CZECH TECHNICAL UNIVERSITY IN PRAGUE
FACULTY OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOTIVE, COMBUSTION ENGINE AND RAILWAY ENGINEERING

Design Proposal for 4-Cylinder Compression-Ignition Engine

Bachelor Thesis

2015 Daniel Stastny

CZECH TECHNICAL UNIVERSITY IN PRAGUE
Faculty of Mechanical Engineering,
Department of Automotive, Combustion Engine and Railway Engineering
Technická 4, 166 07 Praha 6 School year: 2014/2015

BACHELOR THESIS ASSIGNMENT

Student: **Daniel STASTNY**

Study program: **Theoretical Fundamentals of Mechanical Engineering**

Field of study: **without specialization**

Title: **Design Proposal for 4-Cylinder Compression-Ignition Engine**

The thesis will be concerned with the following items:

1. Work out a concept design for 4-cylinder 4-stroke compression--ignition engine with displacement of 1600 to 2000 cm3 and DOHC valvetrain with 4 valves per cylinder.
2. Basic information about the design process and the engine will be provided by the project leader.
3. Use CATIA, CREO (Pro-E) or similar CAD software for generation of engine components and complete assembly.
4. Perform stress analysis of selected components or analysis of performance of engine systems.

Thesis extent:	20 pages
Graphical extent:	CAD models

Specialized literature list:

[1] Spalovaci Motory I, Jan Macek

[2] Introduction to Internal Combustion Engines, Richard Stone

[3] Internal Combustion Engine Fundamentals, John Heywood

Supervisor:	Ing. Mikulec Antonín
Specialist:	Ing. Bogomolov Sergii
Date of thesis assignment:	30 April 2015
Deadline of submission:	19 June 2015

If the student does not submit his/her thesis in time he/she must justify in advance this fact in writing. If the dean recognizes this excuse (handed to the Dean via the Study Department) he will grant the student an alternative term for his/her State Final Exam (two terms still remain). If the student does not present a relevant excuse or if the dean does not recognize the excuse, he will also grant the student an alternative term for his/her State Final Exam. In this case, however, only one term of the State Final Exam remains (Student Examination Code, Art.22, par.3, 4.)

The student acknowledges that he/she must elaborate the project by himself/herself, without any help except consultations with his/her supervisor. A list of used literature, other sources and names of consultants must be listed in the thesis.

I received the assignment on (date): ....
Student

Doc. Ing. Oldřich VÍTEK, Ph.D.
Head of Department

Prof. Ing. Michael VALÁŠEK, DrSc.
Dean

L.S.

Prague, 7 April 2015

Acknowledgements

First of all, I want to thank Ing. Mikulec Antonín for his guidance and for his support in which he provided me all the necessary materials for starting this Thesis. I also want to thank doc. Ing. Luděk Jančík, CSc. for his help and for his Lecture notes throughout my study. Last but not least I want to thank my Family for their beloved support and their patience and making it possible to study here at CVUT.

Declaration

I hereby, declare that this thesis is my own work and effort and that it has not been submitted anywhere else for any award. All used sources of information have been cited.

Prague ………………………

Daniel Stastny

Annotation

This bachelor thesis describes the design proposal for a four – cylinder compression – ignition engine by means of parametrization which is computed in Microsoft Excel being followed by the modeling of each part with the Software Autodesk Inventor.

Abstract

Nowadays, most of the vehicles runs on Diesel engines. We do not use coal-powered steam engines any more. Rudolf Diesel played a vital role in the modern era of transportation by inventing Diesel engines. The need for even bigger and more powerful engines brings the designers of such machineries to a challenging time but with the help of the fast growing Software industry, the challenges for better and fast design concepts can help the designers in any step of the technologically advanced production.

The main goal of the thesis is to get knowledge on how a four cylinder four stroke compression-ignition Diesel engines operates and how each components that fall under it plays an important role in complete working of the engine by taking into the consideration of all the necessary design steps needed, starting with the dimensioning, strength checks of the crank mechanisms, finishing with a 3D CAD model and creating an animation of the engine.

As per the objective, a workout is done with a concept design of the 4-cylinder 4-stroke compression ignition engine with the displacement of 1600-2000 cm3 and DOHC valve train with 4 valves per cylinder. The project leader provides basic information about the design process and the engine. CAD software is used for generating engine components and complete assembly. Last but not the least a stress analysis of the selected components of analysis of the performance of the engine system is carried out.

Content

1. Introduction

The compression ignition engine or as most people know it under the common name "The Diesel engine" is a type of Internal combustion engine (ICE) in which a fossil fuel in this case "Diesel fuel" is mixed with air. The mixt fuel is then compressed and ignites due to its high temperature, which a gas achieves when being compressed. For this reason, Diesel engines tend to have a high compression ratio varying with the type of Injection system in use. In an Indirect injection engine the ratio can be found between 21:1 and 23:1. Direct injection engines use a lower ratio, which is generally between 14:1 and 20:1. This also leads to a very high thermal efficiency in which the Diesel engine has the highest. At the current stage of Diesel technology, only two and four-stroke engines are found. Other types such as the Pistonless rotary engine for Diesel fuel are still under testing. Today we see the Diesel engine but also every other Engine in which we can travel as almost for granted. It has been however, a long journey since the first built engine until todays Diesel Engine of how we know and see it today. Nobody back then could have imagined the importance of this historical invention that makes our live so much easier today than it used to be in the past centuries. Ever since the Invention of the first Diesel engine in the year 1893, it gained a lot of popularity and is used in many daily appliances. Besides the usage in cars, its most common application is found in Ships since the Diesel engine allows the use of heave fuels, which makes it attractive for shipping. Other applications are in heavy-duty equipment as well as in Power Plants.

Todays advanced technology in CAD simulation and modelling allows manufactures to design and build the engine in a virtual state to simulate and to analyse any possible changes within the design concept and to minimize the demand on strength, saving not only money but also lowering the total weight of the engine.

2. Objectives

The overall objective of this Thesis is to get knowledge of how a Diesel Engine operates and how each component of a 4-Cylinder Diesel Engine plays role in a complete working machine, taking into consideration all the necessary design steps needed, starting with the dimensioning and strength checks and finishing with a 3D model as well as making a animation of the working engine.

3. The Historical Background

Bag in the 19th century steam engines where the dominant working engines at the time. For more than 100 years, these heavy and loud machineries constituted the daily life at industry. Then one Man, his name was Rudolf Diesel, a German Engineer, was planning to build a new type of heat engine, very different and unique for its time. While still a student, he realized that the steam engine, while it was in operation, wasted most of its energy when compared to Carnot's ideal energy formulation. He started working on a concept at replacing the steam and using only the energy contained in coal in form of coal dust, which was the primary source of energy at the time. When he claimed his first Patent, in which he worked on a method of finding a way of compressing pure air so intense inside a cylinder that the temperature generated because of this would be far above the ignition temperature of the fuel itself.

4. Parameters and Parametrization

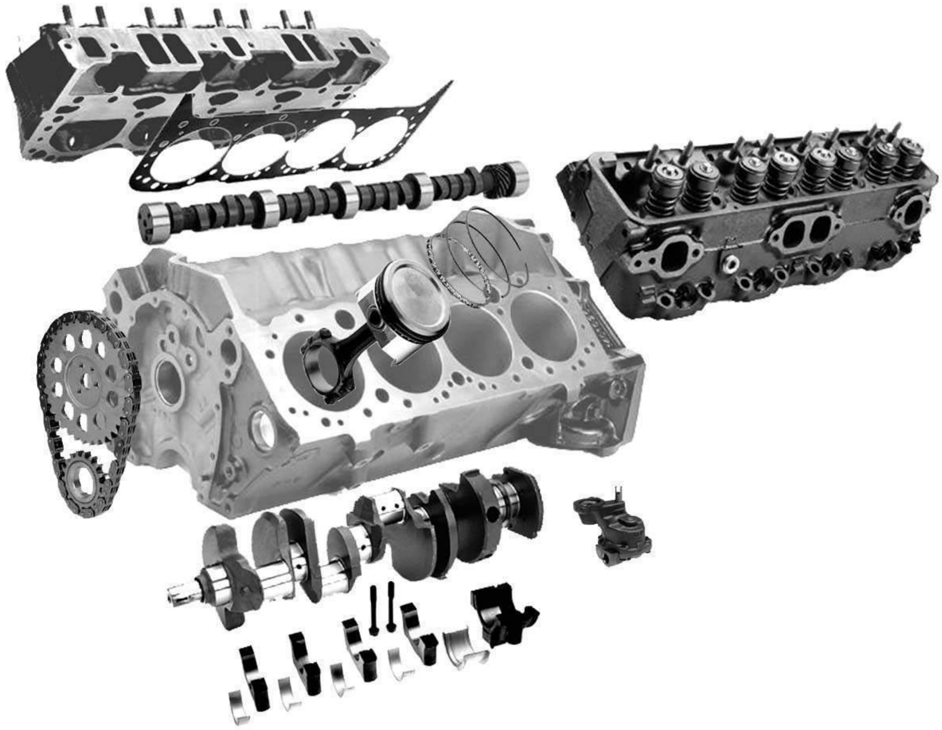
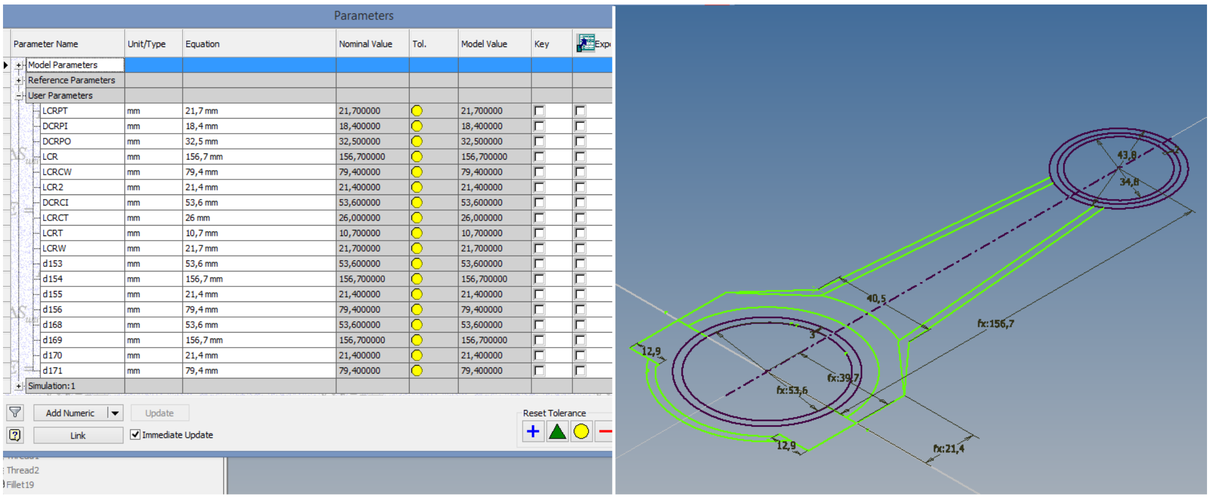


FIGURE 1. EXAMPLE OF A DIESEL ENGINE ASSEMBLY

When designing engine components according to previously Imported or set parameters, it makes it very easy to change those afterwards if later required. As an example, if you wish to change the bore size or the stroke of an engine you may do so by simply adjusting the parameter you want to change and the other parameter will adjust respectively to the linked equations set earlier. Each parameter can be used as a function of each other making it possible to create or change a component very fast, saving time and therefore money having to adjust each dimension manually.



Parameters

Parameter Name	Unit/Type	Equation	Nominal Value	Tol.	Model Value	Key	Expo
Model Parameters							
Reference Parameters							
User Parameters							
LCRPT	mm	21,7 mm	21,700000		21,700000		
DCRPI	mm	18,4 mm	18,400000		18,400000		
DCRPO	mm	32,5 mm	32,500000		32,500000		
LCR	mm	156,7 mm	156,700000		156,700000		
LCRCW	mm	79,4 mm	79,400000		79,400000		
LCR2	mm	21,4 mm	21,400000		21,400000		
DCRCI	mm	53,6 mm	53,600000		53,600000		
LCRCT	mm	26 mm	26,000000		26,000000		
LCRT	mm	10,7 mm	10,700000		10,700000		
LCRW	mm	21,7 mm	21,700000		21,700000		
d153	mm	53,6 mm	53,600000		53,600000		
d154	mm	156,7 mm	156,700000		156,700000		
d155	mm	21,4 mm	21,400000		21,400000		
d156	mm	79,4 mm	79,400000		79,400000		
d168	mm	53,6 mm	53,600000		53,600000		
d169	mm	156,7 mm	156,700000		156,700000		
d170	mm	21,4 mm	21,400000		21,400000		
d171	mm	79,4 mm	79,400000		79,400000		
Simulation:1							

FIGURE 2. PARAMETRIZATION OF CONNECTING ROD ACCORDING TO IMPORTED TABLE OF PARAMETER

To be able to link other components together makes it very easy to create a whole Assembly in a very short time and changing it with only a few adjustments. Tolerances also can be used and linked to dimensions.

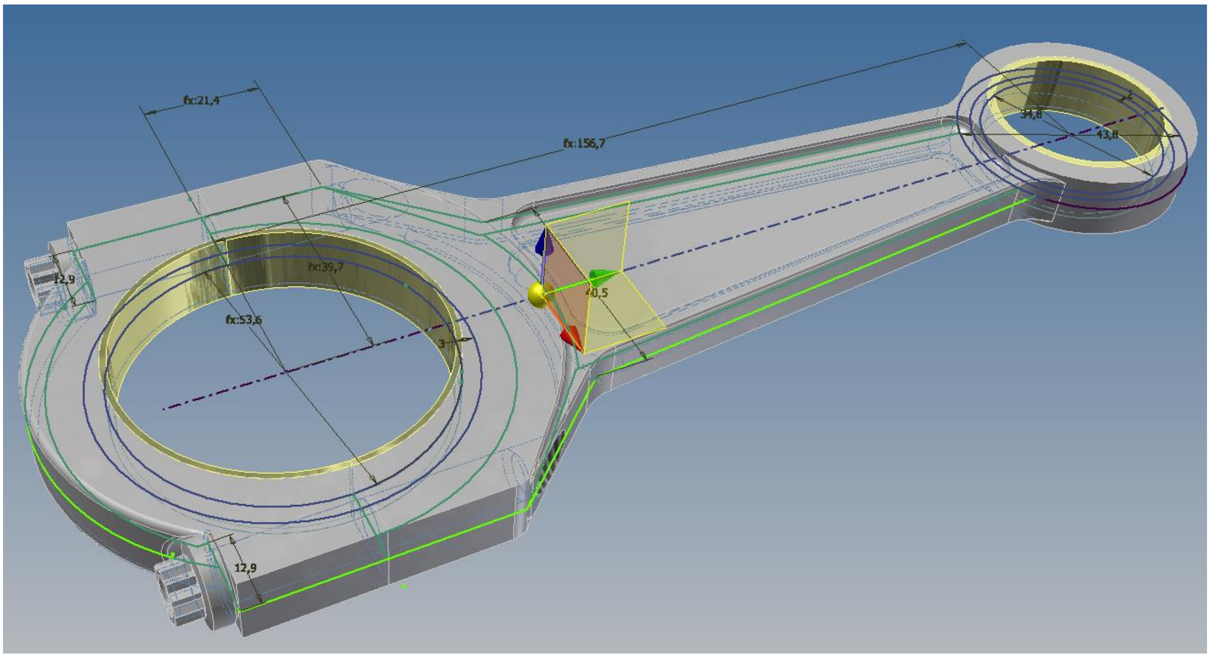


FIGURE 3. DIMENSIONED CONNECTING ROD WITH CENTER OF GRAVITY AND INSERTED BEARING SHELLS

The main concern of each design is that it has to fulfill the strength check but being at the same time as light as possible for saving weight. This can be achieved with a parametric dimensioned Simulation Analysis.

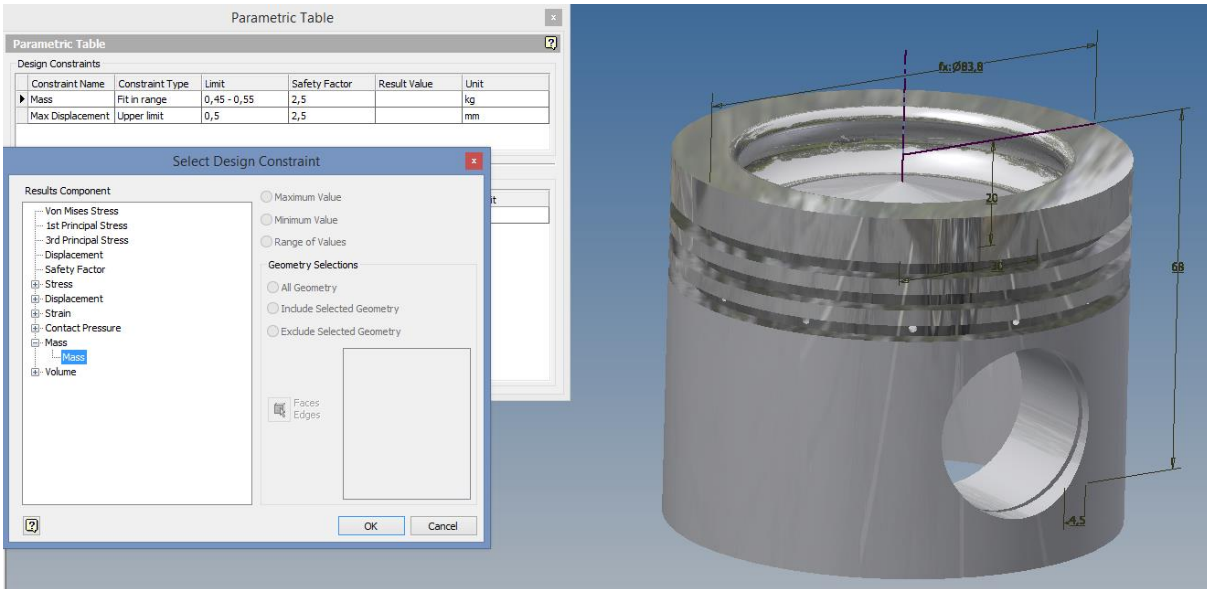


FIGURE 4. PARAMETRIC TABLE WITH OPENED DESIGN CONSTRAINTS OF PISTON

5. Components of Crank mechanism

Today Diesel Engines have to face very high requirements with regard to performance, fuel economy as well as the noise and exhaust emissions. Since engines are operating at very high speeds, only little time is left for the fuel to be injected and for the valves to perform. Therefore, high-performance injection systems with high injection pressure must be used, allowing a more precise control of the whole injection process as well as controlling the fuel quantity.

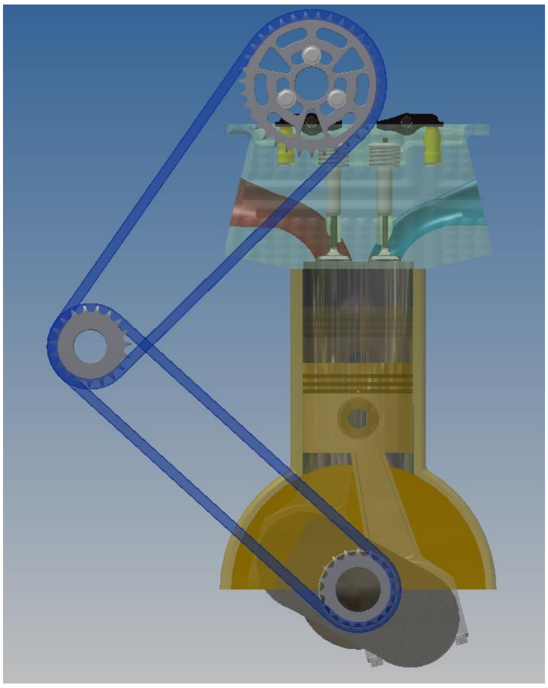


FIGURE 5. SIDE VIEW OF ENGINE

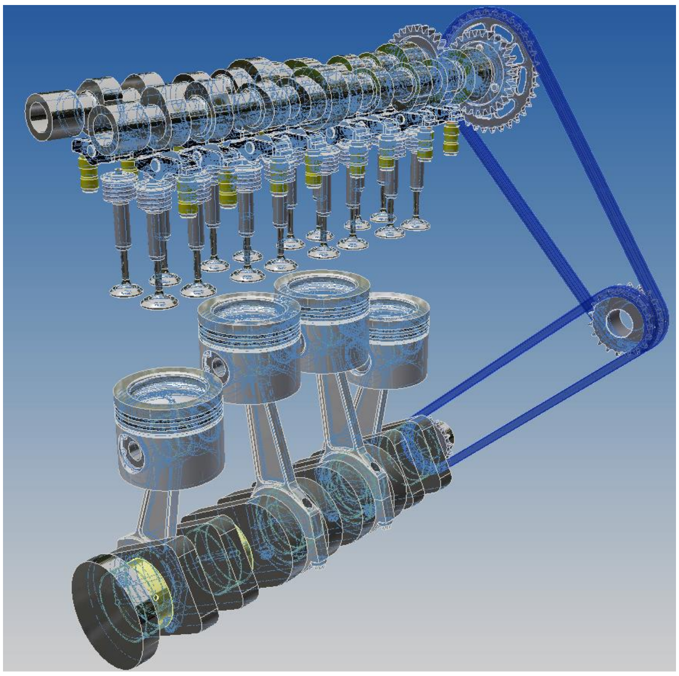


FIGURE 6. ENGINE WITHOUT ENGINE BLOCK AND CYLINDER HEAD

Diesel Engines tend to be divided into two types of injection systems.

- Indirect injection system

- Direct injection system

5.1 Engine Configuration

Engines can perform Power in many types of configurations

- Inline engine
- V-engine
- Horizontally opposed engine

Single as well as two-cylinder engines are very common in use in motorcycles. However, to achieve higher performance and stability, car engines us from up to 4 to 12 cylinder engines. In some cases, even 16 cylinder engines have been found in use.

Probably the most common and simplest form of car engine can be found in the configuration of an Inline 4 – Cylinder. They have one crankshaft and use only one crank pin per cylinder. The Firing order of such a configuration can be found in the orders of

- 1 – 3 – 4 – 2
- 1 – 4 – 3 – 2

V-engines instead use two connecting rods per crank pin. The cylinder bank angle varies from 60° to 90° depending on the number of cylinders in use.

5.2 Piston

FIGURE 7. PISTON SECTION VIEW

It can be sad that the Piston is one of the most important parts in a crank mechanism and that is not only because it is the most loaded element and therefore has to withstand not only the gas forces, which arise within the combustion chamber, but also the inertial forces as well as the heat loading.

The Piston is transferring the force from the expanding medium via a connecting rod to the crankshaft.

5.3 Piston ring

FIGURE 8. PISTON RINGS

The primary function of a piston ring is to seal the piston off from the expanding gas in the combustion process inside the cylinder. Furthermore, they are used for the dissipation of the heat from the piston.

5.4 Connecting Rod

FIGURE 9. CONNECTING ROD

The Connecting Rod links the Piston with the Crankshaft and from this transferring the Power to the Crankshaft. The fact that this component is periodically under a lot of load, makes it very predisposed to failure. Therefore, the demand is set high, trying to prevent a premature failure.

5.5 Crankshaft

FIGURE 10. CRANKSHAFT

The Crankshaft is transferring reciprocating liner piston motion into rotational motion. The main criterion for the design of such a component is to ensure smooth running and balancing out the gas forces.

5.6 Piston Pin

FIGURE 11. PISTON PIN

The Piston Pin connects Piston with connecting rod creating a link which transfers force from the expanding medium which acts on the piston head onto the connecting rod. The demands are set on maximum stiffness at low weight and adequate strength and ductility.

6. Kinematics of Crank Mechanism

D = 84 mm **-Bore size**
r =45 mm **-crank radius**
L =156,7 mm **-connecting rod length**

First and second order equations for finding the pistons position, velocity and acceleration respectively.

$x = r\,.\,(1 - \cos\alpha + 0{,}5\lambda\,.\,\sin^2\alpha)$ [1]
$v = r\,.\,\varpi * (\sin\alpha + 0{,}5\lambda\,.\,\sin 2\alpha)$ [2]
$a = r\,.\,\varpi^2 * (\cos\alpha + \lambda\,.\,\cos 2\alpha)$ [3]

Average piston speed

$$\mathbf{c_s = \frac{l\,.n}{30} = \frac{0{,}09\,.4000}{30} = 12\ m\ s^{-1}}$$

Mean indicated pressure

$\mathbf{p_{is} = 1{,}9\ MPa}$

Mean indicated Power

$$\mathbf{P_{is} = 0{,}25\,.\,\pi\,.\,D^2\,.\,p_{is}\,.\frac{c_s}{\tau} = 31{,}59\ kW}$$

Mean mechanical Power

$\mathbf{P_{em}} = P_{is}\,.\,(0{,}75 \div 0{,}85) = 31{,}59\,.0{,}83 = \mathbf{26{,}22\ kW}$

Total power of engine

$\mathbf{P_{em}\,.\,z(cylinder) = 26{,}22\,.\,4 = 104{,}9\ kW}$

α [°]	β [°]	x [mm]	x1 [mm]	x2 [mm]
0	0	0	0	0
30	8,26	0,007644	0,006029	0,001615
60	14,40	0,027346	0,022500	0,004846
90	16,69	0,051461	0,045000	0,006461
120	14,40	0,072346	0,067500	0,004846
150	8,26	0,085586	0,083971	0,001615
180	0,00	0,090000	0,090000	0
210	-8,26	0,085586	0,083971	0,001615
240	-14,40	0,072346	0,067500	0,004846
270	-16,69	0,051461	0,045000	0,006461
300	-14,40	0,027346	0,022500	0,004846
330	-8,26	0,007644	0,006029	0,001615
360	0	0	0	0

α [°]	β [°]	x [mm]	x1 [mm]	x2 [mm]
390	8,26	0,007644	0,006029	0,001615
420	14,40	0,027346	0,022500	0,004846
450	16,69	0,051461	0,045000	0,006461
480	14,40	0,072346	0,067500	0,004846
510	8,26	0,085586	0,083971	0,001615
540	0,00	0,090000	0,090000	0
570	-8,26	0,085586	0,083971	0,001615
600	-14,40	0,072346	0,067500	0,004846
630	-16,69	0,051461	0,045000	0,006461
660	-14,40	0,027346	0,022500	0,004846
690	-8,26	0,007644	0,006029	0,001615
720	0	0	0	0

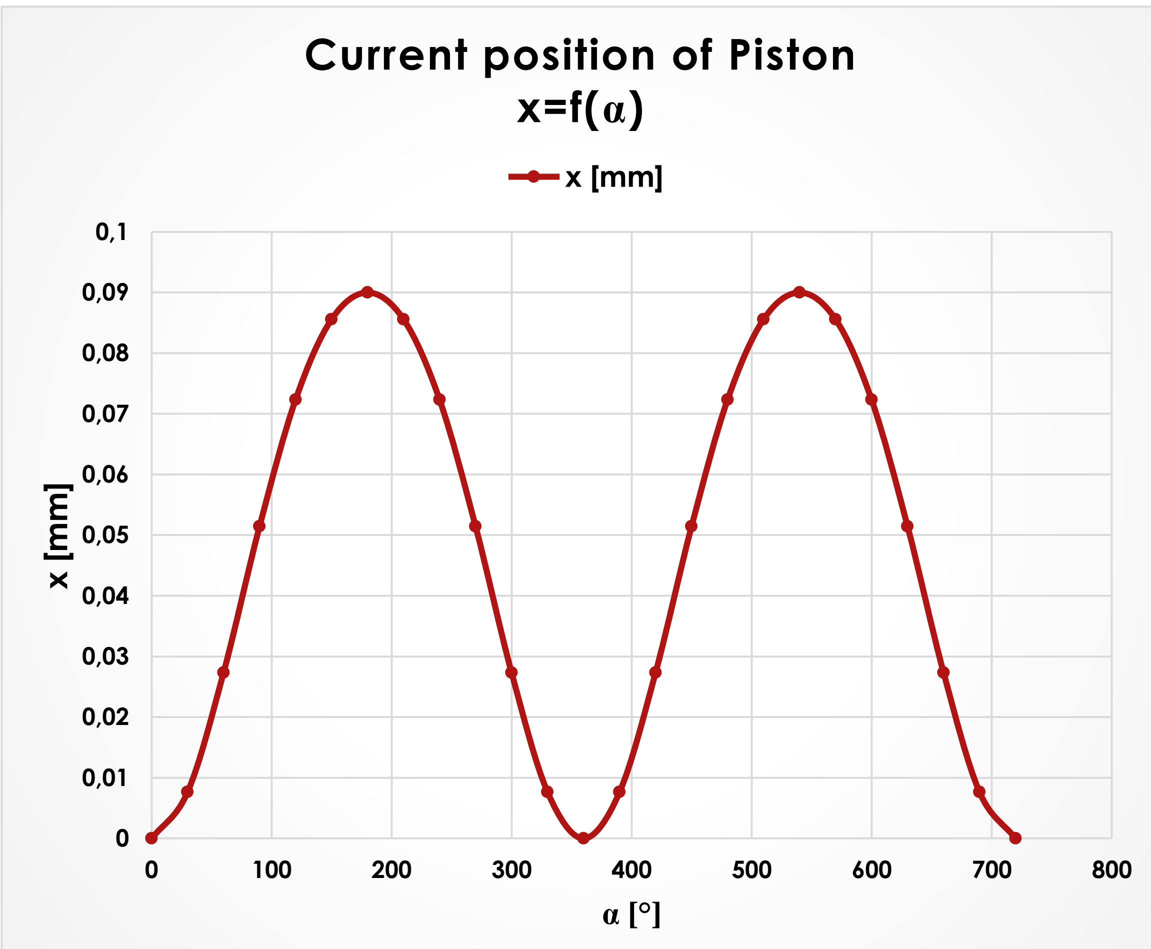


Figure 12. Current Position of Piston. Referring to equation [1]

α [°]	β [°]	v [m/s]	v1 [m/s]	v2 [m/s]
0	0	0	0	0
30	8,26	11,7687	9,4248	2,3439
60	14,40	18,6681	16,3242	2,3439
90	16,69	18,8496	18,8496	0,0000
120	14,40	13,9803	16,3242	-2,3439
150	8,26	7,0808	9,4248	-2,3439
180	0	0	0	0
210	-8,26	-7,0808	-9,4248	2,3439
240	-14,40	-13,9803	-16,3242	2,3439
270	-16,69	-18,8496	-18,8496	0,0000
300	-14,40	-18,6681	-16,3242	-2,3439
330	-8,26	-11,7687	-9,4248	-2,3439
360	0	0	0	0

α [°]	β [°]	v [m/s]	v1 [m/s]	v2 [m/s]
390	8,26	11,7687	9,4248	2,3439
420	14,40	18,6681	16,3242	2,3439
450	16,69	18,8496	18,8496	0,0000
480	14,40	13,9803	16,3242	-2,3439
510	8,26	7,0808	9,4248	-2,3439
540	0	0	0	0
570	-8,26	-7,0808	-9,4248	2,3439
600	-14,40	-13,9803	-16,3242	2,3439
630	-16,69	-18,8496	-18,8496	0,0000
660	-14,40	-18,6681	-16,3242	-2,3439
690	-8,26	-11,7687	-9,4248	-2,3439
720	0	0	0	0

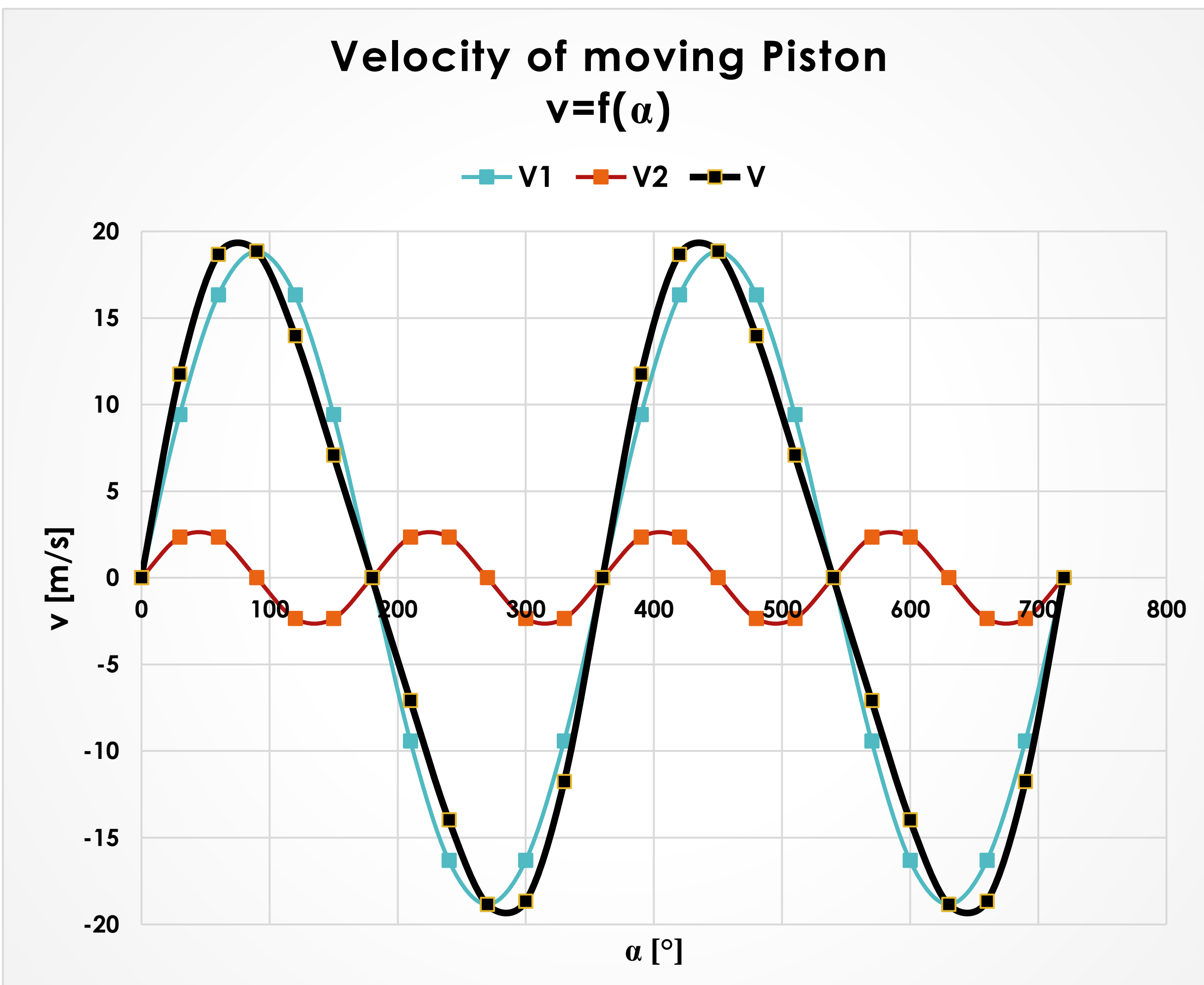


Figure 13. Velocity of moving Piston. Referring to equation [2]

α [°]	β [°]	a [m/s²]	a1 [m/s²]	a2 [m/s²]
0	0	10163,11	7895,68	2267,43
30	8,26	7971,58	6837,86	1133,71
60	14,40	2814,13	3947,84	-1133,71
90	16,69	-2267,43	0	-2267,43
120	14,40	-5081,56	-3947,84	-1133,71
150	8,26	-5704,15	-6837,86	1133,71
180	0	-5628,26	-7895,68	2267,43
210	-8,26	-5704,15	-6837,86	1133,71
240	-14,40	-5081,56	-3947,84	-1133,71
270	-16,69	-2267,43	0	-2267,43
300	-14,40	2814,13	3947,84	-1133,71
330	-8,26	7971,58	6837,86	1133,71
360	0	10163,11	7895,68	2267,43

α [°]	β [°]	a [m/s²]	a1 [m/s²]	a2 [m/s²]
390	8,26	7971,58	6837,86	1133,71
420	14,40	2814,13	3947,84	-1133,71
450	16,69	-2267,43	0	-2267,43
480	14,40	-5081,56	-3947,84	-1133,71
510	8,26	-5704,15	-6837,86	1133,71
540	0	-5628,26	-7895,68	2267,43
570	-8,26	-5704,15	-6837,86	1133,71
600	-14,40	-5081,56	-3947,84	-1133,71
630	-16,69	-2267,43	0	-2267,43
660	-14,40	2814,13	3947,84	-1133,71
690	-8,26	7971,58	6837,86	1133,71
720	0	10163,11	7895,68	2267,43

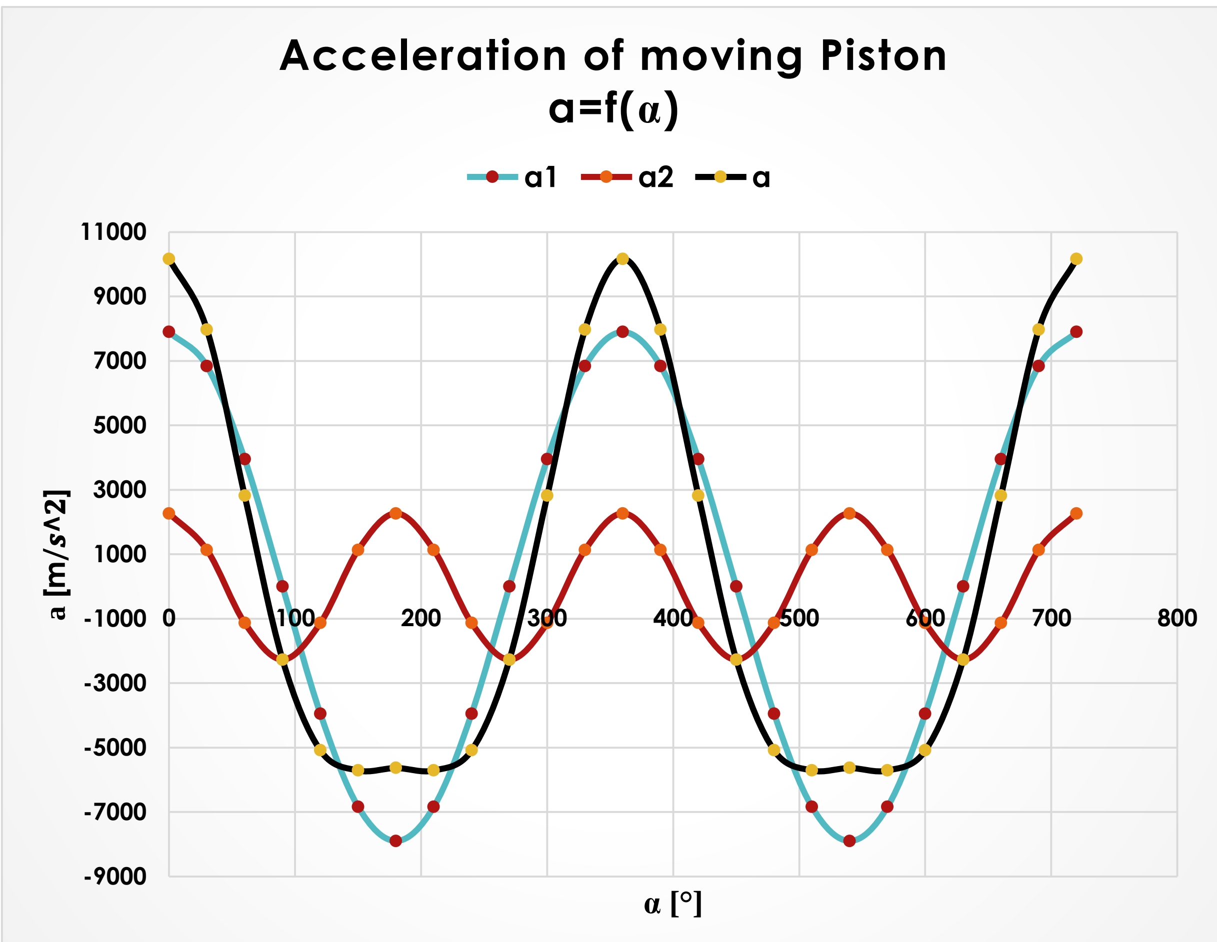


Figure 14. Acceleration of moving Piston. Referring to equation [3]

7. Calculations of Engine Components

7.1 Gas Force (pressure from the expanding medium)

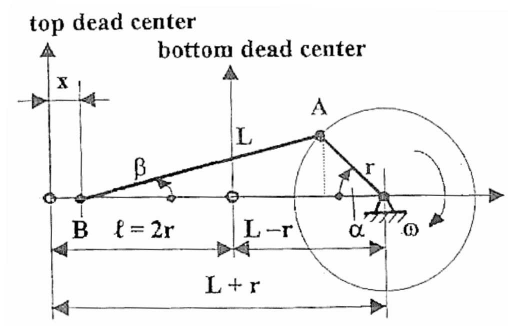


FIGURE 15. PARAMETERS OF CRANK MECHANISM

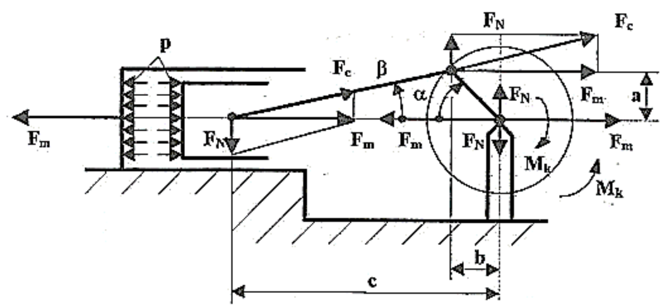


FIGURE 16. FORCES ACTING ON CRANK MECHANISM

$$\mathbf{F_m = \frac{\pi \,.\, D^2}{4} \,.\, (p_{max} - p_{atm}) = \frac{\pi \,.\, (0,084)^2}{4} \,.\, (10 - 0,1) \,.\, 1x10^6 = 54863,52\ N}$$

To find the Force $F_{c\ max}$(Maximum Force that acts on the connecting rod) we need to find the angleβ_{max}. As we can see from the illustration, the maximum slope angle of the connecting rod is when $\alpha = 90°$.

$$\sin\beta = \left(\frac{r}{L}\right) . \sin\alpha \ \rightarrow \ \beta = 16{,}69°$$

Where r/L is the ratio between the crank radius and the length of the connecting rod, also denoted asλ.

7.2 Force acting in the connecting rod

$$F_{c\ max} = \frac{F_m}{\cos\beta} = \frac{54863{,}52}{\cos 16{,}69} = 57276{,}42\ N$$

<u>Piston side thrust</u>

$$F_{N\ max} = F_m \,.\, \tan\beta = 54863{,}52 \,.\, \tan 16{,}69 = 16449{,}41\ N$$

7.3 Inertial Forces of Crank mechanism

Inertial force of feeding mass

$F_{s\,max} = -m_{feed} \,.\, a_{max} = -m_{feed} \,.\, r \,.\, \varpi^2 \,.\, (1+\lambda)$

$m_{feed} = m_{piston} + m_{piston\ ring} + m_{piston\ rod} + m_{piston\ rod\ ring} + m_{piston\ pin} + m_{cB}$

$m_{piston} = 0{,}436\ kg$ $\quad m_{piston\ rings} = 0{,}053\ kg$ $\quad m_{piston\ rod} = 0{,}345\ kg$

$m_{piston\ rod\ rings} = 0{,}02\ kg$ $\quad m_{piston\ pins} = 0{,}004\ kg$ $\quad m_{cB} = 0{,}145\ kg$

$\rightarrow m_{feed} = 1{,}003\ kg$

$F_{s\,max} = -10193{,}6\ N$

Rotating mass

$m_{rot} = m_{cA} + m_{cs\,,rod}$

$m_{cA} = 0{,}378\ kg$

$m_{cs\,,red} = (m_{cp} + 2m_a \,.\, r_a/r)$

$m_{cp} = m_{crank\ pin} = 0{,}427\ kg$ $\quad m_a = m_{crank\ arm} = 1{,}387\ kg$ $\quad r_a = 8{,}873\ mm$

$m_{cs\,,red} = 0{,}974\ kg$

$m_{rot} = m_{cA} + m_{cs\,,red} = 1{,}352\ kg$

7.4 Balancing of crankshaft

To counterbalance the centrifugal force **C** of all the unbalanced rotating masses we apply the following formula:

$C = m_{rot} \,.\, r \,.\, \varpi^2 \rightarrow m_{cb} \,.\, r_{cb} \,.\, \varpi^2$

From this, we can find the mass we need to add to each crank arm.

$m_{cb} = m_{rot} . r/r_{cb}$

We choose $r_{cb} = 70\ mm$

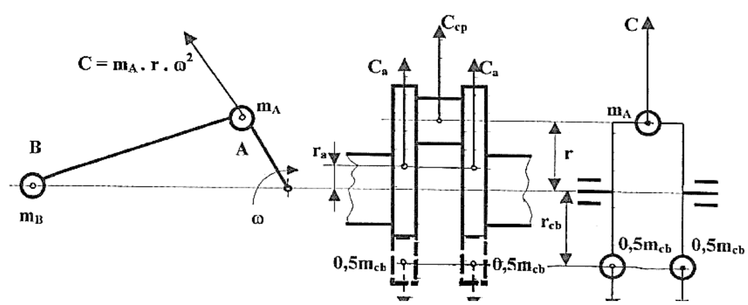


FIGURE 17. BALANCING OF CRANKSHAFT

$$m_{cb} = 1{,}352 \,.\, \frac{45}{70} = 0{,}869 \text{ kg}$$

$\therefore \frac{0{,}869}{2} = 0{,}435 \text{ kg}$ - needs to be added to the end of each crank arm

7.5 Piston

The dimensioning of the Piston is according to a size ratio found in several literatures. Every literature uses their own size ratios, which makes it hard to generalize.

$GL = (0{,}8 \div 1{,}3).D \quad \rightarrow \quad GL = 68 \text{ mm}$

$KH = (0{,}5 \div 0{,}8).D \quad \rightarrow \quad KH = 42 \text{ mm}$

$BO = (0{,}35 \div 0{,}4).D \quad \rightarrow \quad BO = 32 \text{ mm}$

$F = (0{,}1 \div 0{,}2).D \quad \rightarrow \quad F = 8{,}4 \text{ mm}$

$St = (1{,}5 \div 4{,}5) \quad \rightarrow \quad St = 3{,}1 \text{ mm}$

$SL = (0{,}5 \div 0{,}9).D \quad \rightarrow \quad SL = 46{,}05 \text{ mm}$

$AA = (0{,}27 \div 0{,}4).D \quad \rightarrow \quad AA = 34 \text{ mm}$

$s = (0{,}1 \div 0{,}15).D \quad \rightarrow \quad s = 8{,}4 \text{ mm}$

Groove height $\rightarrow$ 3 mm

$r = 37{,}4$ mm - Internal radius of piston

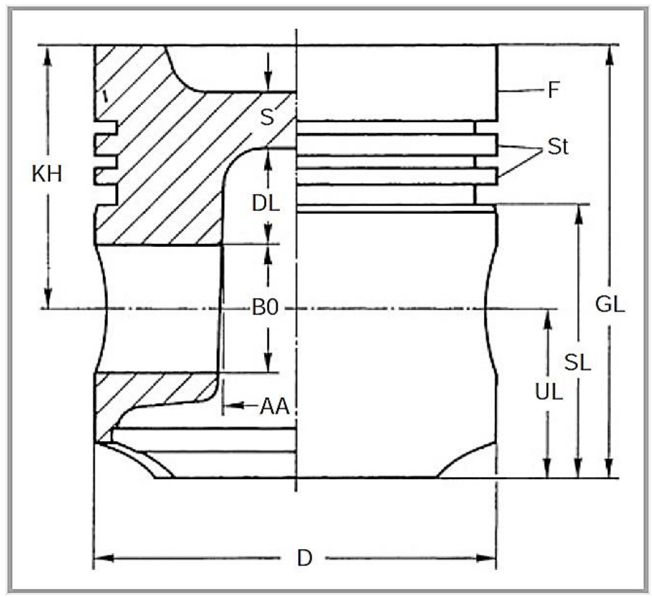


FIGURE 18. PISTON PARAMETERS

Maximum bending stress

$$\sigma_o = p_{max} \cdot \left(\frac{r}{\delta}\right)^2 = 10 \,.\left(\frac{37,4}{8,4}\right)^2 = 198,24 \text{ MPa} \le \sigma_{all} = 200 \text{ MPa}$$

$\boldsymbol{\sigma_{all} = 200}$ **MPa** – allowable bending stress for aluminium alloy

Since this value is only valid for a general and simple type of piston model, we need to consider that the real stress, while the piston is seen as restrained by the cylinder, will be lower.

$$\sigma_o = 0,25 \,. p_{max} \cdot \left(\frac{r}{\delta}\right)^2 = 49,56 \text{ MPa} \le \sigma_{all} = 50 \text{ MPa}$$

$\boldsymbol{\sigma_{all} = 50}$ **MPa** – allowable bending stress with low grooves

The weakest point of the piston is at the location of the scraper ring due to the holes made for oiling.

$$\sigma_t = \frac{F_{p\,max}}{S_x} \le \sigma_{all}$$

$S_x = 0,0021 \text{ m}^2$ - area at section X-X (from CAD model)

$$F_{p\,max} = \frac{\pi \,. D^2}{4} \,. p_{max} = \frac{\pi \,. 0,084^2}{4} \,. 10 = 0,0554 \text{ MN}$$

$$\sigma_t = 26,38 \text{ MPa} \le \sigma_{all} = 30 \text{ MPa}$$

$\boldsymbol{\sigma_{all} = 30}$ **MPa** allowable stress at compression

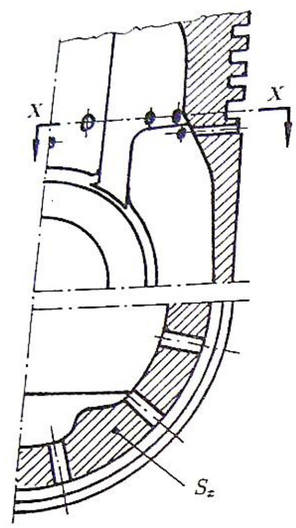


FIGURE 19. SECTION VIEW OF PISTON AT OIL RING LOCATION

Inertial Force at section X-X

$$F_{spx} = m_x \,. r \,. \varpi^2 (1 - \lambda)$$

$m_x = 0,221 \text{ kg}$ - mass of piston above section X-X (from CAD model)

$$\varpi = \frac{\pi \,. n_{max}}{30} = \frac{\pi \,. 5000}{30} = 523,6 \text{ ms}^{-1}$$

$$F_{spx} = m_x \,. r \,. \varpi^2 (1 - \lambda) = 0,221 \,. 0,045 \,. 523,6^2 \left(1 - \left(\frac{45}{156,7}\right)\right) = 0,00194 \text{ MN}$$

Tension load at section X-X

$$\sigma_t = \frac{F_{spx}}{S_x} = \frac{0,00194}{0,0021} = 0,92\ \mathrm{MPa} \leq \sigma_{all} = 4\ \mathrm{MPa}$$

$\sigma_{all} = 4$ MPa - allowable stress at tension at section X-X

Dimensions of the ring land between the first and second piston ring

$d_m = 76\ \mathrm{mm}$

$St = 3,1\ \mathrm{mm}$

Force acting on ring land

$$F_m = \frac{\pi}{4}\left(D^2 - {d_m}^2\right).(\ 0,9.\,p_{max} - 0,22\,.\,p_{max})$$

$$F_m = \frac{\pi}{4}(0,084^2 - 0,076^2).(\ 0,9.10 - 0,22\,.\,10) = 6832,64\ \mathrm{N}$$

Bending Moment at X-X

$$M_o = F_m\,.\frac{D - d_m}{4} = 6832,64\,.\frac{0,084 - 0,076}{4} = 13,66\ \mathrm{Nm}$$

Section modulus

$$W_o = \frac{1}{6}\,.\,\pi\,.\,d_m.(St)^2 = \frac{1}{6}\,.\,\pi\,.\,0,076\,.\,0,0031^2 = 3,824\mathrm{x}10^{-7}\ \mathrm{m}^3$$

Bending stress at restrained surface

$$\sigma_0 = \frac{M_o}{W_o} = \frac{13,66}{3,824\mathrm{x}10^{-7}} = 35,72\ \mathrm{MPa}$$

Section area at restrained surface

$$S = \pi\,.\,d_m\,.\,St = \pi\,.\,0,076\,.\,0,0031 = 7,402\mathrm{x}10^{-4}\ \mathrm{m}^2$$

Shear stress at tension

$$\tau = \frac{F_m}{S} = \frac{6832{,}64}{7{,}402\text{x}10^{-4}} = 9{,}23 \text{ MPa}$$

Final reduced stress at section X-X

$$\sigma_{red} = \sqrt{{\sigma_0}^2 + 3.\tau^2} = \sqrt{(35{,}72)^2 + 3.(9{,}23)^2} = 39{,}13 \text{ MPa} \leq \sigma_{max} = 40 \text{ MPa}$$

$\sigma_{max} = 40$ MPa - allowable reduces stress at section X-X

7.6 Piston pin

Piston pins are designed and checked for bending loads, contact loads and torsional stresses. Lastly a check for correct ovalization is taking place.

Strength check for contact load

$$d = 32 \text{ mm}$$

$$d_1 = 15{,}4 \text{ mm}$$

$$l = 71 \text{ mm}$$

$$b = 34 \text{ mm}$$

$$a = 12 \text{ mm}$$

The total force **F** acting on the piston pin $F = (F_m + F_s)_{max}$

$\mathbf{p_p}$- Contact pressure between the piston pin and the piston

$\mathbf{p_{p\,all} = 39\ MPa}$ -allowable contact pressure

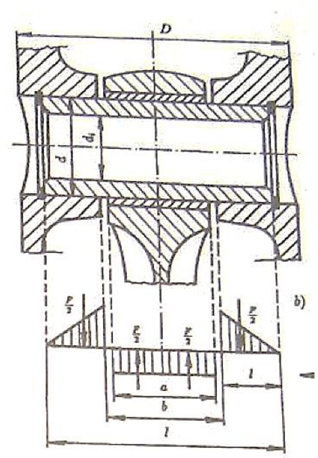


FIGURE 20. CONTACT PRESSURE DISTRIBUTION BETWEEN PISTON PIN, PISTON AND CONNECTING ROD

$$p_p = \frac{F}{d\,.(l - b)} \leq p_{p\,all} \;\rightarrow\; p_p = \frac{44669{,}92}{0{,}032\,.(0{,}071 - 0{,}034)} = 37{,}73 \text{ MPa} \leq p_{p\,all} = 39 \text{ MPa}$$

$\mathbf{p_{cr}}$- Contact pressure between the piston pin and the bearing bush of the connecting rod

$\mathbf{p_{cr\,all} = 50\ MPa}$ -allowable contact pressure

$$p_{cr} = \frac{F}{d\,.\,b} \leq p_{cr\,all} \quad \rightarrow \quad p_{cr} = \frac{44669{,}92}{0{,}034\,.\,0{,}032} = 41{,}06\ \mathrm{MPa} \leq p_{p\,all} = 50\ \mathrm{MPa}$$

Strength Check for bending loads

$$\sigma_o = \frac{M_{o\,max}}{W_o} = \frac{F(l + 2.b - 1{,}5.a)}{1{,}2.d^3(1-\alpha^4)} \leq \sigma_{all}$$

$\boldsymbol{\sigma_{all}} \mathbf{= 150\ MPa}$ -allowable bending stress

$$\boldsymbol{\alpha} = \frac{\mathbf{d_1}}{\mathbf{d}} = \frac{15{,}4}{32} = 0{,}48 \text{ ratio of inner and out diameter of piston pin}$$

$$\sigma_o = \frac{44669{,}92\,.\,(0{,}071 + 2\,.0{,}034 - 1{,}5\,.0{,}012)}{1{,}2\,.\,0{,}032^3(1-0{,}48^4)} = 145{,}16\ \mathrm{MPa} \leq \sigma_{all} = 150\ \mathrm{MPa}$$

Strength Check for Shear stress

$\boldsymbol{\tau_{all}} \mathbf{= 120\ MPa}$ -allowable shear stress

$$\tau_{max} = \frac{0{,}85\,.\,F.\,(1+\alpha+\alpha^2)}{d^2\,.\,(1-\alpha^4)} = \frac{0{,}85\,.44669{,}92\,.\,(1+0{,}48+0{,}48^2)}{0{,}032^2\,.\,(1-0{,}48^4)} = 66{,}98\ \mathrm{MPa} \leq \tau_{all}$$

Checking ovalization of piston pin

$$\Delta d_{ov} = 0{,}09\,.\frac{F}{E\,.\,l}\,.\left(\frac{1+\alpha}{1-\alpha}\right)^3.\,k$$

$$\Delta d_{ov\,max} = (0{,}001 \div 0{,}002).\,d = 0{,}032\ \mathrm{mm}$$

$$k = 1{,}5 - 15\,.\,(\alpha - 0{,}4)^3 = 1{,}5 - 15\,.\,(0{,}48-0{,}4)^3 = 1{,}492$$

$\mathbf{E = 2{,}3x10^5 MPa}$ – Young’s modulus for steel

$$\Delta d_{ov} = 0{,}09 \, . \frac{44669{,}92}{2{,}3x10^5 \, .0{,}071} \, . \left(\frac{1+0{,}48}{1-0{,}48}\right)^3 .1{,}492 = 0{,}0085 \text{ mm} \leq \Delta d_{ov\,max} = 0{,}032 \text{ mm}$$

7.7 Piston rings

Piston rings are strength checked according to the maximum thrust at their operation as well as for bending stress at assembly.

$h = 3 \text{ mm}$

$\frac{d}{a} = (22{,}5 \div 24) = 3{,}7 \text{ mm}$

$m = (2{,}4 \div 4).\, a = 12 \text{ mm}$

$a = 3{,}7 \text{ mm}$

$E = 1x10^5 \text{ MPa}$

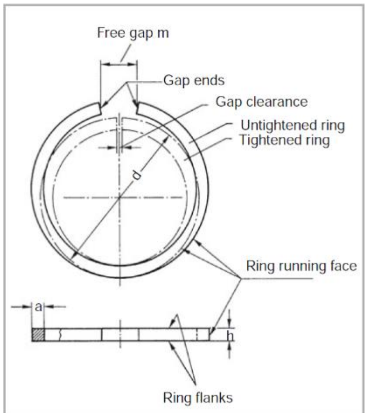


FIGURE 21. PISTON RING DIMENSIONS

Average pressure between piston ring and cylinder

$p_{all} = 0{,}3 \text{ MPa}$ - allowable thrust pressure between piston ring and cylinder

$$p_{avg} = 0{,}152\,.\,E.\frac{\frac{m}{a}}{\left(\frac{d}{a}-1\right)^3 .\frac{d}{a}} = 0{,}152\,.\,1x10^5.\frac{\frac{0{,}012}{0{,}0037}}{\left(\frac{0{,}084}{0{,}0037}-1\right)^3 .\frac{d0{,}084}{0{,}0037}} = 0{,}21 \text{ MPa} \leq p_{all}$$

Maximum tensile stress in bending

$\sigma_{all} = 0{,}3 \text{ MPa}$ - allowable bending stress in piston ring at assembly

$$\sigma_{max} = 3\,.\,p_{avg}.\left(\frac{d}{a}-1\right)^2 = 3\,.\,0{,}21.\left(\frac{0{,}084}{0{,}0037}-1\right)^2 = 296{,}73 \text{ MPa} \leq \sigma_{all} = 300 \text{ MPa}$$

7.8 Connecting rod

Dimensioning of Upper head

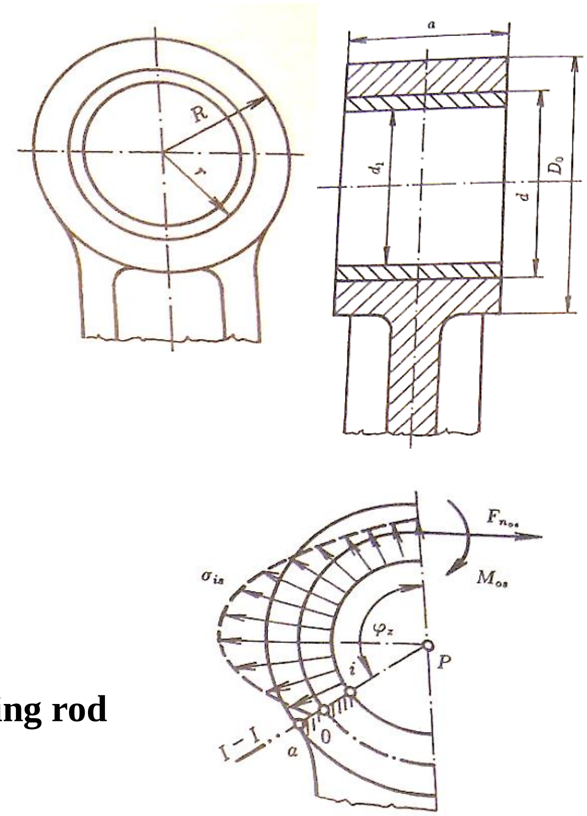


FIGURE 22. CONNECTING ROD UPPER HEAD, FRONT AND SIDE VIEW – FORCE DISTRIBUTION

$$\frac{\mathrm{R}}{\mathrm{r}} = (1{,}2 \div 1{,}3) = \frac{21{,}9}{17{,}4} = 1{,}26$$

Maximum pressure load on connecting rod

$$\mathrm{F}_{\mathrm{p\,max}} = \frac{\pi\,.\,\mathrm{D}^2}{4}\,.\,\mathrm{p}_{\mathrm{max}} = 0{,}0554\ \mathrm{MN}$$

Maximum Inertial Force

$$\mathrm{F'}_{\mathrm{sp}} = \mathrm{m}_{\mathrm{ps}}\,.\,\mathrm{r}\,.\,\varpi^2\,.\,(1+\lambda) = 8730{,}11\ \mathrm{N}$$

$\mathrm{m}_{\mathrm{ps}} = 0{,}859\ \mathrm{kg}$ - mass of piston group

Specific pressure between upper head of connecting rod and piston rod

$\mathrm{a} = 12\ \mathrm{mm} \qquad \mathrm{d}_1 = 32\ \mathrm{mm}$

$$\mathrm{p}_{\mathrm{c}} = \frac{\mathrm{F'}_{\mathrm{p}}}{\mathrm{a}.\,\mathrm{d}_1} \qquad \mathrm{F'}_{\mathrm{p}} = \mathrm{F}_{\mathrm{p\,max}} - \mathrm{F'}_{\mathrm{sp}} = 55400 - 8730{,}11 = 46669{,}89\ \mathrm{N}$$

$$\mathrm{p}_{\mathrm{c}} = \frac{46669{,}89}{0{,}012.\,0{,}032} = 121{,}54\ \mathrm{MPa} \leq \mathrm{p}_{\mathrm{all}} = 200\ \mathrm{MPa}$$

$\mathrm{p}_{\mathrm{all}}$ - allowable specific pressure

Pressing of bush into eye of connecting rod of upper head

$\alpha_{\mathrm{b}} = 1{,}8\mathrm{x}10^{-5}\ \mathrm{K}^{-1}$ - coefficient of linear expansion for bronze (bearing bush)

$\alpha_{\mathrm{b}} = 1\mathrm{x}10^{-5}\ \mathrm{K}^{-1}$ - coefficient of linear expansion for steel (connecting rod)

$E_o = 2{,}2\text{x}10^5\text{MPa}$ – Young's modulus for steel

$E_b = 1{,}15\text{x}10^5\text{MPa}$ – Young's modulus for bronze

$\mu = 0{,}3$ - Poisson coefficient

$\Delta t = 100\ \text{K}$ - heating temperature of upper head

$e = d = 34{,}8\ \text{mm}$ - outer diameter of bearing bush at cold state

$\Delta t = d\,.\,\Delta t(\alpha_b - \alpha_o)$ - change in diameter of bearing bush after heating

$\Delta t = 0{,}0278\ \text{mm}$

Specific surface pressure at heating state between bush and inner surface in upper head eye of connecting rod

$D_0 = 43{,}8\ \text{mm} \quad d = 43{,}8\ \text{mm} \quad d_1 = 43{,}8\ \text{mm}$

$$p' = \frac{e + \Delta t}{d\,.\left(\frac{C_o + \mu}{E_o} + \frac{C_p - \mu}{E_b}\right)} \qquad C_o = \frac{{D_0}^2 + d^2}{{D_0}^2 - d^2} = 4{,}424 \qquad C_p = \frac{d^2 + {d_1}^2}{d^2 - {d_1}^2} = 11{,}95$$

$$p' = \frac{0{,}03483}{d\,.\left(\frac{4{,}424 + 0{,}3}{2{,}2\text{x}10^5} + \frac{11{,}95 - 0{,}3}{1{,}15\text{x}10^5}\right)} = 8{,}15\ \text{MPa}$$

While pressing, the surface tension of the upper head must not exceed limit stress

Stress in outer surface

$$\sigma'_a = p'.\frac{2\,.\,d^2}{{D_0}^2 - d^2} = 27{,}9\ \text{MPa} \leq \sigma_{a\ all} = 100\ \text{MPa}$$

Stress in inner surface

$$\sigma'_i = p'.\frac{{D_0}^2 + d^2}{{D_0}^2 - d^2} = 31{,}1\ \text{MPa} \leq \sigma_{a\ all} = 100\ \text{MPa}$$

Dimensions of lower head

$l = 0{,}0665\ \mathrm{m}$

$l0 = 0{,}0536\ \mathrm{m}$

$s = 0{,}0113\ \mathrm{m}$

$a = 0{,}012\ \mathrm{m}$

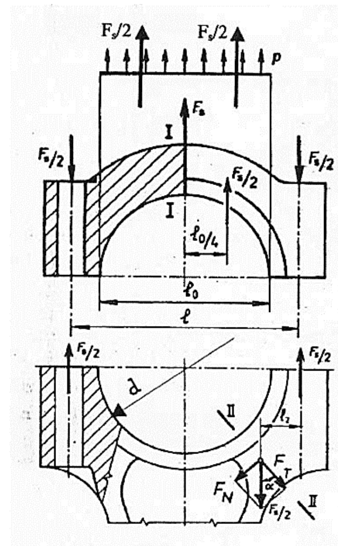


FIGURE 23. FORCES ACTING ON LOWER HEAD OF CONNECTING ROD

Bending stress at Section I-I

$$\sigma_o = \frac{M_0}{W_0} = 0{,}5\,.\,Fs\,.\,(0{,}5\,.\,l - 0{,}25\,.\,l_o).\frac{6}{a\,.\,s^2} \le \sigma_{all}$$

$$\sigma_o = \frac{M_0}{W_0} = 0{,}5\,.\,5721{,}26\,.\,(0{,}5\,.0{,}0665 - 0{,}25\,.0{,}0536).\frac{6}{0{,}012\,.\,0{,}0113^2}$$

$$= 119{,}1\ \mathrm{MPa} \le \sigma_{all}$$

$\sigma_{all} = 150\ \mathrm{MPa}$ - allowable tension stress of material of the connecting rod cover

The resulting reduced normal stress acting in the section II-II

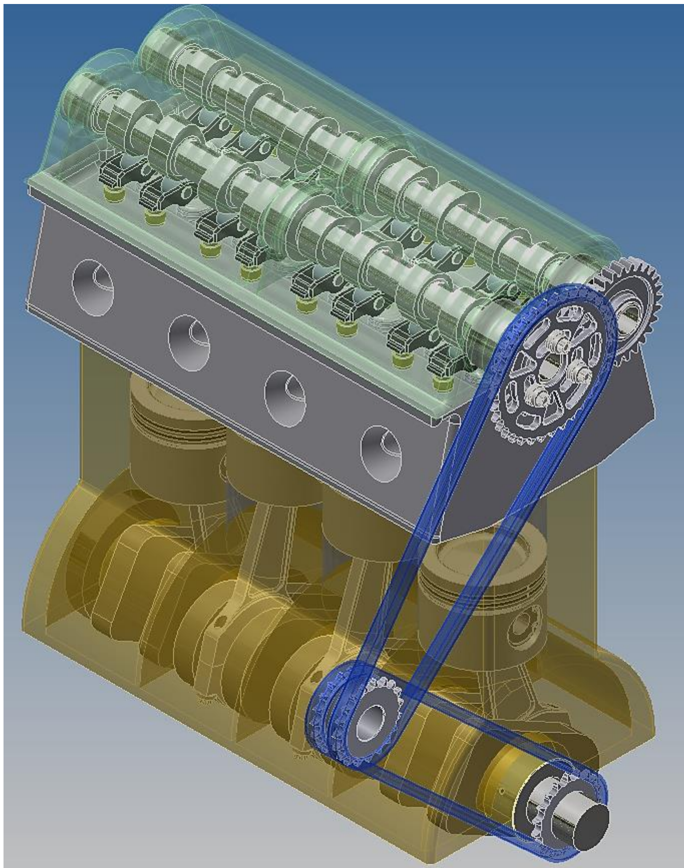
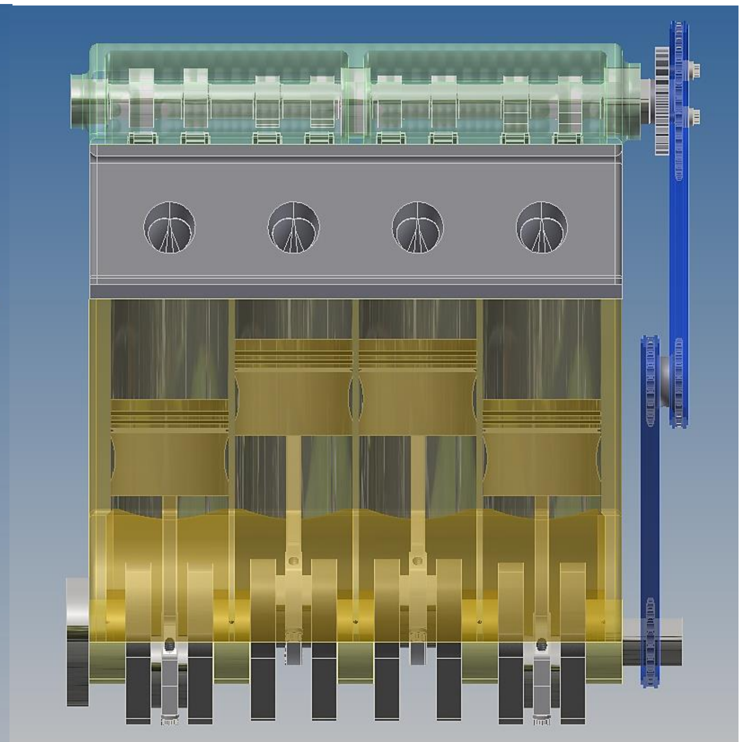
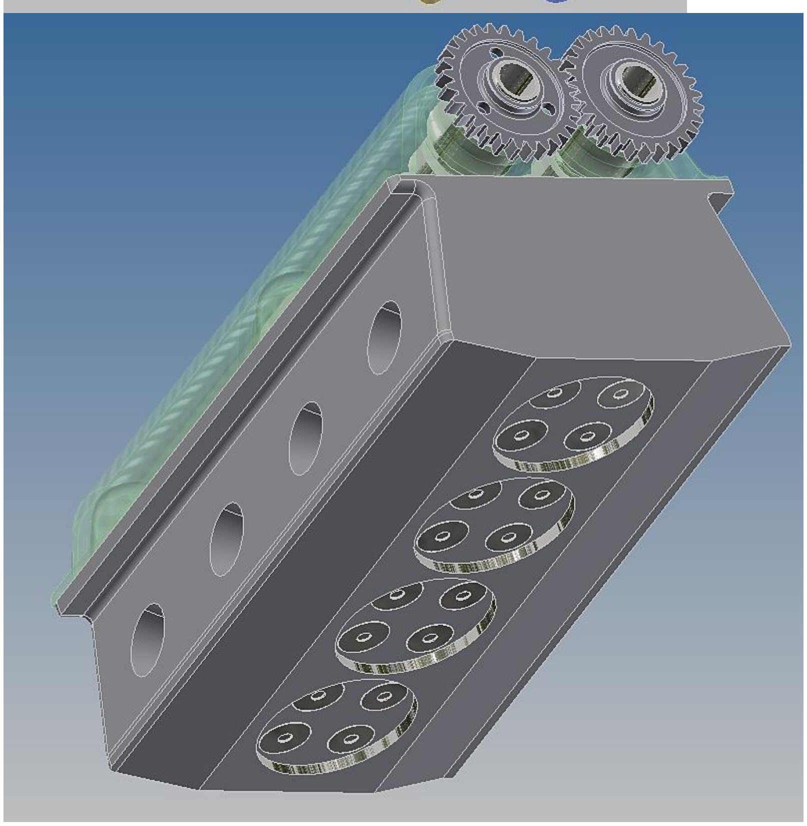
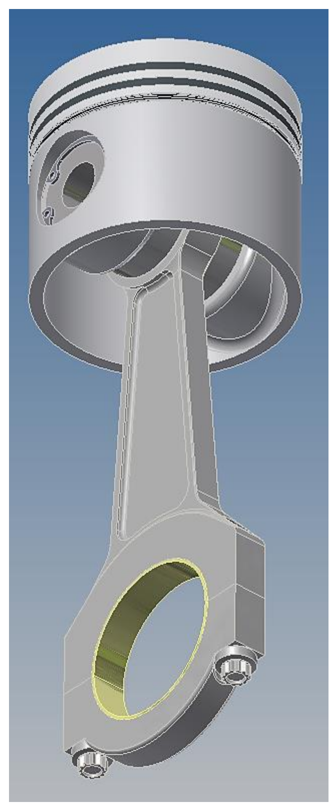
$$\sigma_{red} = \sqrt{(\sigma_t + \sigma_o)^2 + \alpha^2\,.\,\tau^2} \le \sigma_{all}$$

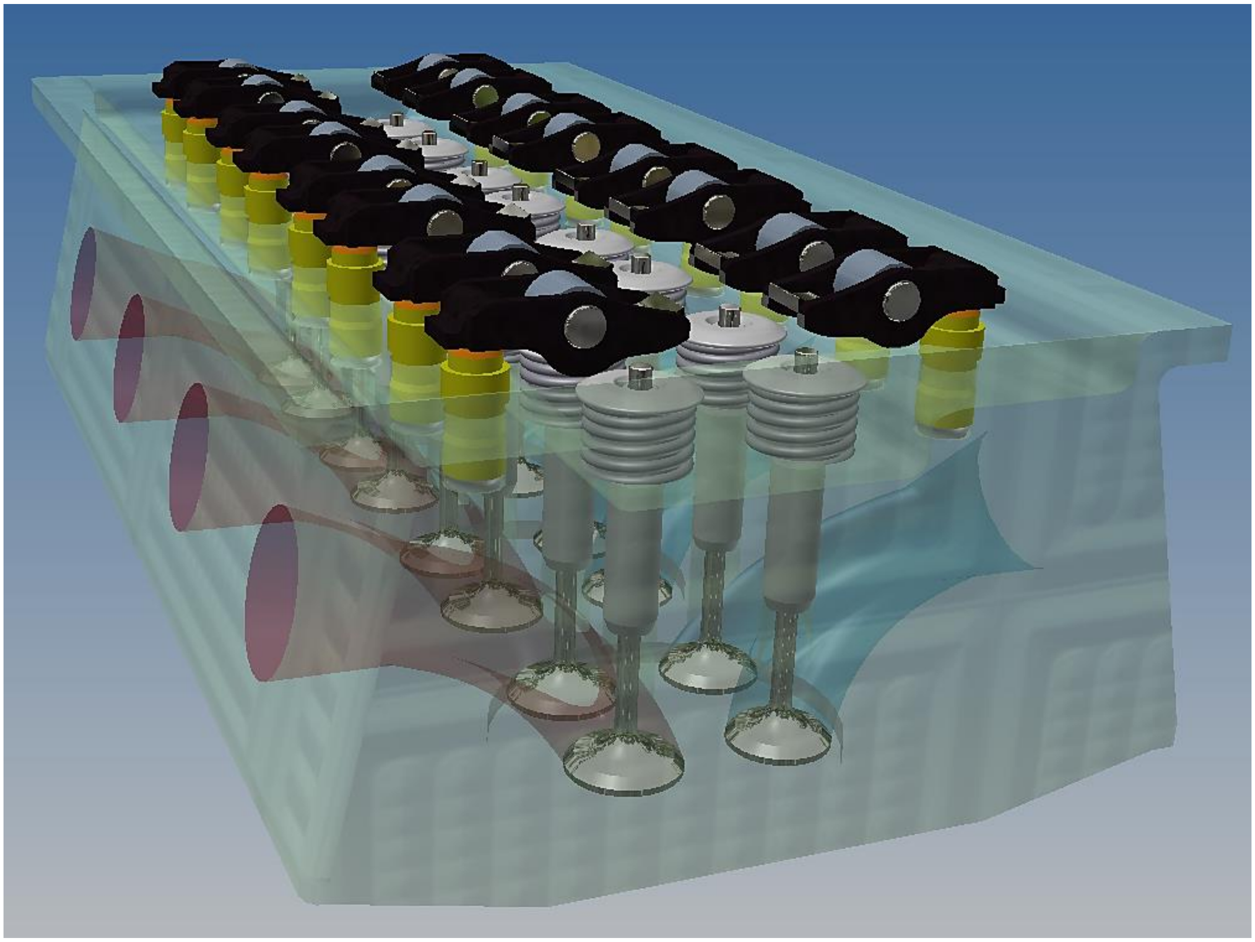
$$\sigma_{red} = \sqrt{(17{,}95 + 92{,}55)^2 + 2^2\,.\,22{,}16^2} = 119{,}1 \le \sigma_{all}$$

$\sigma_{all} = 150\ \mathrm{MPa}$ - allowable tension stress of connecting rod material

$\alpha = 2$ - according to τ_{max} hypothesis

8. CAD Model



9. Conclusion

It has been worked out a design concept for a 4-cylinder 4-stroke compression –ignition engine with the help of Parameters. The most important part of a Crank mechanism is the Strength check followed by a full assembly of the engine. An animation of the working engine has been set up.

Rudolf Diesel was a big pioneer, which showed his Invention. Invented at a time where steam engines ruled the Industrial world and everything relied on only coal. So surely, he placed an important milestone for future Engineers. Today in a world where almost everything is taken for granted, nobody can imagine how it used to be. The Industry relies on it day by day and so do we, the consumers.

The need for even bigger and more powerful engines brigs the designers of such machineries to a challenging time but with the help of the fast growing Software industry, the challenge for better and fast design concepts can help the designers in any step of the technologically advanced production.

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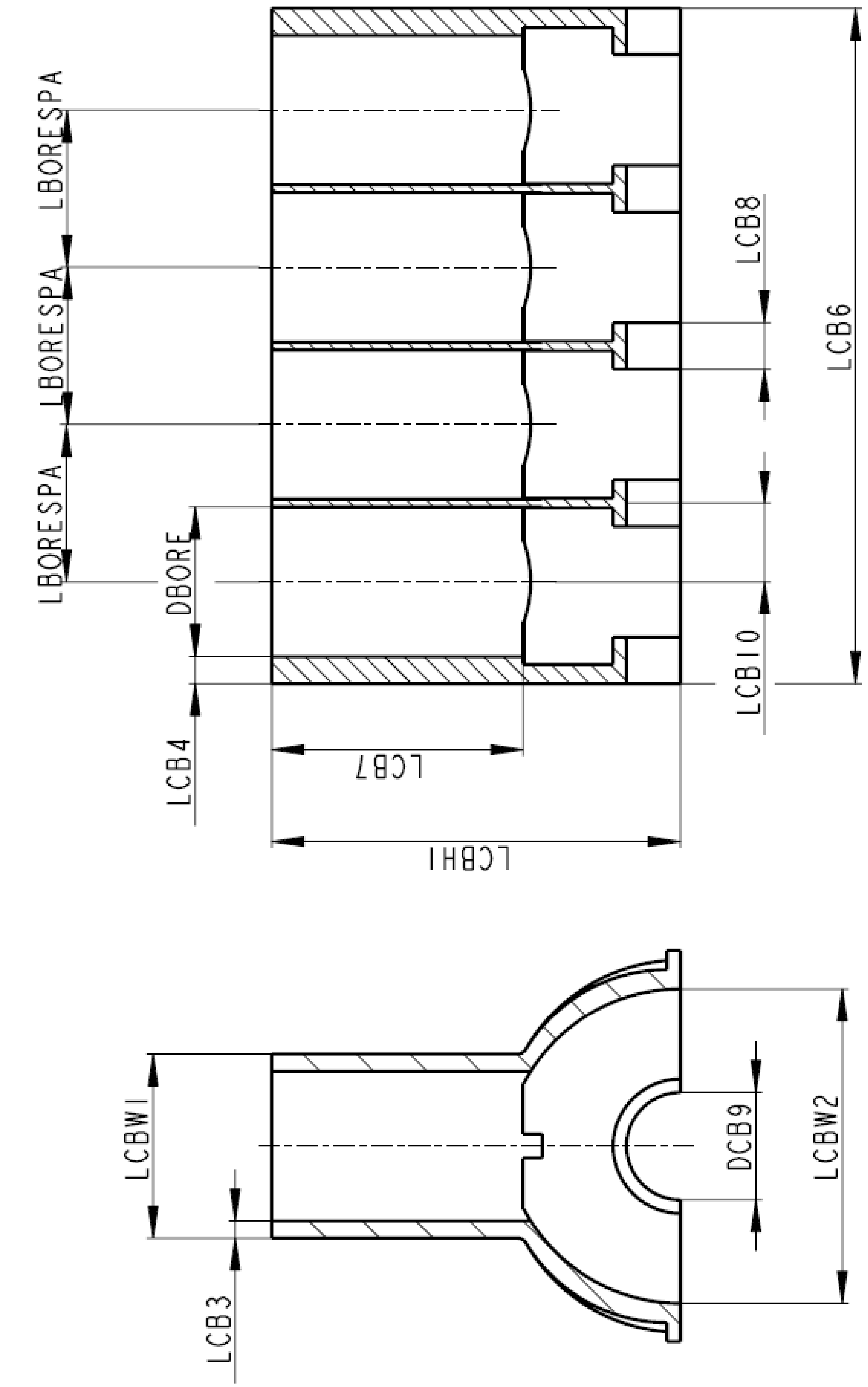
Nomenclature

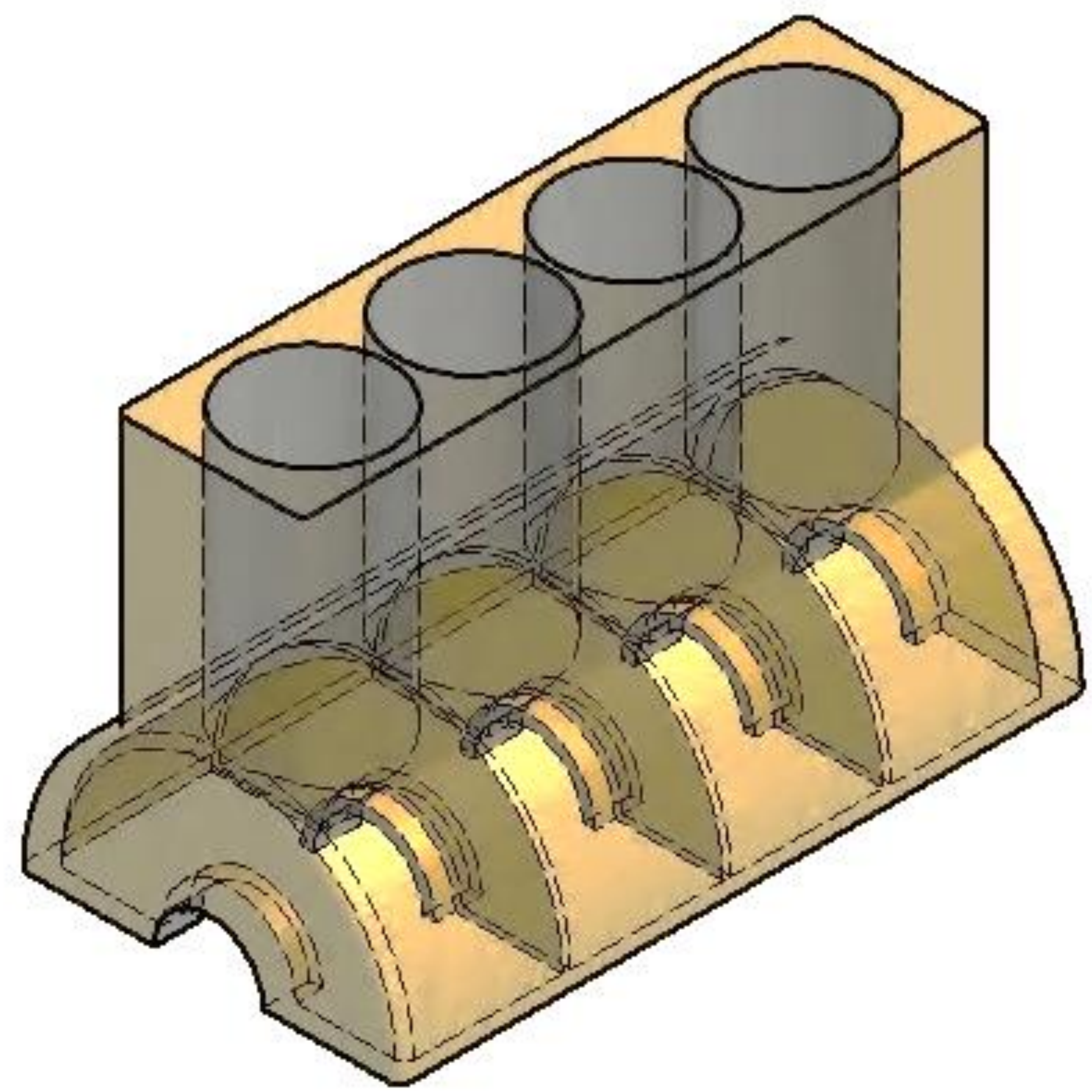
a_2	$[m.s^{-2}]$	Feeding acceleration second order
a_1	$[m.s^{-2}]$	Feeding acceleration first order
$F_{c\ max}$	[N]	Max. Force corresponding to angle β_{max}
F_c	[N]	Force acting in the connecting rod
F_{spx}	[N]	Inertial Force at section X-X of Piston
$F_{s\ max}$	[N]	Max. Inertial Force of feeding mass
m_{rot}	[kg]	Rotating mass of Crankshaft
m_x	[kg]	Mass of Section above X-X of Piston
F_m	[N]	Force from the working medium (gas force, internal force)
p	[Pa]	Pressure of the working medium
p_{max}	[Pa]	max. Combustion chamber pressure
M_o	[Nm]	Bending Moment
σ_{all}	$[m^3]$	Section Modulus
$F_{N\ max}$	[N]	Max. Force corresponding to the angle β_{max}
F_N	[N]	Normal Force acting between the Piston and the cylinder
r_{cb}	[mm]	radius of balancing mass of Crankshaft
m_{cb}	[kg]	balancing mass of Crankshaft
v_1	$[m.s^{-1}]$	Feeding velocity first order
v_2	$[m.s^{-1}]$	Feeding velocity second order
x_1	[mm]	Piston positon first order
x_2	[mm]	Piston positon second order
σ_t	[Pa]	Tension load at Section X-X of Piston
σ_{all}	[Pa]	Allowable bending stress
σ_{red}	[Pa]	reduced stress acting at section II-II at lower head of connecting rod
$\sigma_{o,max}$	[Pa]	Maximum bending stress
a	$[m.s^{-2}]$	Feeding acceleration (first and second order)
E	[Pa]	Young's modulus
L	[mm]	Distance between center of upper and lower head of Connecting rod
MEP, p_e	[Pa]	mean effective pressure
r	[mm]	Crank radius
v	$[m.s^{-1}]$	Feeding velocity of Piston pin (first and second order)
x	[mm]	Distance between top and bottom dead center (Piston position first and second order)
α	[°]	Corresponding angle of crank rotation
α	$[K^{-1}]$	Coefficient of linear thermal expansion of the piston material
β	[°]	Oriented slope angle of connecting rod
Δt	[K]	Temperature difference between both sides of the Piston head
λ	[-]	Ratio between the crank radius and the length of connecting rod
μ	[-]	Poisson's coefficient
ω	$[rad^{-1}]$	Angular velocity of crank rotation

List of Attachments

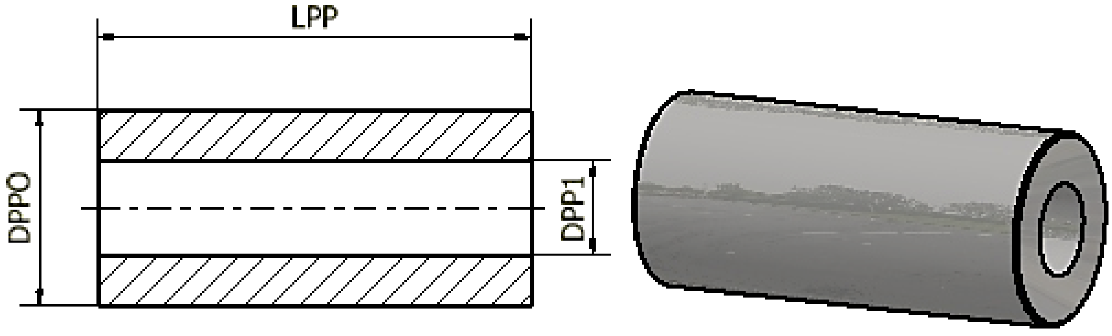
List of Figures



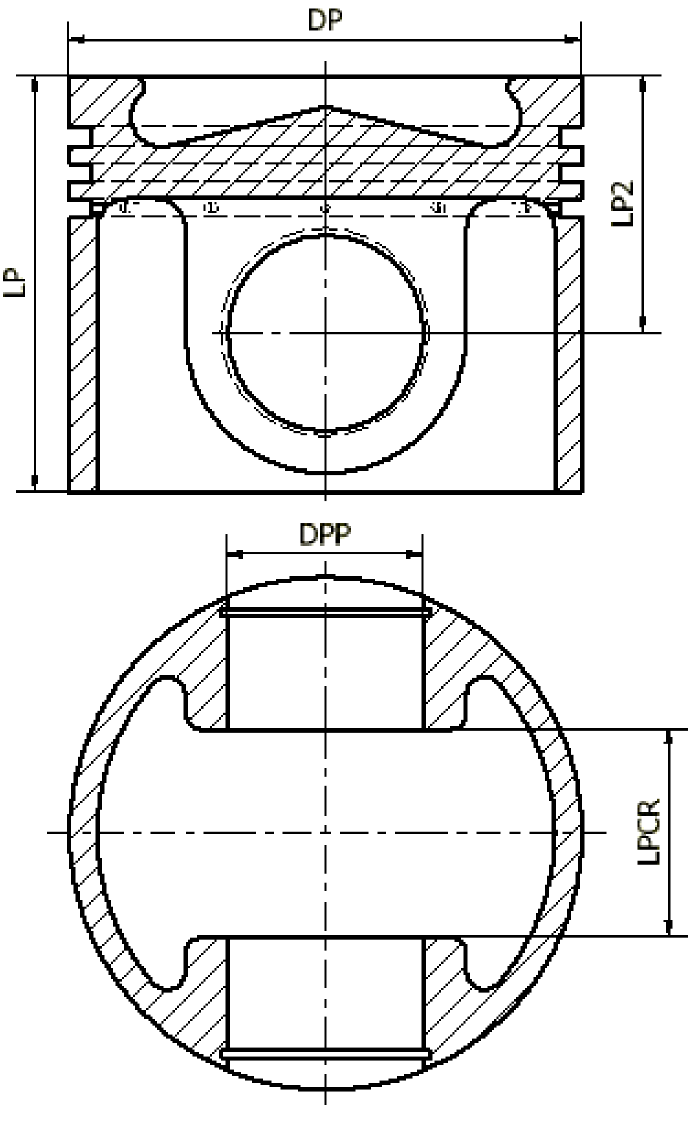
Attachment 1. CYLINDER BLOCK PARAMETERS



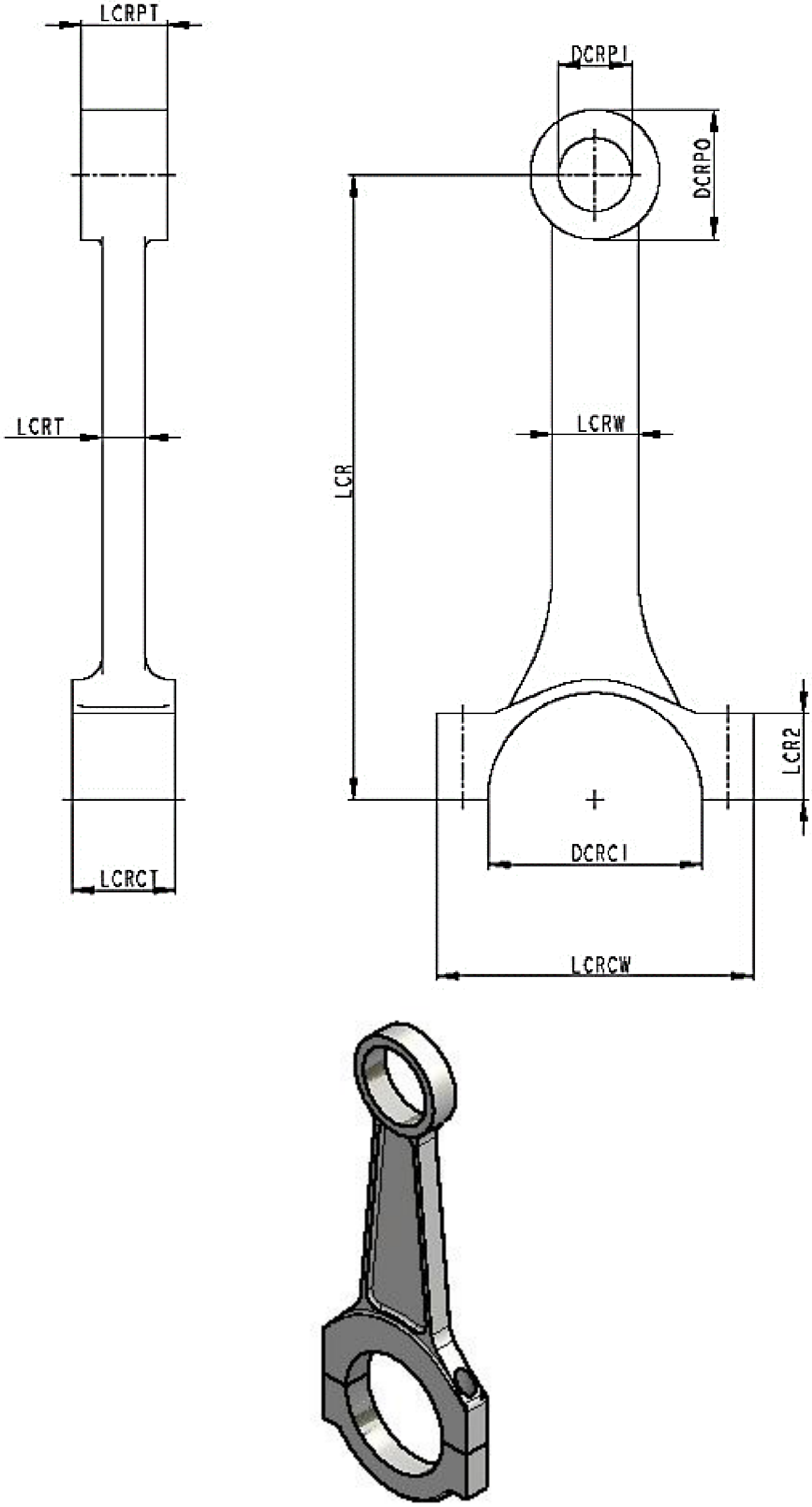
Attachment 2. CYLINDER BLOCK – 3D MODEL



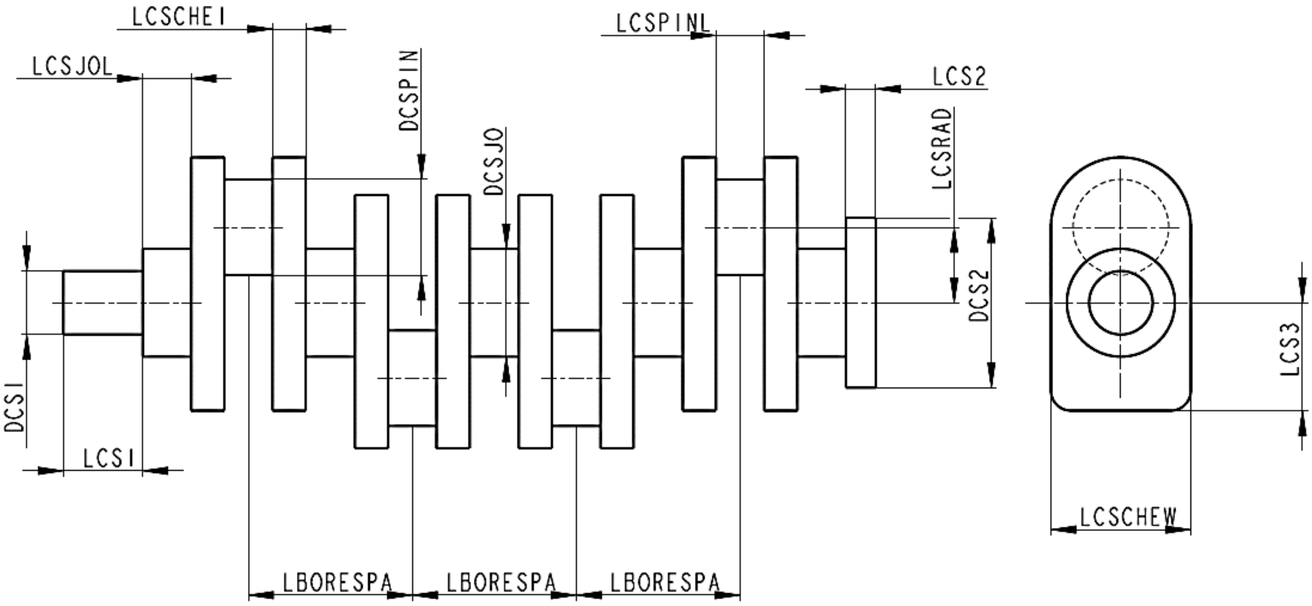
Attachment 3. **PISTON PIN PARAMETERS– 3D MODEL**



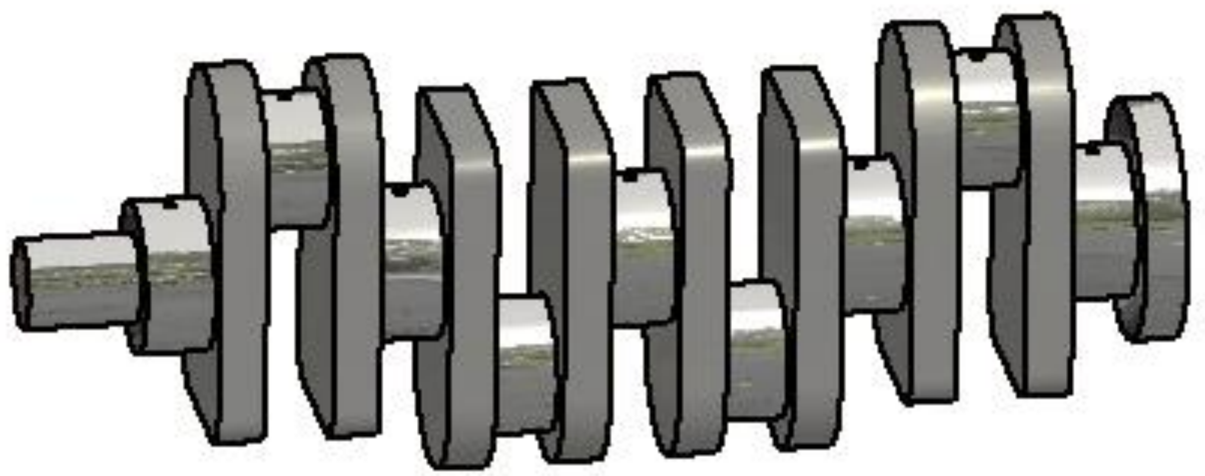
Attachment 4. **PISTON PARAMETERS – 3D MODEL**



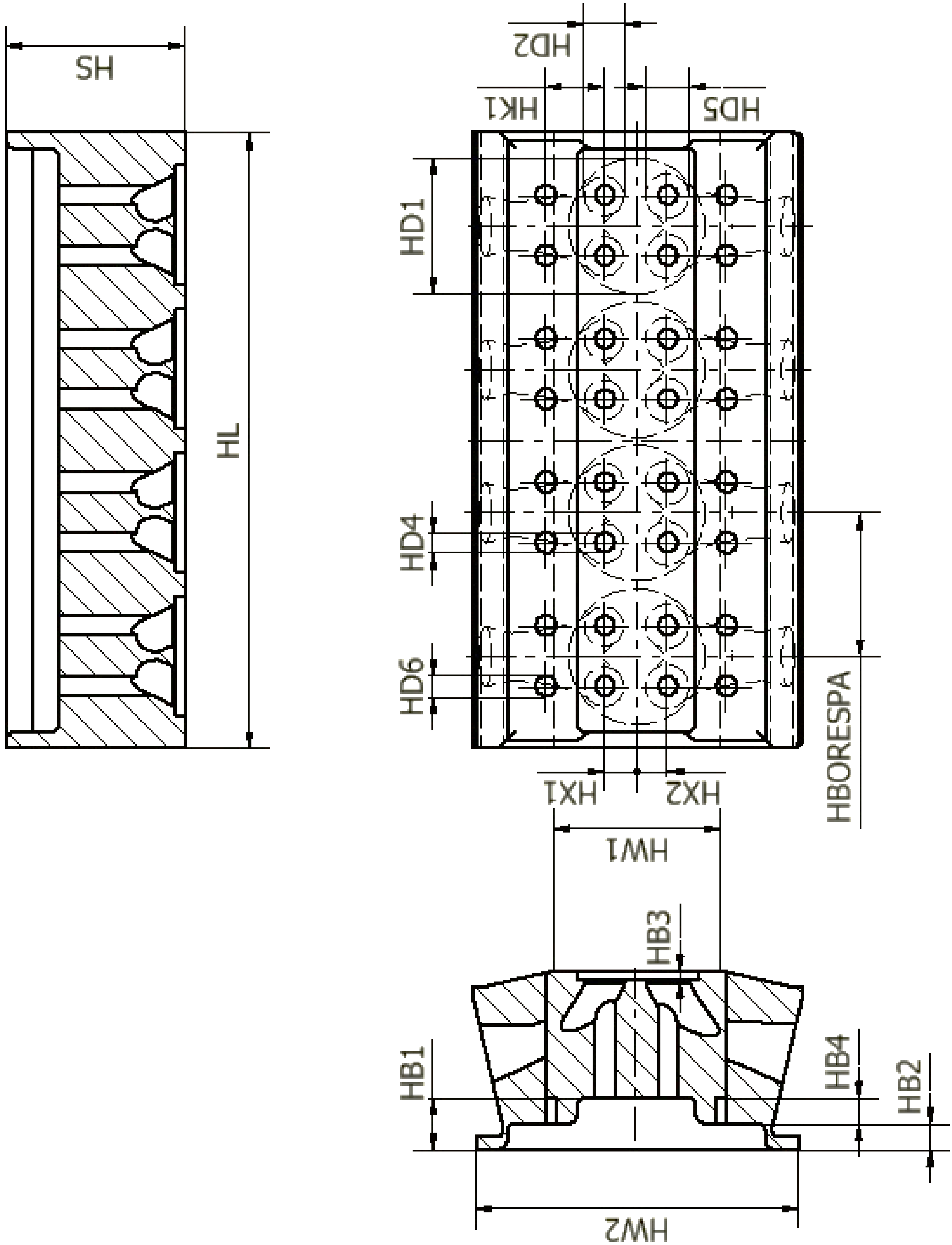
Attachment 5. **CONNECTING ROD PARAMETERS – 3D MODEL**



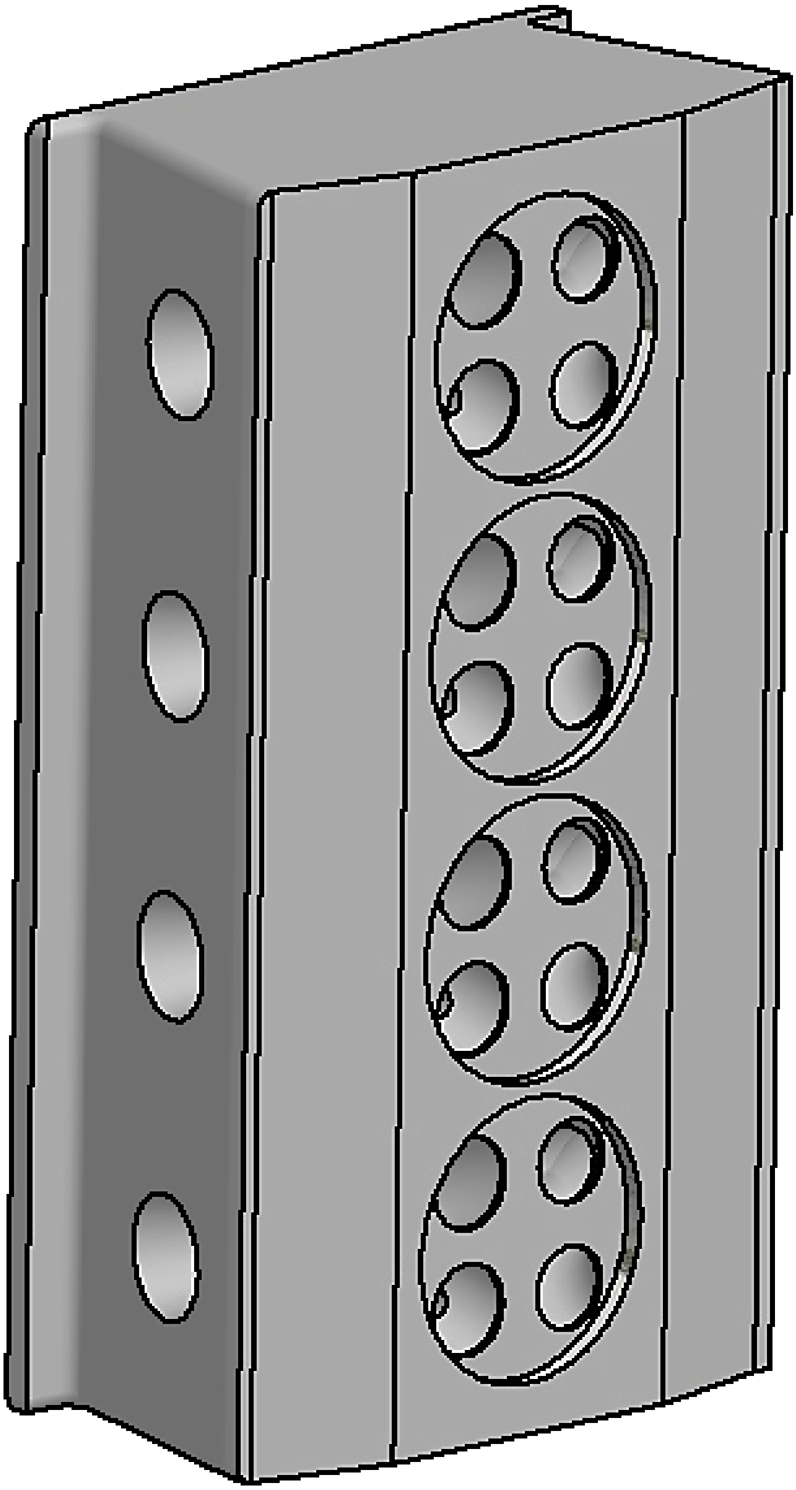
Attachment 6. **CRANKSHAFT PARAMETERS**



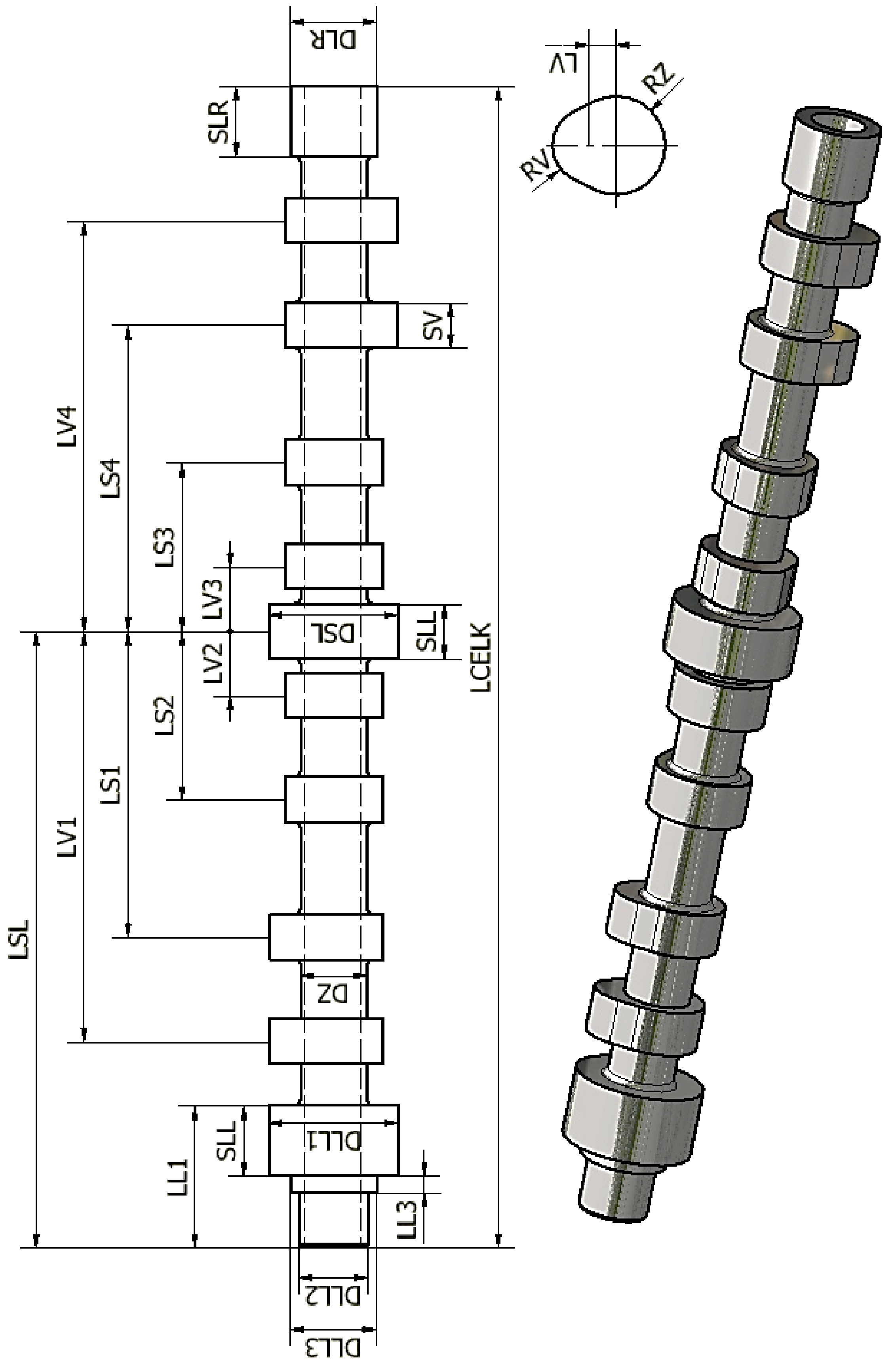
Attachment 7. **CRANKSHAFT – 3D MODEL**



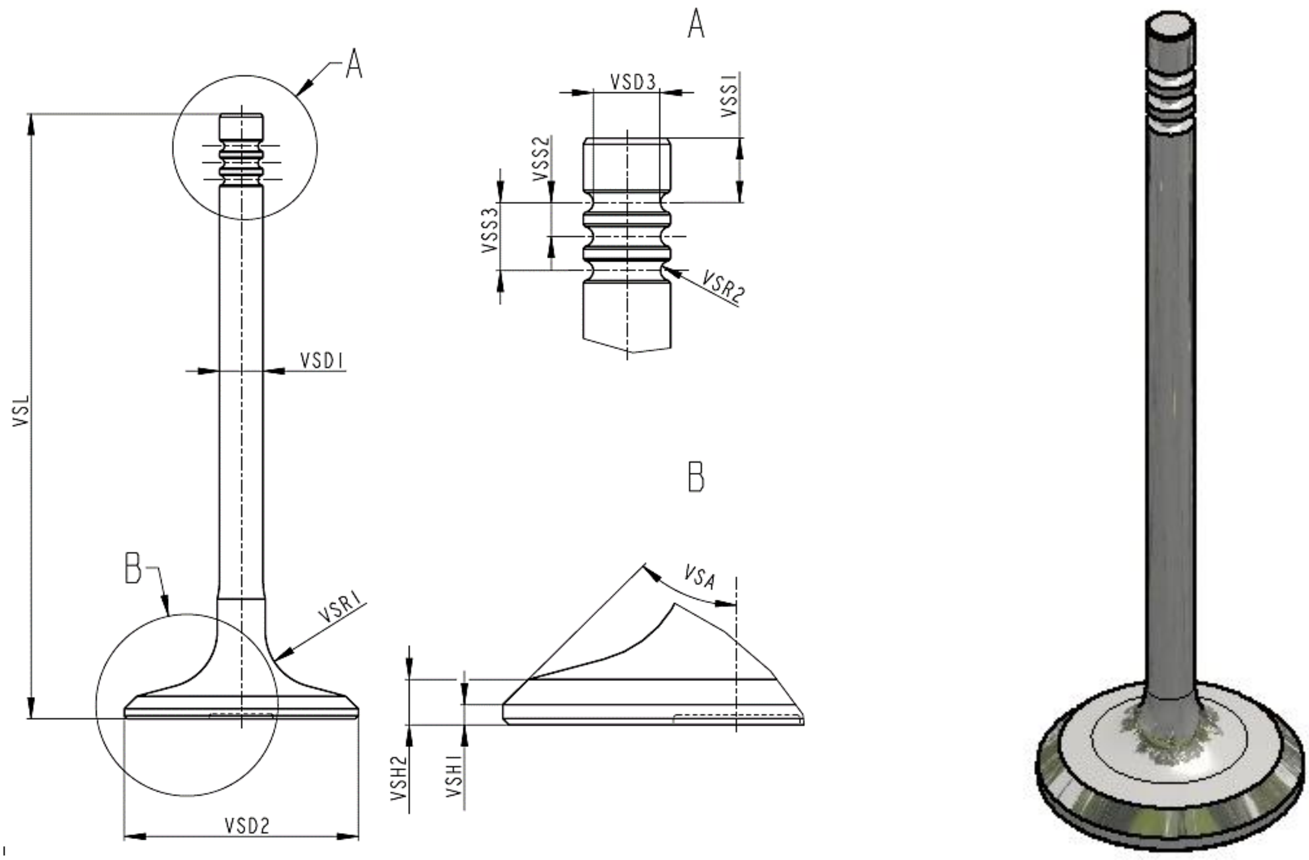
Attachment 8. CYLINDER HEAD PARAMETERS



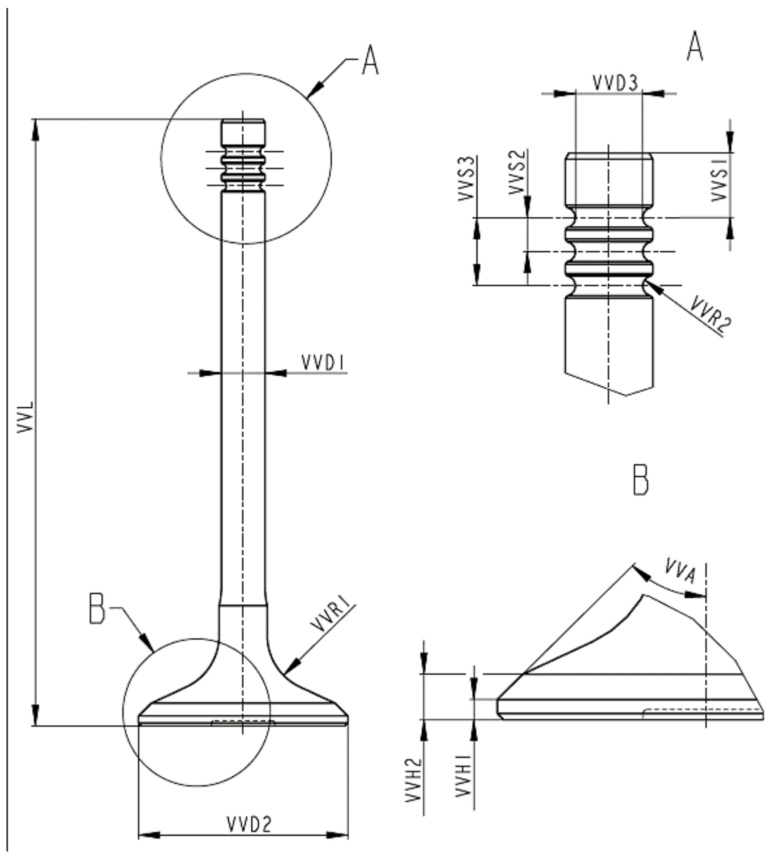
Attachment 9. **CYLINDER HEAD – 3D MODEL**

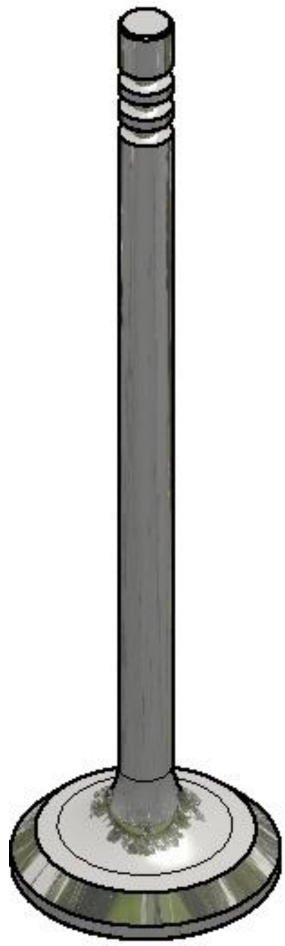


Attachment 10. **CAMSHAFT PARAMETERS – 3D MODE**

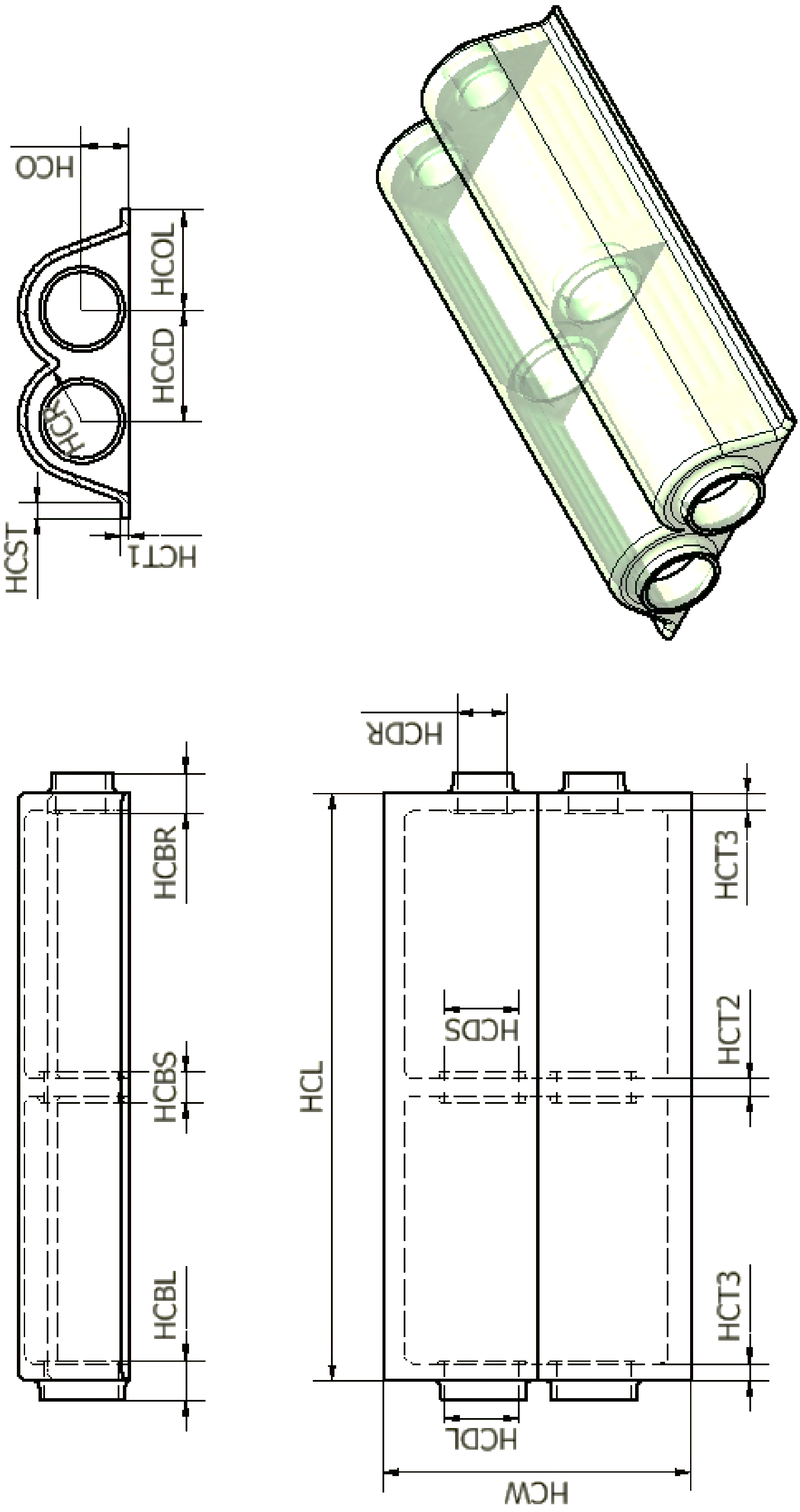


Attachment 11. **INTAKE VALVE PARAMETERS – 3D MODEL**





Attachment 12. **EXHASUT VALVE PARAMETERS – 3D MODEL**



Attachment 13. HEAD COVER PARAMETERS – 3D MODEL

TABLE OF ENGINE PARAMETERS

/* D	84	cylinder bore diameter
/* Z	90	stroke of piston

CYLINDER BLOCK

LCBW1=	103	width of upper area of cylinder block
LCB3 =	9,5	wall thickness at block cross section
DCB9 =	60,2	crankshaft bearing diameter
LCBW2 =	194	width of crank area
LCBH1 =	243,7	height of cylinder block
LCB7 =	149,2	length of bore
LCB4 =	15,1	wall thickness at block longitudinal cross section
DBORE =	84	bore diameter
LCB8 =	26	width of crankshaft bearing
LCB10 =	44,2	distance from bore center to bearing center
LBORESPA =	88,3	bore spacing
LCB6 =	379	length of cylinder block

CYLINDER HEAD

HBORESPA =	88,3	combustion chamber spacing (=bore spacing)
HX1=	20,2	distance from exhaust valve centerline to bore centerline
HX2 =	18,6	distance from intake valve centerline to bore centerline
HD1=	84	diameter of combustion chamber
HL =	379	length of cylinder head
HD3 =	27,2	seat diameter of intake port
HD4 =	12	valve guide hole diameter
HD2 =	24,6	seat diameter of exhaust port
HD6 =	13	hole diameter for hydraulic element
HW1=	103	width of cylinder head
HK1 =	36,2	distance between valve centerline and centerline of hydraulic element
HB2 =	16	depth of plane for hydraulic element
HB4 =	16	depth of hole for hydraulic element
HS =	109,9	height of cylinder head
HW2 =	199	width of upper area of cylinder head
HB1 =	32	depth of plane for spring seat
HB3 =	5,81	depth of combustion chamber

HEAD COVER

HCCD =	71,96	distance between camshafts
HCT3 =	11	thickness of external ribs
HCDL =	48,4	diameter of left bearing
HCT2 =	11	thickness of middle rib
HCL =	379	length of cover
HCW =	199	width of cover
HCO =	30,7	height of bearing centerline
HCR =	36	radius of cover casing
HCT1 =	5	thickness of cover wall
HCST =	10	size of cover outer flange
HCOL =	64,3	distance of bearing centerline to cover edge
HCDR =	32,3	diameter of right bearing
HCBS =	20,4	width of middle bearing
HCBL =	26,0	width of left bearing
HCDS =	48,4	diameter of middle bearing
HCBR =	26,0	width of right bearing

Attachment 14. TABLE OF ENGINE PARAMETERS - PAGE 1

CRANKSHAFT

DCS1 =	34,3	diameter of shaft stud for valvetrain pulley
LCS1 =	42,9	length of shaft stud for valvetrain pulley
DCSPIN =	51,6	diameter of connecting rod journal
LCSRAD =	45,0	length of crank
LCSPINL =	26	length of connecting rod journal
LCS3 =	58,1	length of crankshaft counterweight
LCSCHE1 =	17,9	thickness of crank arm
LCSCHEW =	75	width of crank arm
LCSJOL =	26,5	length of main journal
DCSJO =	58,1	diameter of main journal
DCS2 =	91,5	diameter of flange for flywheel
LCS2 =	16,1	width of flange for flywheel
LBORESPA =	88,3	crank spacing (=bore spacing)

CAMSHAFT

LCELK =	431,9	length of camshaft
LS2 =	62,8	distance of 2. intake cam to middle bearing
LS1 =	113,8	distance of 1. intake cam to middle bearing
SLL =	26,0	width of left bearing
LV =	9,7	distance between centers of base and peak circles of cam
RZ =	18,3	radius of base circle of cam
RV =	14,0	radius of peak circle of cam
LSL =	229,3	distance of middle bearing from left end of shaft
DLL2 =	25,8	diameter of first shoulder of shaft left end
LL1 =	52,8	distance of left bearing from left end of shaft
SLR =	26	width of right bearing
LL3 =	6,2	length of second shoulder of shaft left end
DLL1 =	48,4	diameter of left bearing
DLR =	32,3	diameter of right bearing
SSL =	20,4	width of middle bearing
DSL =	48,4	diameter of middle bearing
DZ =	24,8	diameter of base circle of camshaft
SV =	16,8	width of cam
DLL3 =	32,0	diameter of second shoulder of shaft left end
LV1 =	152,6	distance of 1. exhaust cam to middle bearing
LS4 =	113,8	distance of 4. intake cam to middle bearing
LS3 =	62,7	distance of 3. intake cam to middle bearing
LV2 =	24	distance of 2. exhaust cam to middle bearing
LV3 =	24	distance of 3. exhaust cam to middle bearing
LV4 =	152,6	distance of 4. exhaust cam to middle bearing

CONNECTING ROD

LCRPT =	12,0	width of small end
DCRPI =	34,8	diameter of bearing for piston pin
DCRPO =	43,8	outer diameter of small end
LCR =	156,7	connecting rod length (center to center of bearings)
LCRCW =	79,4	width of big end
LCR2 =	22,0	length of big end
DCRCI =	53,6	diameter of big end bearing
LCRCT =	12,0	width of big end
LCRT =	9,9	thickness of arm
LCRW =	24,6	width of arm [min]

CAP OF CONNECTING ROD

LCR1 =	21,4	length of bearing cap
RCRCO =	38	outer radius of bearing cap
LCRCAPL =	12	width of bearing cap
LCRCW =	79,4	width of big end
DCRC1 =	53,6	diameter of big end bearing

Attachment 15. TABLE OF ENGINE PARAMETERS - PAGE 2

PISTON PIN

LPP =	71,0	length of piston pin
DPPI =	15,4	inner diameter of piston pin
DPPO =	32,0	outer diameter of piston pin

PISTON

DP =	83,8	diameter of piston
LP2 =	42,0	distance of hole centerline from top of piston
LP =	68,0	length of piston
DPP =	32,0	diameter of piston pin
LPCR =	34,0	length of space for small end of connecting rod

EXHAUST VALVE

VVL =	94,0	length of exhaust valve
VVD1 =	5,0	diameter of stem of exhaust valve
VVD2 =	24,6	diameter of head of exhaust valve
VVD3 =	3,2	diameter of slots of exhaust valve stem
VVR1 =	6,7	transition radius between head and stem of exhaust valve
VVR2 =	0,8	radius of slot of exhaust valve
VVS1 =	5,52	distance from head stem to 1. slot of exhaust valve
VVS2 =	2,9	distance of two adjacent slots of exhaust valve
VVS3 =	5,8	distance between 1. and 3. slot of exhaust valve
VVH1 =	1,7	height of unbeveled part of head of exhaust valve
VVH2 =	3,9	height of head of exhaust valve
VVA =	45	angle of bevel on head of exhaust valve

INTAKE VALVE

VSR2 =	0,8	radius of slot of intake valve
VSL =	94,0	length of intake valve
VSD1 =	5,0	diameter of stem of intake valve
VSD2 =	27,2	diameter of head of intake valve
VSD3 =	3,2	diameter of slots of of intake valve
VSR1 =	6,7	transition radius between stem and head of intake valve
VSA =	45	angle of bevel on head of intake valve
VSH2 =	3,9	height of head of intake valve
VSH1 =	1,7	height of unbeveled part of head of intake valve
VSS3 =	5,8	distance between 1. and 3. slot of intake valve
VSS2 =	2,9	distance of two adjacent slots of intake valve
VSS1 =	5,52	distance of head stem to 1. slot of intake valve

Attachment 16. TABLE OF ENGINE PARAMETERS - PAGE 3