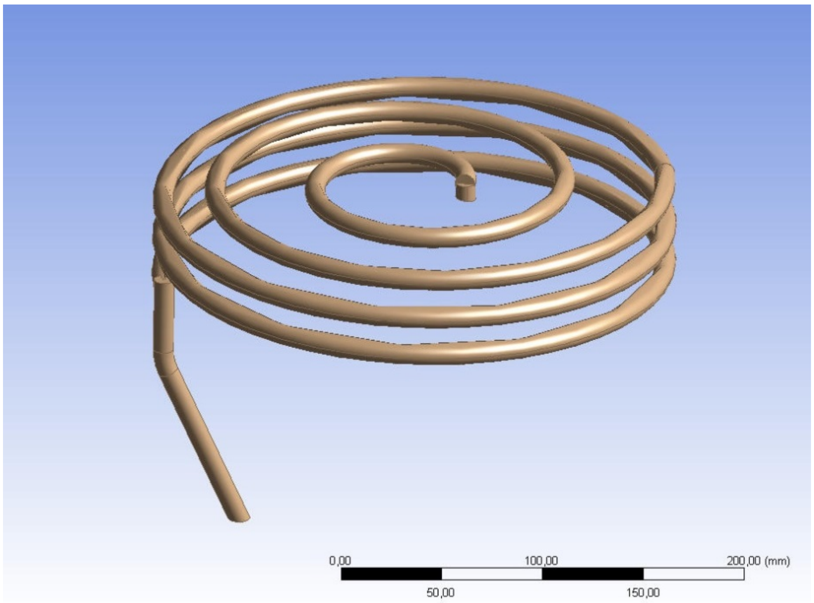


Fig. 5.3. Refrigeration pipe

The shape of the piston pipe helps the piston rings at the time of refrigerating. The most important part is the top. It absorbs great part of the heat coming from the combustion gases. Without the pipe the temperature in the piston crown would be too high to maintain for the material itself.

5.4. Piston crown surface

The piston crown surface is one of the most important parts of the piston. Where the combustion takes place and where the maximum pressure is held by the combustions gases.

As far the pressure is concerned no simplification is made, it is a steady state analysis so to take the worst case the maximum pressure is taken. The value is p = 3 MPa.

When comes to the heat exchange between the gases and the piston crown surface, everything becomes a little bit more challenging. First approach made was to fix a temperature in the piston crown surface, afterwards it was seen that this lead to incorrect results. In order to correct this mistake the Woschni's formula has been used to get the proper convection coefficient. Woschni's correlation, can be summarized as:

$$h_c(w/m^2\cdot K) = 3{,}26B(m)^{-0{,}2}p(kPa)^{0{,}8}8T(K)^{-0{,}55}w(m/s)^{0{,}8} \qquad (29)$$

Being w: average gas velocity, p: instantaneous cylinder pressure, T: instantaneous working-fluid temperature and B: Cylinder diameter.

Given real working conditions it is possible to obtain a real curve of the convection in one cycle which corresponds to 720° of crankshaft rotation (Fig. 5.4).

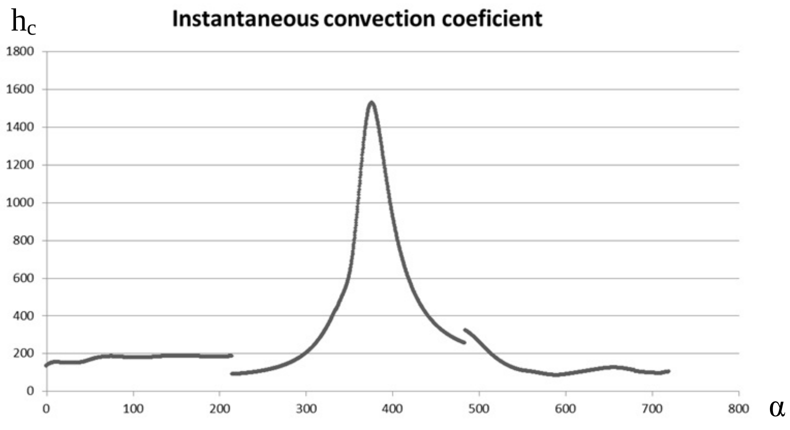


Fig. 5.4. Value of convection coefficient in every position of the crankshaft

This graphic was calculated with the program of W. Mitianiec. "Engine4s", which calculates thermodynamic engine parameters with taking into account non-steady gas flow in the inlet and outflow pipes. In the program the Woschni model of heat convection coefficient in order to calculate heat flow to the engine walls (cylinder, piston and cylinder head walls).

What is interesting for this study is the mean of this value because the study is done in steady conditions. When calculating the obtained value is: 268.35 $w/m^2\cdot{}^oC$. and the temperature 750°C.

6. Simulation of piston loads and temperatures in ANSYS

6.1 First simulation structural approach

The first simulation done in ANSYS only takes into account the pressure applied to the piston. This will give an idea of what will be the tensions in the whole piston.

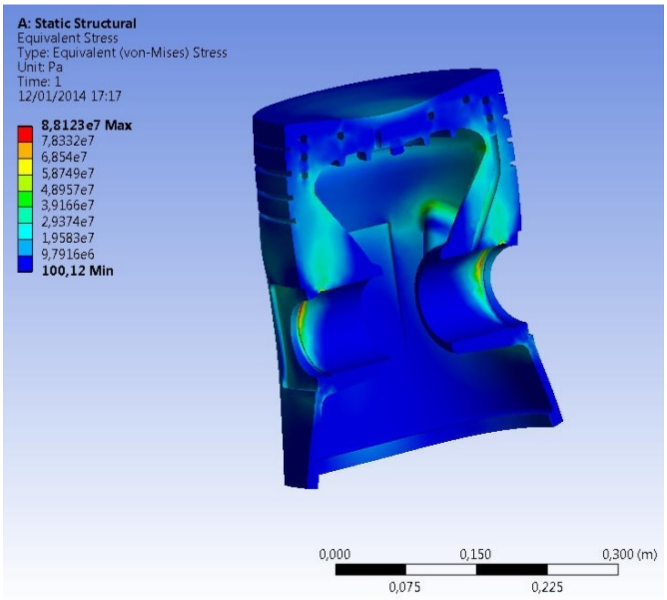


Fig. 6.1. Equivalent stress with no thermal conditions applied

In Fig. 6.1 can be seen that the parts that suffer the most are the ribs witch support the pressure from the combustion chamber. The pressure applied is 3MPa. The values obtained are acceptable. The weakest part is the hubs. As expected because is the part that holds the whole pressure. The radius in the ribs should be increased in order to decrease the value of the tension obtained.

6.2. Piston with no cooling system

6.2.1. Piston crown surface temperature fixed

These results were obtained with fixing the top temperature at 650 ºC. The second step is to apply the thermal conditions and see how the piston reacts to it.

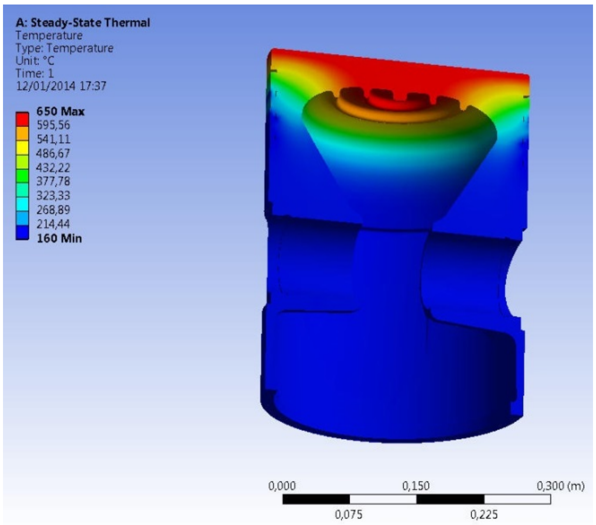


Fig. 6.2. Temperature distribution

The temperatures show that a piston with no cooling system is non-viable as it can be seen in Fig. 6.2. The crown surface is subject to a much higher temperature than the material can support.

This simulation also gives an idea of what will be the deformation like in the whole piston. Both conditions are applied: Structural and thermal.

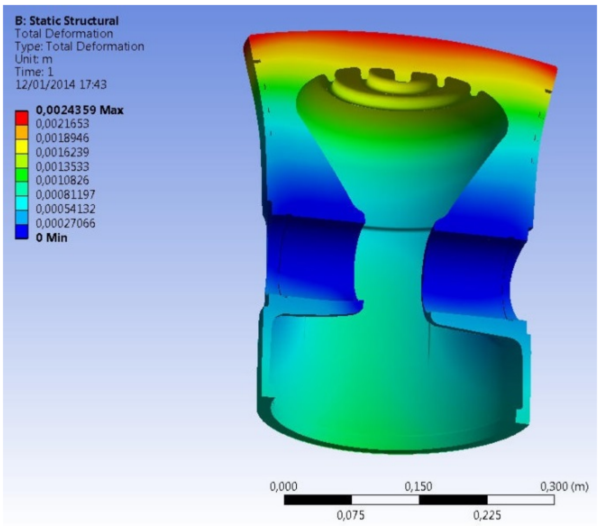


Fig. 6.3 Total deformation

The part that is most deformed is the piston crown in Fig 6.3. This is very important to know because when redesigning the piston a barrel shape will have to be modelled. Giving this barrel shape to the piston will then make the perfect shape at working conditions.

6.2.2. Piston crown with convection at the top

To see different results it is interesting to simulate the piston crown with heat exchange. The value obtained in section 5.4 was 268.35 w/m^2·ºC.

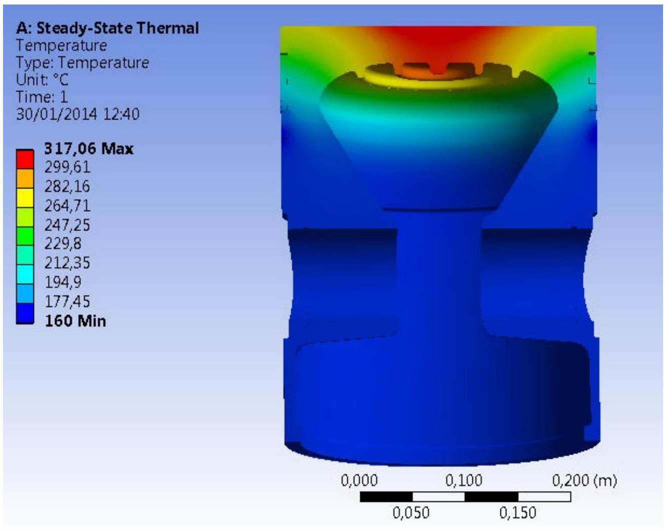


Fig. 6.4. Temperature distribution with convection at the top

As it can be seen the temperature has been lowered a lot but it can be seen that there is a huge gradient of temperatures as it can be seen in Fig 6.4. This will lead to incredible high tensions in the piston.

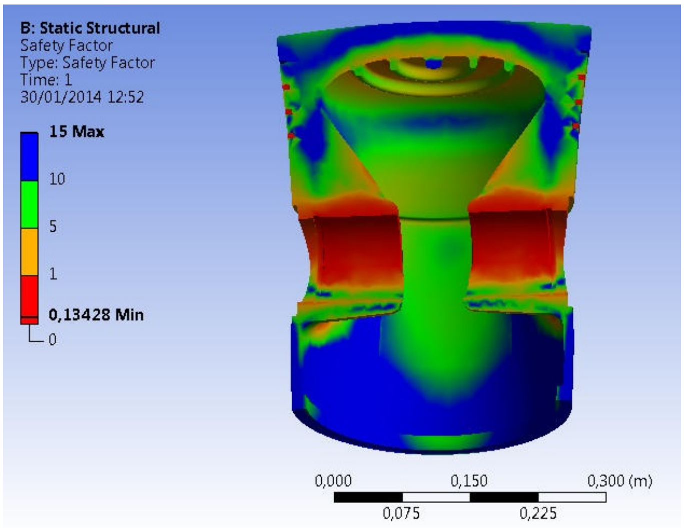


Fig. 6.5. Safety factor piston with no cooling

Having a look at Fig. 6.5 gives an idea of what should be improved in the next design. The lower base of the piston crown has a very small safety factor and the principal problem is the piston hubs. More material will have to be added to the piston hubs in order to help supporting the pressure coming from the combustion gases.

6.3. Piston with cooling system

After seeing that temperature could be a problem a cooling system is proposed in order to decrease the temperature in the piston crown.

6.3.1 Piston crown surface temperature fixed

The effect of the refrigeration pipe can be clearly seen in this case. The reduction of temperature is significant. Even it does not solve the problem with the tensions but it helps to see what way should be followed.

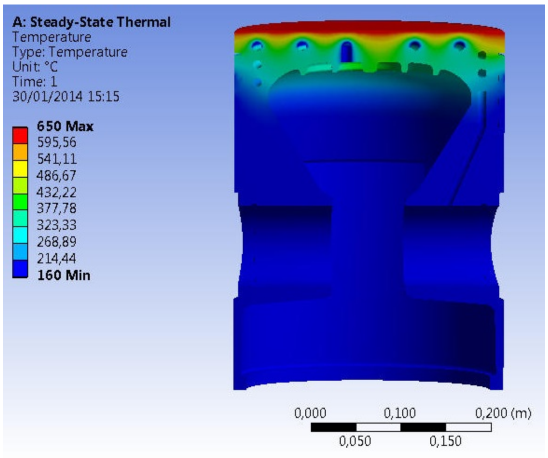


Fig. 6.6. Temperature gradient using refrigeration pipe

It can be seen that the effect of the pipe is enormous comparing to the results obtained in section 6.2.1.

6.3.2. Piston crown with convection at the top

It is very interesting and important to see the effect of the refrigeration pipe with the convection in the top.

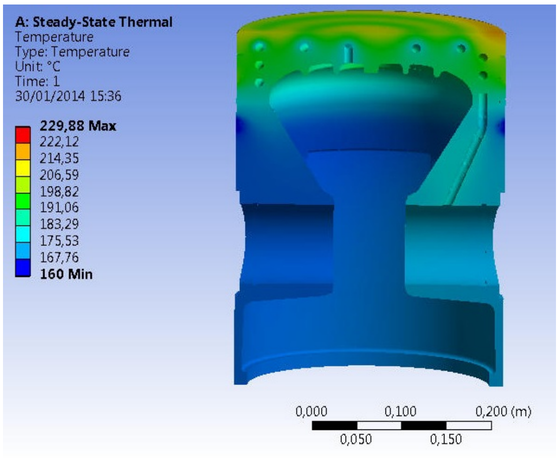


Fig. 6.7. Piston temperature gradient

The effect here is very important and helps the piston maintain good working conditions. With lower temperatures the deformation in the top is also smaller. It can be perfectly seen in Fig 6.7. This is good when designing the piston and obtaining the barrel shape.

6.4. Piston redesign improvements

After the three first analyses a new design of the piston can be made taking into account all the results seen before.

The changes made here were:

1. Reduction of the thickness of the piston crown.
2. Improving the temperature of the piston rings to obtain more accurate results.
3. Increase of the high of the hubs in order to reduce the stress.
4. Introduction of the barrel shape at the top of the piston.
5. Increase the high of the scrap ring in order to adjust to reality.

The results that will be shown in the following figures only include the simulation with convection at the top of the piston crown. The simulation that fixed a temperature in the top of the piston crown leads to non-accurate results which did not adjust to reality.

6.4.1. Simulation in working conditions

Working conditions are:

- Pressure in the piston crown surface p = 3 MPa.
- Heat convection in the top h_c = 268,35 W/m^2.

Following figures show the deformation in the piston. It can be seen that biggest deformation takes place in the border of the top of piston crown surface, and also, in the lowest part of the skirt of the piston.

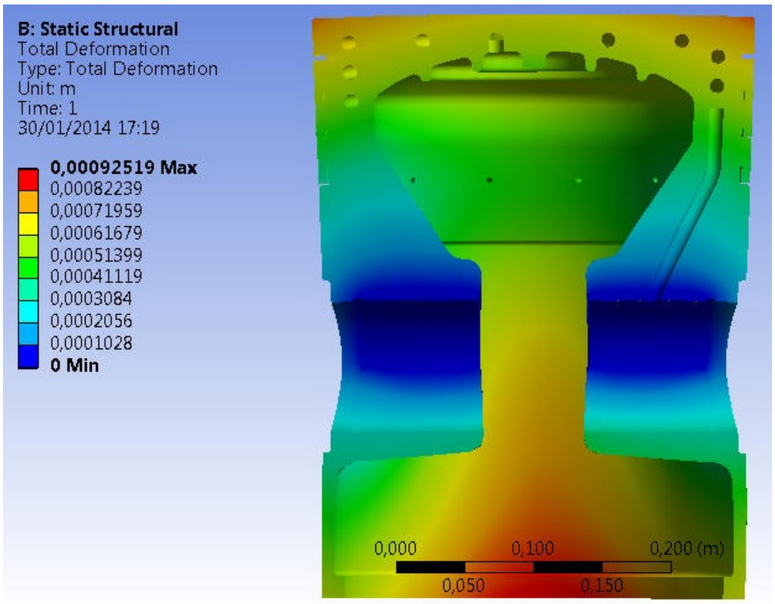


Fig. 6.8. Section of total deformation

Complete view of the deformation of the piston.

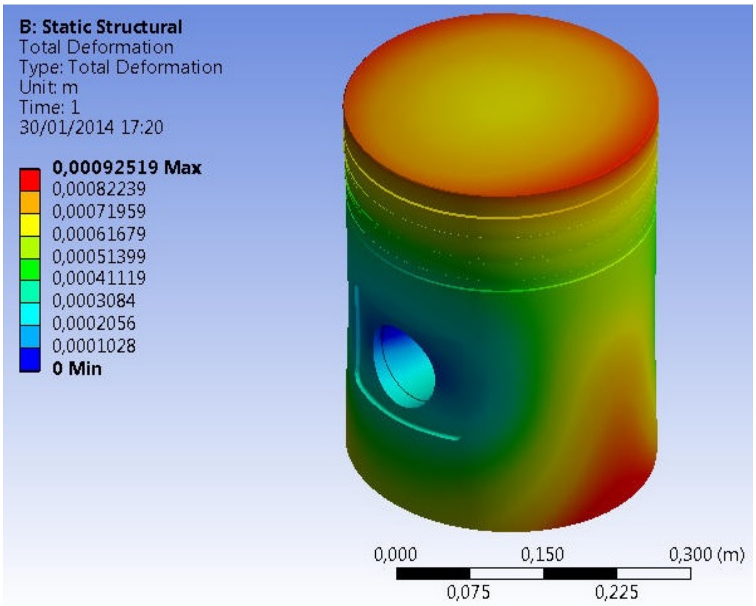


Fig. 6.9. Total deformation

In Fig. 6.9 heat flow shows that the pipe and the piston rings absorb the major part of the heat.

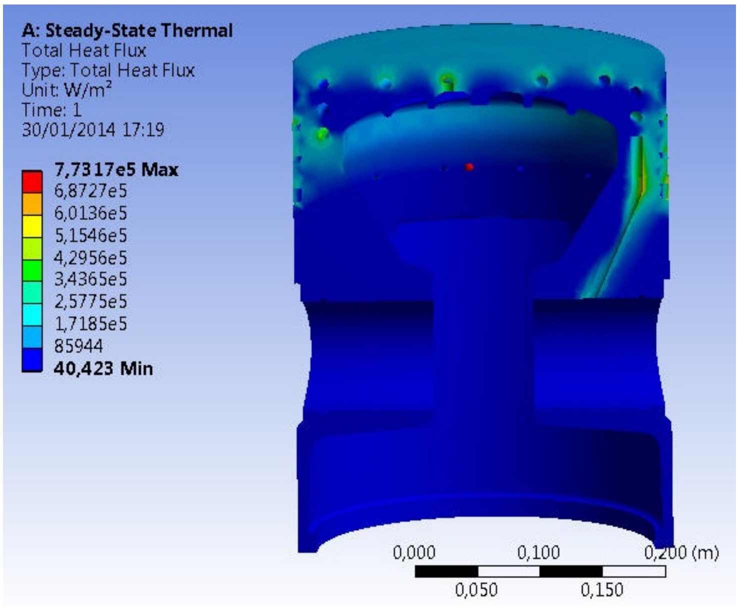


Fig. 6.10. Heat flux

Temperature distribution is very important to see wich of the parts are sumited to a higher temperature. If some is larger than the maximum the material can stand, then some other desing should be thought. Fig 6.11 helps to determine it.

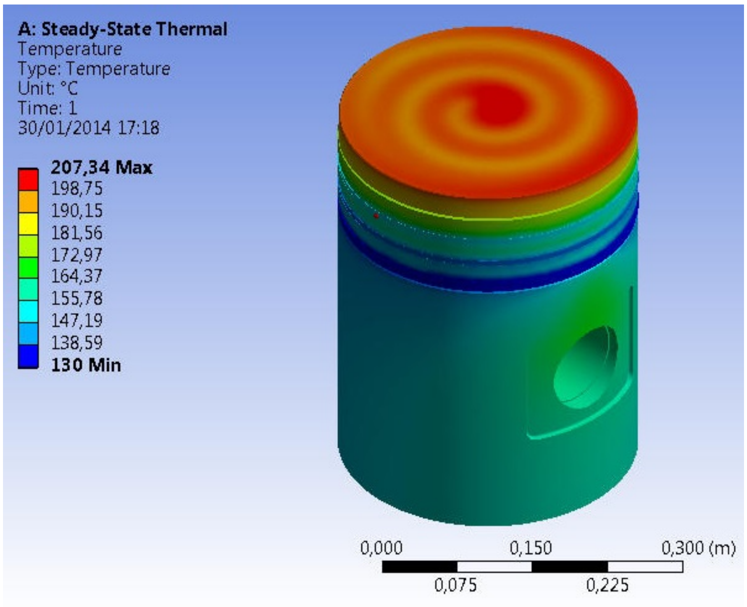


Fig. 6.11. Temperature distribution

The cross section of the temperature distribution is very important to see the effect of the refrigeration pipe (Fig. 6.12).

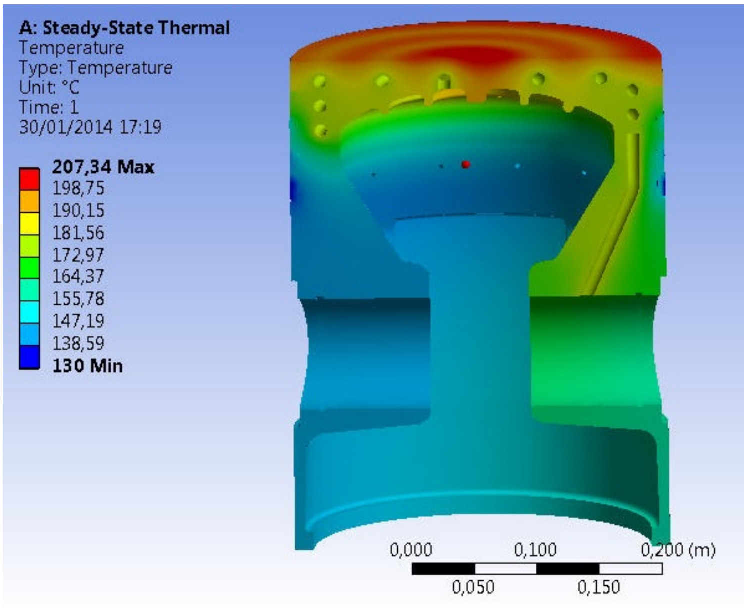


Fig. 6.11. Cross section of temperature distribution

Safety factor (Fig. 6.12) gives a light in the parts which are submitted to highest stresses. Hubs are the ones that suffer the most.

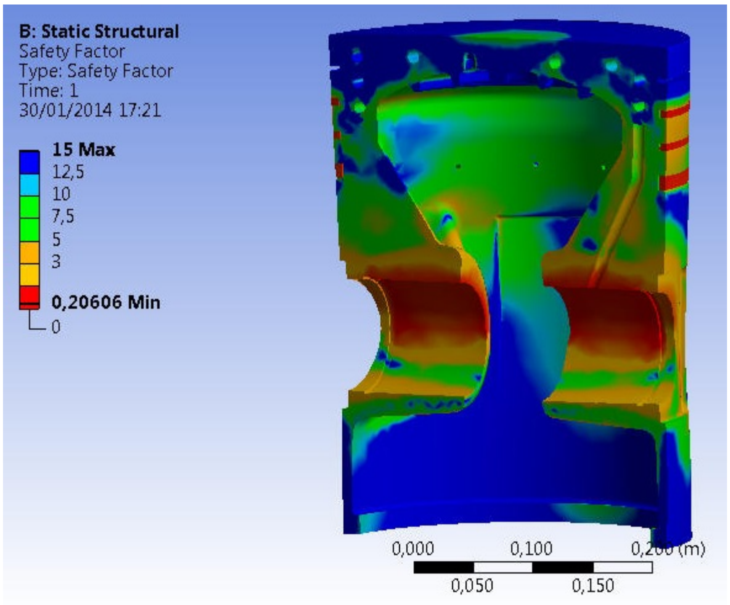


Fig. 6.12. Safety factor

Von-Mises diagram shows the stress distribution in the piston.

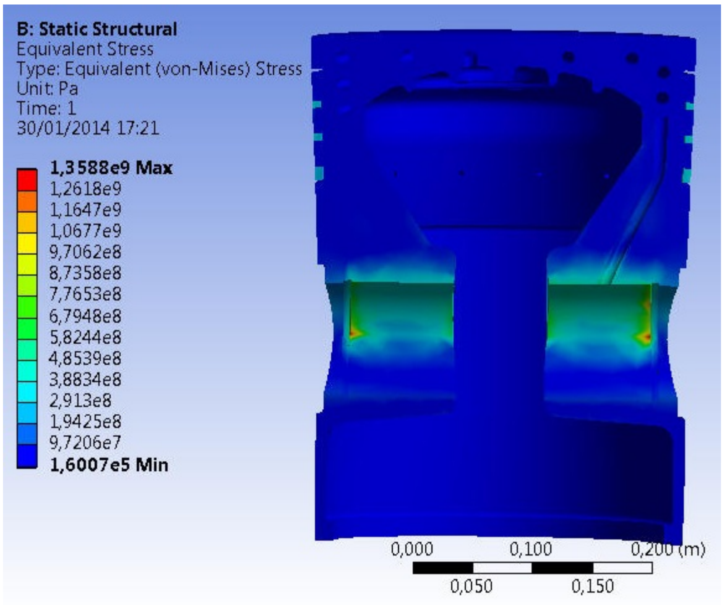


Fig. 6.13. Equivalent von-Mises stress

6.4.2. Simulation in different working conditions

In order to see if the working conditions can be a little bit harder the working conditions have been changed. The new working conditions are:

- Pressure in the piston crown surface p = 5 MPa.
- Heat convection in the top h_c = 500 W/m^2.

The results obtained from simulation are presented in below shown figures:

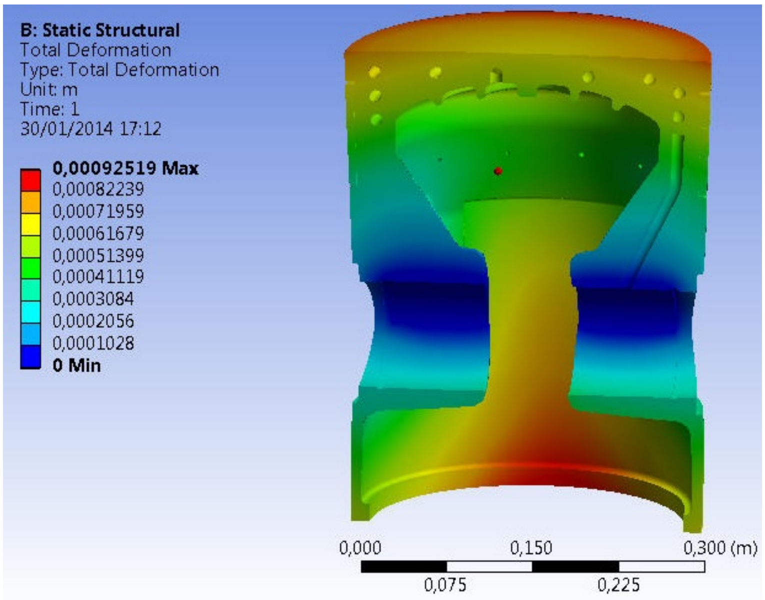


Fig. 6.14. Section of total deformation

Total deformation has not changed much compared to the changing in working conditions (Fig. 6.15). Even increasing them, the deformation is almost the same as in the previous simulation.

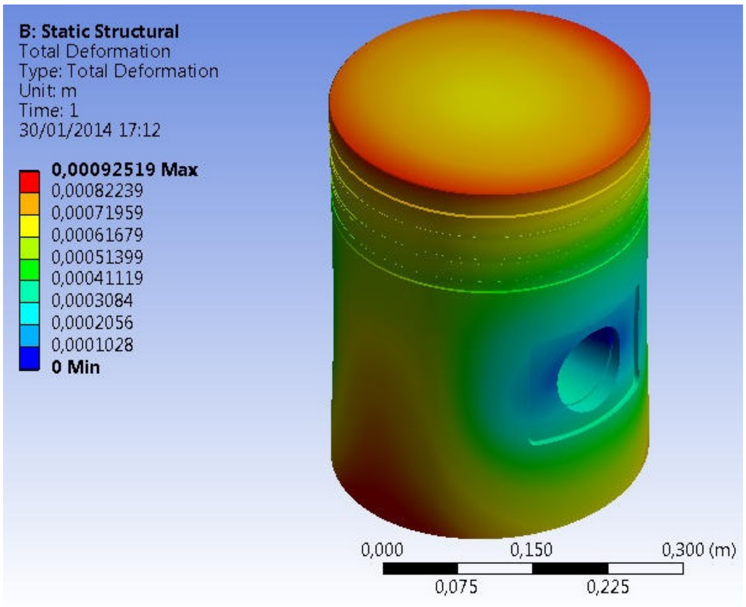


Fig. 6.15. Total deformation

Heat flow is once again concentrated around the refrigeration pipe (Fig. 6.16).

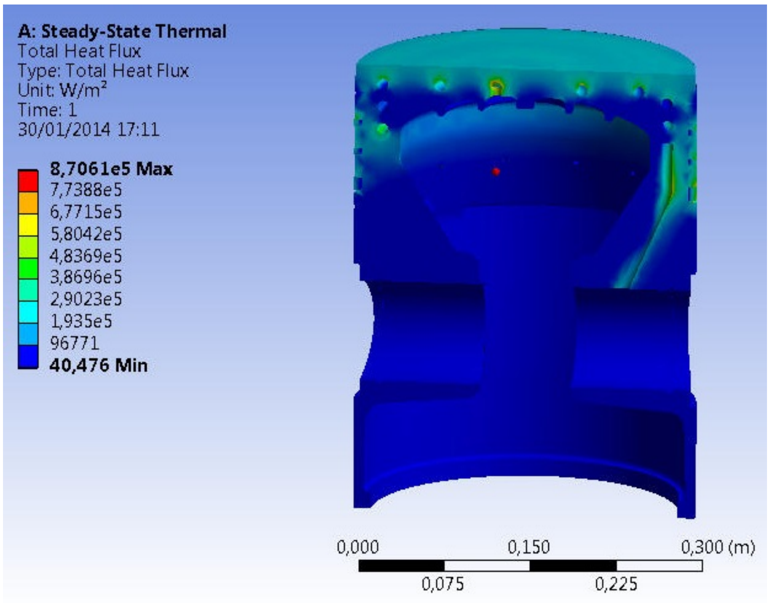


Fig 6.16 Heat flux

Temperatures are a little bit higher than before. Still it is not a problem. Aluminium alloy can support these temperatures.

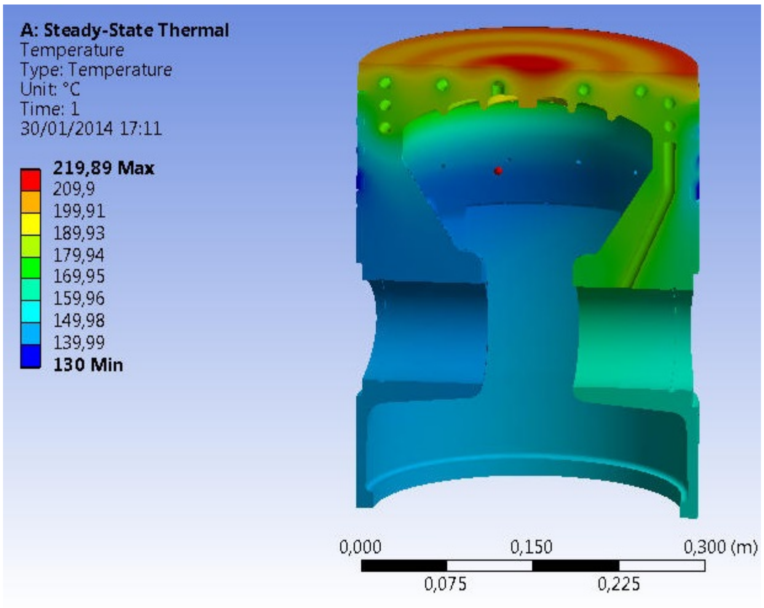


Fig. 6.17. Cross section of temperature distribution

Safety factor (Fig. 6.18) has lowered a little bit but still is not a problem for the piston.

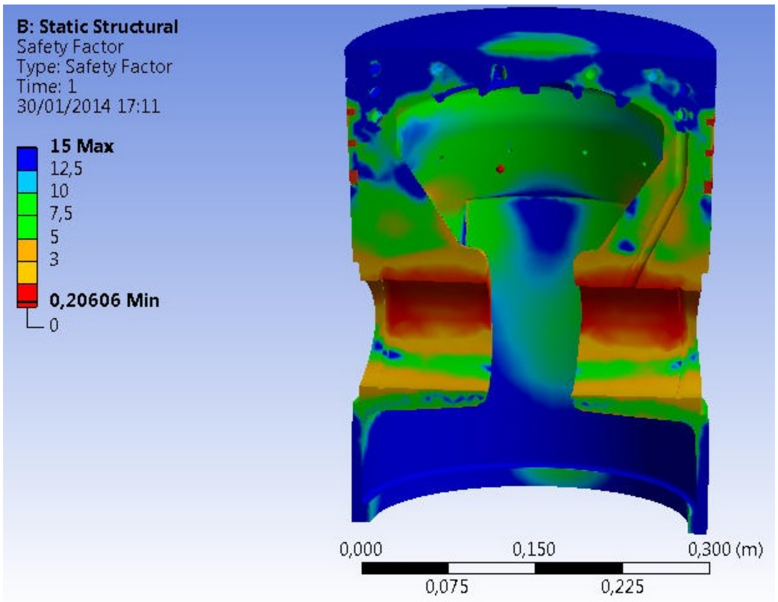


Fig. 6.18. Safety factor

Von-Mises stress distribution is higher in the hubs, as it is shown in Fig. 6.19. As expected because they are in charge of holding the entire pressure coming from the exhaust gases.

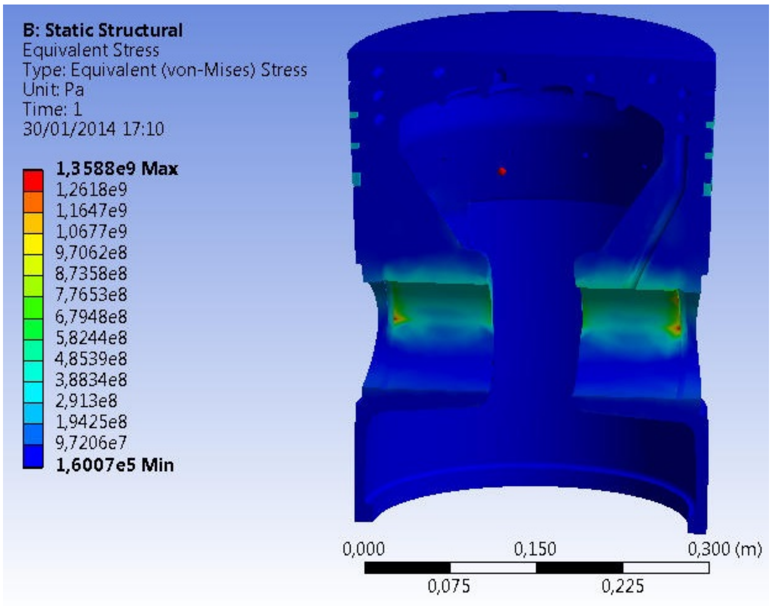


Fig. 6.19. Equivalent von-Mises Stress

7. Verification of piston design on the simulation process

Some changes were made from the initial design to the final one. This was made to improve the shape of the piston and also to reduce stresses in some areas that could lead to a failure of the piston. Other changes were made to improve the cooling of the piston, this lead to a reduction of stresses to.

7.1 Reduction of the thickness of the piston crown.

Initially the piston crown had a thickness of 35 mm, during the simulation it was seen that the piston could withstand the pressure and the heat applied.

Seeing this, the thickness was reduced from 35mm to 25 mm. the result was that the piston could work normally with the new thickness. This is due to two factors: the ribs introduced in the bottom of the piston crown and the refrigeration coming from the pipe.

With the new simulation done, showed that a new reduction could be done. Another improvement could be reducing the thickness of the piston crown to 20 mm or even 15 mm. of course these reduction must be tested in order to see if it can be applied or on the other hand could lead to a piston failure.

7.2 Piston crown heat convection

It is very important to simulate the piston in the most real conditions. If not, the result will not join the reality.

At first the simulations where made fixing the temperature of the piston crown surface to 650ºC. It was seen, later on, that the correct way to simulate the piston was introducing convection from the gases to the piston. The value of the mean of heat convection was fixed to 268,35 W/m^2. This result was obtained from real data of another engine.

7.3. Piston rings temperature

The piston rings take a very important place when cooling the piston. The first simulations were made with the temperature of the piston crown fixed. The value of the top of the piston crown was 650ºC. The piston cannot work in these conditions. The pipe and the convection of the piston itself could not absorb that heat.

When fixing convection in the top of the piston surface it turned out to be that temperature of the piston rings was too high.

The temperature was reduced as follows.

- First piston ring: from 200ºC to 150ºC
- Second piston ring: from 180ºC to 140ºC
- Third piston ring: from 160ºC to 130ºC

With these new conditions the piston ring cools the piston instead of heating it.

7.4 Increase of the high of the hubs

All the pressure that comes from the gases is supported by the hubs. The early simulation showed that the thickness and radius of the hub was not enough to support the stress that in it was caused.

To reduce the stresses some modifications were made. Thickness and radius of the hubs were increased:

Initial dimensions: thickness 12,5mm and final radius joining the piston skirt 10mm

Final dimensions: the top of the hub was ovalized to have a thickness of 27mm and the radius was increased to 13 mm.

7.5. Introduction of the barrel shape at the top of the piston

Deformation in the top of the piston was a problem. Because when the heat is applied in the piston it appeared a deformation of 1mm.

To solve that problem a barrel shape in the top of the piston was applied. The barrel shape starts where the third piston ring ends. It grows gradually until the top of the piston crown. This was very important because it allows the piston to work as it is supposed to be.

If the barrel shape was not applied there could be problems and lead to a failure of the piston because of too much friction in the top. Even the lubrication would not be enough because the clearance between the piston and the cylinder walls is smaller than the deformation that appeared in the top of the piston.

7.6. Increase the high of the scrap ring in order to adjust to reality

The third piston ring is the scrap ring. Its function is to scrap the oil coming from the lubrication. At first the high of the scrap ring was 3 mm but it was too small.

To adjust to reality the high was increased to 6 mm. Moreover some holes were done in order to help the oil go inside the piston and help the scraping process.

8. Conclusions and remarks

The piston analysed in this work is a really large piston, used in large two-stroke diesel engines. The pressures and heat applied are very large. Some especial devices are needed in order to maintain the piston in good working conditions.

Refrigeration is the key system. The refrigeration pipe allows the piston to keep in a good range of temperatures. There are lots of refrigeration devices but this one is really simple to apply. This system is not sealed so there is no much problem in designing it. Further ahead it should be studied the amount of flow of the oil that goes through the pipe. Depending of the loads applied in the piston.

As it can be seen in the simulation the stresses in the piston are not too high. This leads to think that a greater improvement could be done in the terms of materials usage. Some parts do not need as much material as it is used.

Simulation showed that deformation at the top of the cylinder was 1mm. Consequently barrel shape was adopted. The nominal radius at the top of the piston crown surface was reduced by 1 mm. This is very important because now there is no problem in the clearances of the piston.

Piston ovalization was extrapoled from previous designs. Assuming that linear extrapolation was correct. This way piston deformation should compensate normal forces acting against it.

9. Literature (bibliography)

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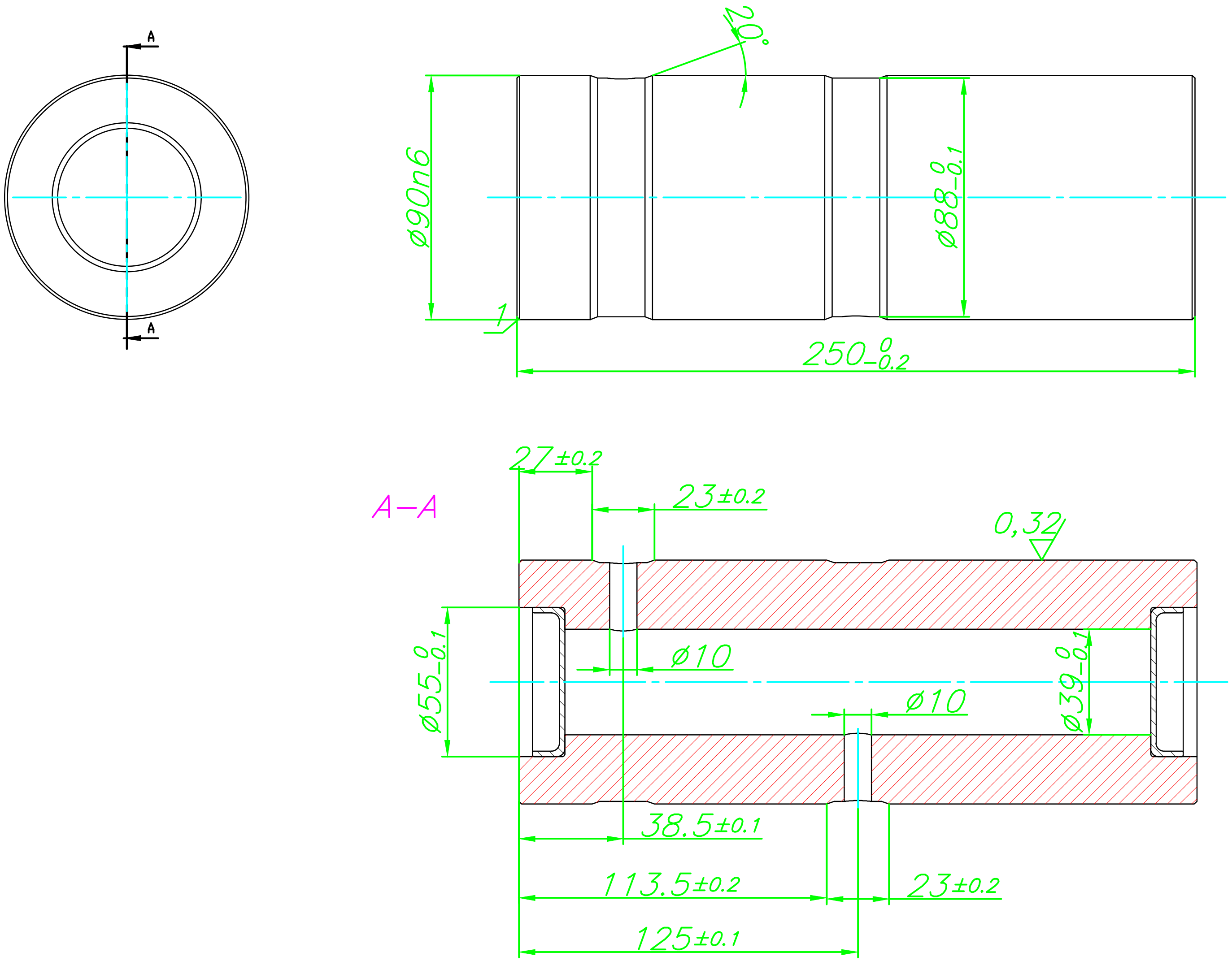
Technical programs used in the work are: SolidWorks, ANSYS, AutoCAD and Engine4s.



Remark:
Surface of the pin carburised
All not dimensional angles in the alloy are 1,5°.
All non–specified tolerances follow the norm UNE 227/68–1m

HRC 60±2

Cracow University of technology	Name: J. Anguera	Material: 20MnCr5	Object's name: Piston Pin	Plane number: 2.01	Scale 1:2.5