FINAL PROJECT

"Design a Radial Engine"

Project and Engineering Department

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CHAPTER 1

I. Radial Engine

The Radial Engine is a reciprocating type internal combustion engine configuration in which the cylinders point outward from a central crankshaft like the spokes on a wheel.

This type of engine was commonly used in most of the aircrafts before they started using turbine engines.

In a Radial Engine, the pistons are connected to the crankshaft with a master-and-articulating-rod assembly. One of the pistons has a master rod with a direct attachment to the crankshaft. The remaining pistons pin their connecting rods` attachments to rings around the edge of the master rod. Four-stroke radials always have an odd number of cylinders per row, so that a consistent every-other-piston firing order can be maintained, providing smooth operation. This is achieved by the engine taking two revolutions of the crankshaft to complete the four strokes. Which means the firing order for a 9-cylinder radial engine is 1,3,5,7,9,2,4,6,8 and then again back to cylinder number 1.This means that there is always a two-piston gap between the piston on its power stroke and the next piston on fire(the piston on compression). If an even number of cylinders was used the firing order would be something similar to 1,3,5,7,9,2,4,6,8,10 which leaves a three-piston gap between firing pistons on the first crank shaft revolution, and only one-piston gap on the second crankshaft revolution. This leads to an uneven firing order within the engine, and is not ideal.

The Four-stroke consequence of every engine is:

a) Intake
b) Compression
c) Power
d) Exhaust

Most radial engines use overhead poppet valves driven by pushrods and lifters on a cam plate which is concentric with the crankshaft, with a few smaller radials. A few engines utilize sleeve valves instead.

II. History of the Radial Engine

The very first design of internal combustion aero engine made was that of Charles Manly, who built a five-cylinder radial engine in 1901 for use with Langley`s ‘aerodrome’ , as the latter inventor decided to call what has since become known as the aero-plane. Manly made a number of experiments, and finally decided on radial design, in which the cylinders are so rayed round a central crank-pin that the pistons act successively upon it. By this arrangement a very short and compact engine is obtained, with a minimum of weight, and a regular crankshaft rotation and perfect balance of inertia forces.

When Manly designed his radial engine, high speed internal combustion engines were in their infancy, and the difficulties in construction can be partly realized when the lack of manufacturing methods for this high-class engine work, and the lack of experimental data on the various materials, are taken into account. During its tests, Manly`s engine developed 52.4 brake horsepower at a speed of 950 revolutions per minute, with the remarkably low weight of only 1.09 kg per horsepower, this latter was increased to 1.64 kg when the engine was completed by the addition of ignition system, radiator, petrol tank, and all accessories, together with the cooling water for the cylinders.

In Manly`s engine, the cylinders were of steel, machined outside and inside to 1.625 of a mm thickness. On the side of the cylinder, at the top end, the valve chamber was brazed, being machined from a solid forging. The casing which formed the water-jacket was of sheet steel, 0.52 of a mm in thickness, and this also was brazed on the cylinder and to the valve chamber. Automatic inlet valves were fitted, and the exhaust valves were operated by a cam which had two points, 180 degrees apart. The cam was rotated in the opposite direction to the engine at one -quarter engine speed. Ignition was obtained by using a one-spark coil and vibrator for all cylinders, with a distributor to select the right cylinder for each spark – this was before the days of the high-tension magneto and the almost perfect ignition systems that makers now employ. The scheme of ignition for this engine was originated by Manly himself, and he also designed the sparking plugs fitted in the tops of the cylinders. Through fear of trouble resulting if the steel pistons worked on the steel cylinders, cast iron liners were introduced in the latter 1.625of a mm thick.

The connecting rods of this engine were of virtually the same type as is employed on nearly all modern radial engines. The rod for one cylinder had a bearing along the whole of the crank pin, and its end enclosed the pin. The other four rods had bearings upon the end of the firs rod, and did not touch the crank pin. The bearings of these rods did not receive any of the rubbing effect due to the rotation of the crank pin, the rubbing on them being only that of the small angular displacement of the rods during each revolution, thus there was no difficulty experienced with the lubrication.

Another early example of the radial type of engine was French Anzani, of which type one was fitted to the machine with which Bleriot first crossed the English Channel—this was of 25 horse-powers. The earliest Anzani engines were of three-cylinder fan type, one cylinder being vertical, and the other two placed at an angle of 72 degrees on each side, as the possibility of over lubrication of the bottom cylinders was feared if a regular radial construction were adopted. In order to overcome the unequal balance of this type, balance weights were fitted inside the crankcase.

The final development of this three-cylinder radial was the 'Y' type of engine in which the cylinders were regularly disposed at 120 degrees apart, the bore was 4.1, stroke 4.7 inches and the power developed was 30 brake horse-powers at 1300 revolutions per minute.

Critchley`s list of aero engines being constructed in 1910 shows twelve of the radial type, with powers of between 14 and 100 horse-power and with from three to ten cylinder—this last is probably the greatest number of cylinders that can be successfully arranged in circular form. Of the twelve types of 1910, only two were water-cooled, and it is to be noted that these two ran at the slowest speeds and had the lowest weight per horse- power of any.

The Anzani radial was considerably developed special attention being paid to this type by its makers and by 1914 the Anzani list comprised seven different sizes of air-cooled radials. Of these the largest had twenty cylinders, developing 200 brake horse-powers—it was virtually a double radial—and the smallest was the original 30 horse-power three-cylinder design. A six-cylinder model was formed by a combination of two groups of three cylinders each, acting upon a double-throw crankshaft; the two crankpins were set at 180 degrees to each other, and the cylinder groups were staggered by an amount equal to the distance between the centers of the crank pins. Ten-cylinder radial engines are made with two groups of five cylinders acting upon two crank pins set at 180 degrees to each other, the largest Anzani 'ten' developed 125 horse-power at 1200 revolutions per minute, the ten cylinders being each 114.3 mm in bore with stroke of 149.86 mm, and the weight of the engine being (1.678 kg) per horse-power. In the 200 horse-power Anzani radial the cylinders are arranged in four groups of five each, acting on two crank pins. The bore of the cylinders in this engine is the same as in the three-cylinder, but the stroke is increased to 139.7 mm. The rated power is developed at 1300 revolutions per minute, and the engine complete weights 1.5422 kg per horse-power.

With this 200 horse-powers Anzani, a petrol consumption of as low as 0.222 kg of fuel per brake horse-power per hour has been obtained, but the consumption of lubricating oil is compensatingly high, being up to one-fifth of the fuel used. The cylinders are set desaxe with the crank shaft, and are of cast-iron, provided with radiating ribs for air-cooling; they are attached to the crank case by long bolts passing through bosses at the top of the cylinders, and connected to other bolts at right angles through the crank case. The tops of the cylinders are formed flat, and seats for the inlet and exhaust valves are formed on them. The pistons are cast-iron, fitted with ordinary cast-iron spring rings. An aluminum crank case is used, being made in two halves connected together by bolts, which latter also attach the engine to the frame of the machine. The crankshaft is of nickel steel, made hollow, and mounted on bell-bearings in such a manner that practically a combination of ball and plain bearings is obtained; the central web of the shaft is bent to bring the centers of the crank pins as close together as possible, leaving only room for the connecting rods, and the pins are 180 degrees apart. Nickel steel valves of the cone-seated, poppet type are fitted, the inlet valves being automatic, and those for the exhaust cam-operated by means of pushing rods. With an engine having such a number of cylinders a very uniform rotation of the crankshaft is obtained, and in actual running there are always five of the cylinders giving impulses to the crankshaft at the same time.

An interesting type of pioneer radial engine was the Farcot, in which the cylinders were arranged in a horizontal plane, with a vertical crankshaft which operated the air-screw through bevel gearing. This was an eight-cylinder engine, developing 64 horse-powers at 1200 revolutions per minute. The R.E.P. type, in the early days, was a 'fan' engine, but the designer, M. Robert Pelterie, turned from this design to a seven-cylinder radial engine, which at 1100 revolutions per minute gave 95 horse-powers. Several makers entered into radial engine development in the years immediately preceding the War, and in 1914 there were some twenty-two different sizes and types, ranging from 30 to 600 horse-powers, being made, according to report; the actual construction of the latter size at this time, however is doubtful.

Probably the best example of radial construction up to the outbreak of War was the Salmson (Canton-Unne) water-cooled, of which in 1914 six sizes were listed as available. Of these the smallest was a seven-cylinder 90 horse-power engine and the largest, rated at 600 horse- power, had eighteen cylinders. These engines, during the War, were made under license by the Dudbrige Ironworks in Great Britain.

The patent planetary gear gives exactly the same stroke to all pistons. The complete 200 horse power engine has fourteen cylinders, of forged steel machined all over, and so secured to the crank case that anyone can be removed without parting the crank case. The water-jackets are of spun copper brazed on to the cylinder, and corrugated so as to admit of free expansion; the water is circulated by means of a centrifugal pump. The pistons are of cast-iron, each fitted with three rings, and the connecting rods are connected to a central collar, carried on the crank pin by two ball-bearings. The crankshaft has a single throw, and is made in two parts to allow the cage for carrying the big end-pins of the connecting rods to be placed in position. The casting is in two parts, on one of which the brackets for fixing the engine are carried, while the other part carries the valve-gear. Bolts secure the two parts together. The mechanically operated steel valves on the cylinders are each fitted with double springs and the valves are operated by rods and levers. Two Zenith carburetors are fitted on the rare half of the crank case and short induction pipes are led to each cylinder; each of the carburetors is heated by the exhaust gases. Ignition is by two high tension magnetos, and a compressed air self-starting arrangement is provided. Two oil pumps are fitted for lubricating purposes, one of which forces oil to the crankshaft and connecting-rod bearings while the second forces oil to the valve gear, the cylinders being so arranged that the oil which flows along the walls cannot flood the lower cylinders. The engine operates upon a six-stroke cycle, a rather rare arrangement for internal combustion engines of the electrical ignition type; this is done in order to obtain equal angular intervals for the working impulses imparted to the rotating crankshaft as the cylinders are arranged in groups of seven, and all act upon the one crankshaft. The angle, therefore between the impulses is 77 1/7 degrees. A diagram is inset giving a side view of the engine in order to show the grouping of the cylinders.

The 600 horse-power Salmson engine was designed with a view to fitting to airships, and was in reality two nine-cylindered engines, with a gear-box connecting them; double air screws were fitted, and these were so arranged that either or both of them might be driven by either or both engines; in addition to this, the two engines were complete and separate engines as regards carburetion and ignition, so that they could be run independently of each other. The cylinders were exceptionally 'long stroke', being 149.86 mm bore to 210.05 mm stroke, and the rated power was developed at 1200 revolutions per minute, the weight of the complete engine being only 1.859 kg per horse-power at the normal rating.

A type of engine specially devised for airship propulsion is that in which the cylinders are arranged horizontally instead of vertically, the main advantages of this form being the reduction of head resistance and less obstruction to view of the pilot. A casing, mounted on the top of the engine, supports the air-screw, which is driven through bevel gearing from the upper end of the crankshaft. With this type of engine a better rate of air-screw efficiency is obtained by gearing the screw down to half the rate of revolution of the engine, this giving a more even torque. The petrol consumption of the type is very low, being only 0.2177 kg per horse-power per hour, and equal economy is claimed as regards lubricating oil, a consumption of as little as 0.018 kg per horse-power per hour being claimed.

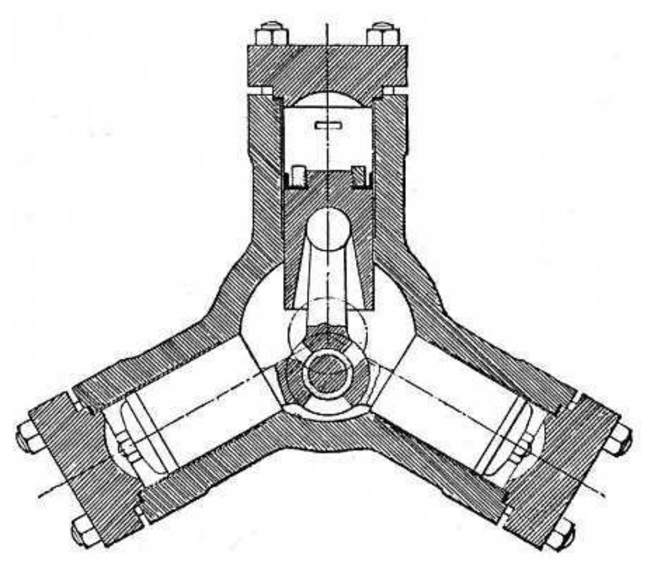
Certain American radial engines were made previous to 1914, the principle being the Albatross six-cylinder engines of 50 and 100 horse-powers. Of these the smaller size was air cooled. With cylinders of 114.3 mm bore and 13 mm stroke, developing the rated power at 1230 revolutions per minute, with a weight of about 2.267 kg per horse-power. The 100 horse-power size had cylinders of 139.7 mm bore, developing its rated power at 1230 revolutions per minute, and weighing only 1.247kg per horse power. This engine was markedly similar to the 6-cylinder Anzani, having all the valves mechanically operated, and with auxiliary exhaust ports at the bottoms of the cylinders, overrun by long pistons. These Albatross engines had their cylinders arranged in two groups of three, with each group of three pistons operating on one of two crank pins, each 180 degrees apart.

The radial type of engine, thanks to Charles Manly, had the honor of being the first in the field as regards aero work. Its many advantages, among which may be specially noted the very short crankshaft as compared with vertical, Vee, or ‘broad arrow’ type of engine, and consequent greater rigidity, ensure it consideration by designers of to-day, and render it certain that the type will endure. Enthusiasts claim that the ‘broad arrow’ type, or Vee with a third row of cylinders inset between the original two, is just as much a development from the radial engine as from the vertical and resulting Vee; however this may be, there is a place for the radial type in air-work for as long as the internal combustion engine remains as a power plant.

III. Radial engines nowadays

At least five companies build radials today. Vedeneyev engines produces the M-14P model, 360 Hp (270kW)(up to 450 Hp (340kW) radial used on Yakovlevs and Sukhoi, Su-26 and Su-29 aerobic aircraft. The M-14P has also found great favor among builders of experimental aircrafts, such as the Culp`s Special and Culps Sopwith Pup, Pitts S12 “Monster” and the Murphy “Moose”. Engines with 110 Hp (82kW) 7-cylinders and 150 Hp (110 kW) 9-cylinders are available from Australia’s Rotec Engineering. HCI Aviation offers the R180 5-cylinders (75 Hp (56kW)) and R220 7-cylinders (110 Hp (82kW)), available “ready to fly” and as a “build it yourself” kit. Verner Motor from the Czech Republic now builds several radial engines. Models range in power form 71 Hp (53 kW) to 172 Hp (128 kW). Miniature radial engines for model airplane use are also available from Seidel in Germany, OS and Saito Seisakusho of Japan, and Technopower in the USA. The Saito firm is known for making 3 different sizes of 3-cylinder engines, as well as a 5-cylinder example, as the Saito firm is the specialist in making a large line of miniature four-stroke engines for model use in both methanol-burning glow plug and gasoline-fueled spark plug ignition engine formats.

CHAPTER 2



Radial Engine Characteristics

Rpm =6000

Piston diameter D_p= 70 mm.

Master-rod length L_{mr}=120 mm.

Crank Length R_{cr}=30mm.

I. Kinematical and Dynamical Calculations

1. Ratio

Between the crank of the crankshaft and the master-rod length:

$$\lambda = \frac{R_{cr}}{L_{mr}} = \frac{0,03}{0,12} = 0,25$$

2. Angular velocity

Specifies the angular velocity of the object and the axes about which the object is rotating.

$$\omega = \frac{\pi.n}{30} = \frac{3,14.6000}{30} = 628,32s^{-1}$$

3. Current Piston Stroke

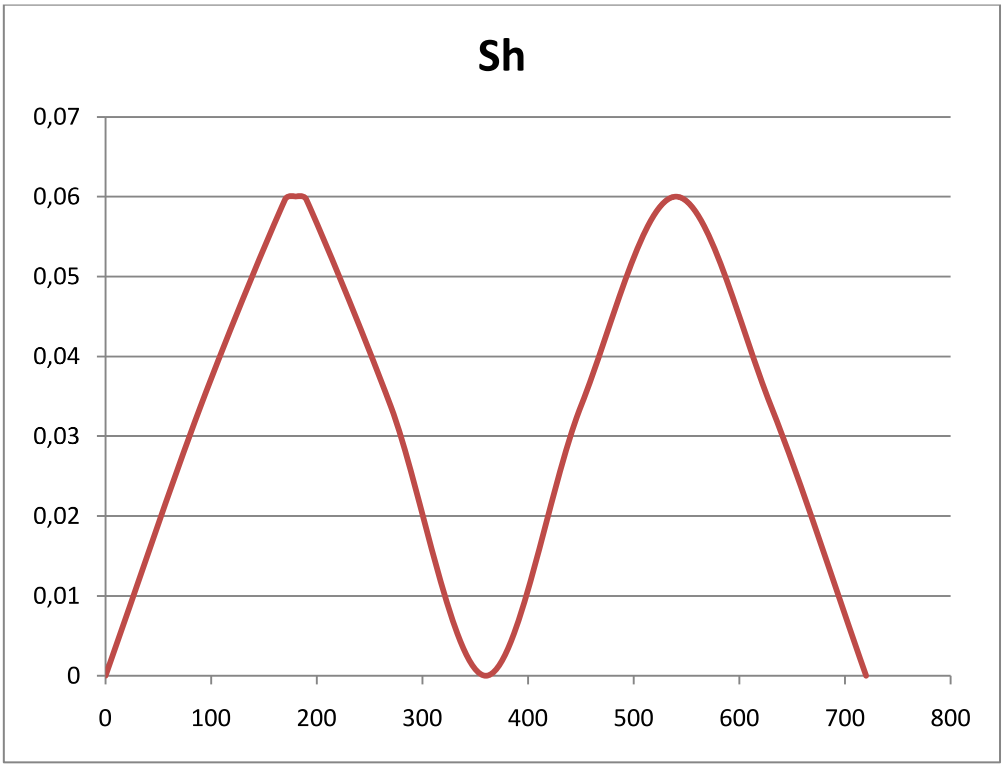
Reciprocating motion, used in reciprocating engines and other mechanisms is back-and-forth motion. Each cycle of reciprocation consists of two opposite motions, there is a motion in one direction and then a motion back in the opposite direction. Each of them is called a **stroke.**

$$S_h = R.\left[\left(1-\cos\varphi\right)+\frac{\lambda}{4}\left(1-\cos 2\varphi\right)\right]$$

In the table below I will show you the behavior of the master rod.

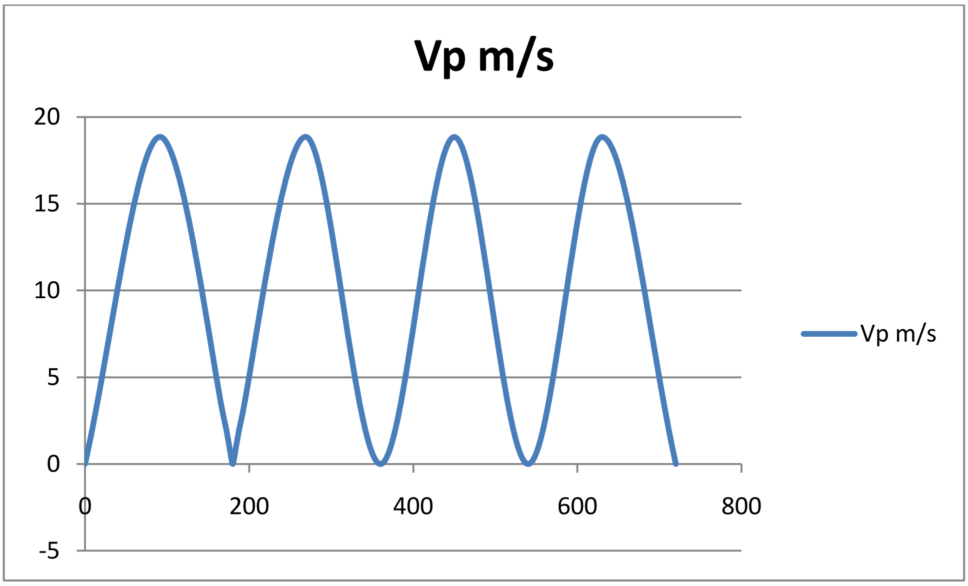
φ	R	R	L_{cr}	λ	ω	V_p	J_p	S_p
deg	mm	m	m	m	s^{-1}	m/s	m/s^2	m
0	30	0,03	0,12	0,25	628,3185307	0	14804,4066	0
90	30	0,03	0,12	0,25	628,3185307	18,84955592	2960,88132	0,03375
170	30	0,03	0,12	0,25	628,3185307	2,467325058	8881,277192	0,059657309
180	30	0,03	0,12	0,25	628,3185307	1,73201E-15	8882,643961	0,06
190	30	0,03	0,12	0,25	628,3185307	2,467325058	8881,277192	0,059657309
270	30	0,03	0,12	0,25	628,3185307	18,84955592	2960,88132	0,03375
360	30	0,03	0,12	0,25	628,3185307	5,77338E-15	14804,4066	0
450	30	0,03	0,12	0,25	628,3185307	18,84955592	2960,88132	0,03375
540	30	0,03	0,12	0,25	628,3185307	5,19604E-15	8882,643961	0,06
630	30	0,03	0,12	0,25	628,3185307	18,84955592	2960,88132	0,03375
720	30	0,03	0,12	0,25	628,3185307	1,15468E-14	14804,4066	0

Table N.1

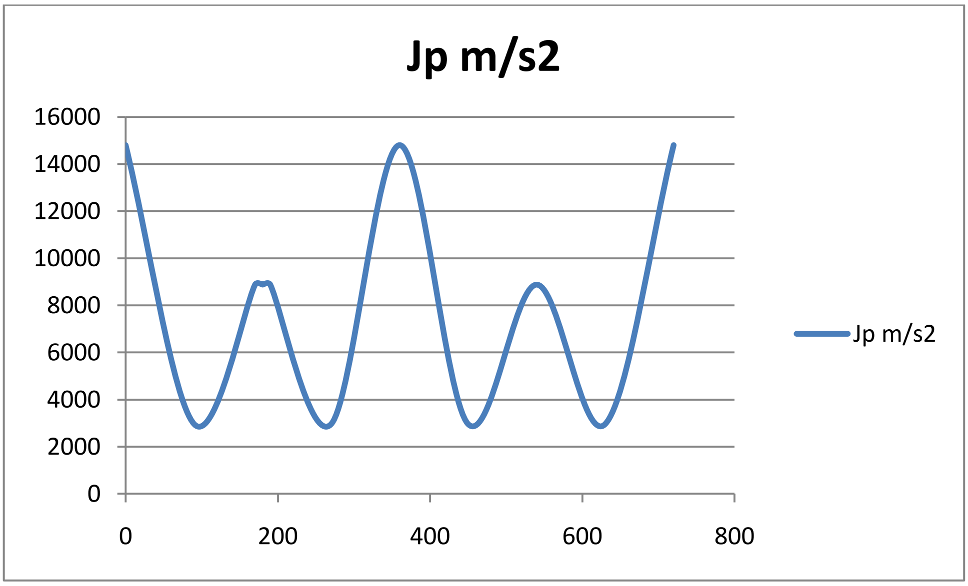


Graph N.1

The following tables show the behavior of the linear velocity of the master-rod and its acceleration.



Graph N.2



Graph N.3

4. Area of the piston head:

$$F_p = \frac{\pi D^2}{4} = \frac{3{,}14.(70.10^{-3})^2}{4} = 3846{,}5.10^{-6}$$

So: F_p=0,003846m^2

5. Different forces acting on the master-rod:

- Gas Forces, P_g, N - Analytical calculation of the gas forces as a function of the angle of rotation of the crankshaft φ

  Is done according to the next formula:

$$P_g = \left[p_b (\frac{S_h + S_c}{S_c + S_x})^n - p_{opp.} \right] .F_p$$

$$B = (\frac{S_h + S_c}{S_c + S_x})^n$$

S_h- Working stroke;

S_c- The stroke according to the height of the combustion chamber

$P_{opp.}$=0,1MPa- The pressure acting on the opposite side of the piston. It is equal to this in 4-stroke engines.

$P_{b.}$- That is the pressure in the beginning

$$\varphi = 0 \div 180^0 \rightarrow p_{b.} = p_a = 0{,}13MPa$$

$$\varphi = 180 \div 360^0 \rightarrow p_{b.} = p_a = 0{,}13MPa$$

$$\varphi = 360 \div 540^0 \rightarrow p_{b.} = p_j = 0{,}611MPa$$

$$\varphi = 540 \div 720^0 \rightarrow p_{b.} = p_r = 0{,}126 MPa$$

n-indicator that is changing in the following borders:

$$\varphi = 0 \div 180^0 \rightarrow n = 0;$$

$$\varphi = 180 \div 360^0 \rightarrow n = n_1 = 1{,}375$$

$$\varphi = 360 \div 540^0 \rightarrow n = n_2 = 1{,}25$$

$$\varphi = 540 \div 720^0 \rightarrow n = 0$$

- Inertia Forces:

- Inertia Forces of the objects with linear motions:

$$P_j = -m_j.\omega^2.R.\left[\frac{\cos(\varphi+\beta)}{\cos\beta} + \lambda.\frac{\cos^2\varphi}{\cos^2\beta}\right].10^{-6}, MN$$

- Inertia Forces of the objects with radial motions:

$$P_R = 10^{-6}.m_R.\omega^2.R = const, MN$$

*The masses m_j and m_R we define as follow:

$$m_j = m_{p.gr} + m_{mr.gr}$$

$$m_R = m_{cr.} + m'_{mr.gr}$$

Where:

m_{cr}. - is the mass of the crank.

$m'_{mr.gr}$ - Is the mass of the part of the master- rod that is brought to the axis of the crank.

$m_{p.gr}$. - is the mass of the piston group.

$m_{mr.gr}$ - is the mass of the part of the master-rod that is brought to the axis of the piston bolt.

- Other Forces and Moments acting on the Crankshaft mechanism- resulting of the forces of the gas, the inertia forces and the centrifugal forces the crankshaft mechanism is loaded with forces that could be calculated analytically

$N = P_{\Sigma}.\tan\beta$ - outside forces, MN

$S = P_{\Sigma}.\dfrac{1}{\cos\beta}$ - the force acting on the axis of the master-rod

$T = P_{\Sigma}.\dfrac{\sin(\varphi+\beta)}{\cos\beta}$ - tangential force

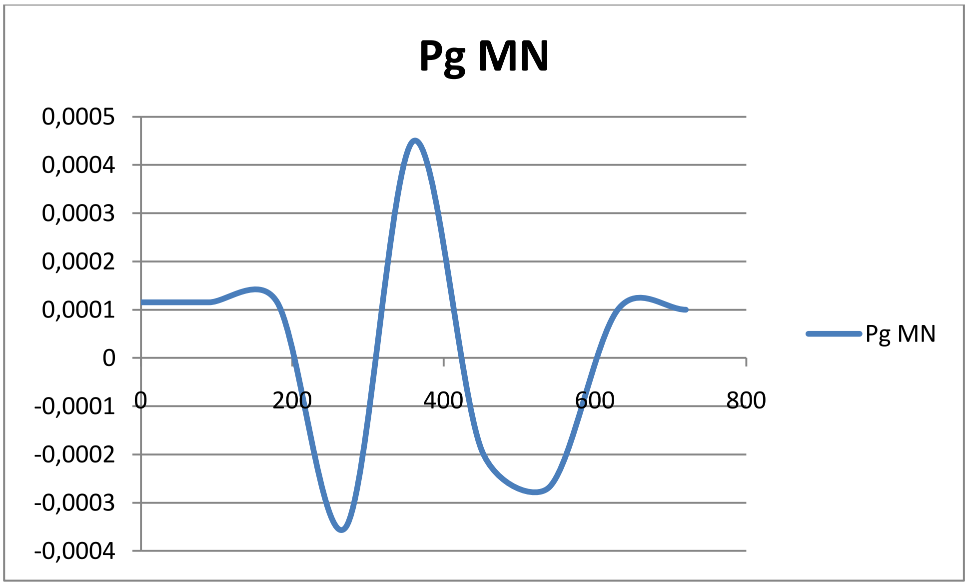
$Z = P_{\Sigma}.\dfrac{\cos(\varphi+\beta)}{\cos\beta}$ - normal force

Where:

$$P_{\Sigma} = P_g + P_j$$

φ	β	P_h	B	$P_{op.}$	F_p	P_g
deg.	**deg.**	**Mpa**		**Mpa**	m^2	**MN**
0	0	0,13	1	0,1	0,003846	0,00011538
90	14,45	0,13	1	0,1	0,003846	0,00011538
180	0	0,13	1	0,1	0,003846	0,00011538
270	14,45	0,13	0,0652	0,1	0,003846	-0,000352
360	0	0,13	1,6686	0,1	0,003846	0,000449667
450	14,45	0,611	0,0826	0,1	0,003846	-0,0001905
540	0	0,611	0,05	0,1	0,003846	-0,0002671
630	14,45	0,126	1	0,1	0,003846	0,000099996
720	0	0,126	1	0,1	0,003846	0,000099996

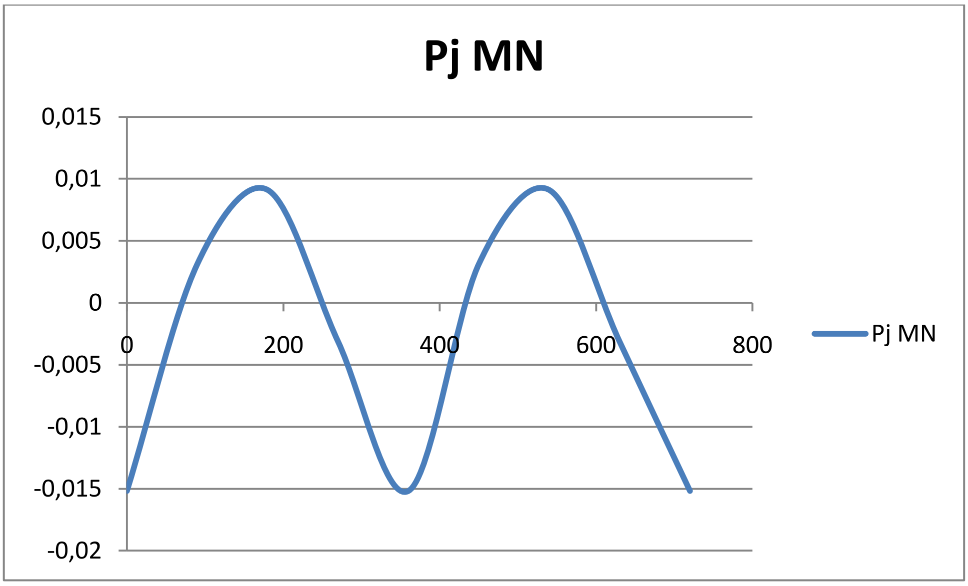
Table N.2



Graph N.4

φ	β	λ	m_j	R	cos(φ+β)	$cos^2φ$	ω	P_j
deg.	deg.		kg	m	cosβ	$cos^2β$	s^{-1}	MN
0	0	0,25	1,025	0,03	1	1	628,32	-0,015174588
90	14,45	0,25	1,025	0,03	-0,2576868	4,00165E-33	628,32	0,003128232
180	0	0,25	1,025	0,03	-1	1	628,32	0,009104753
270	14,45	0,25	1,025	0,03	0,25768676	3,60148E-32	628,32	-0,003128232
360	0	0,25	1,025	0,03	1	1	628,32	-0,015174588
450	14,45	0,25	1,025	0,03	-0,2576868	1,00041E-31	628,32	0,003128232
540	0	0,25	1,025	0,03	-1	1	628,32	0,009104753
630	14,45	0,25	1,025	0,03	0,25768676	1,96081E-31	628,32	-0,003128232
720	0	0,25	1,025	0,03	1	1	628,32	-0,015174588

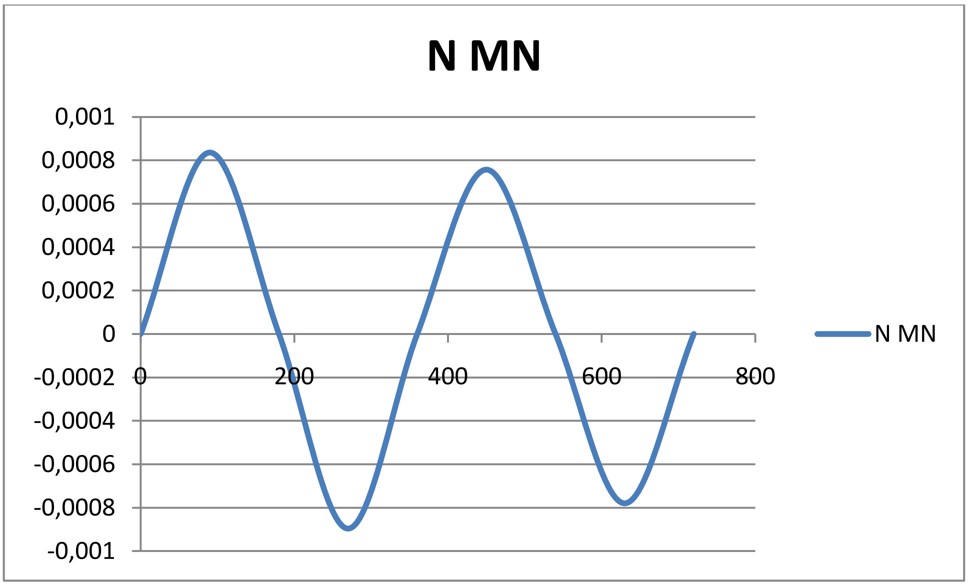
Table N.3



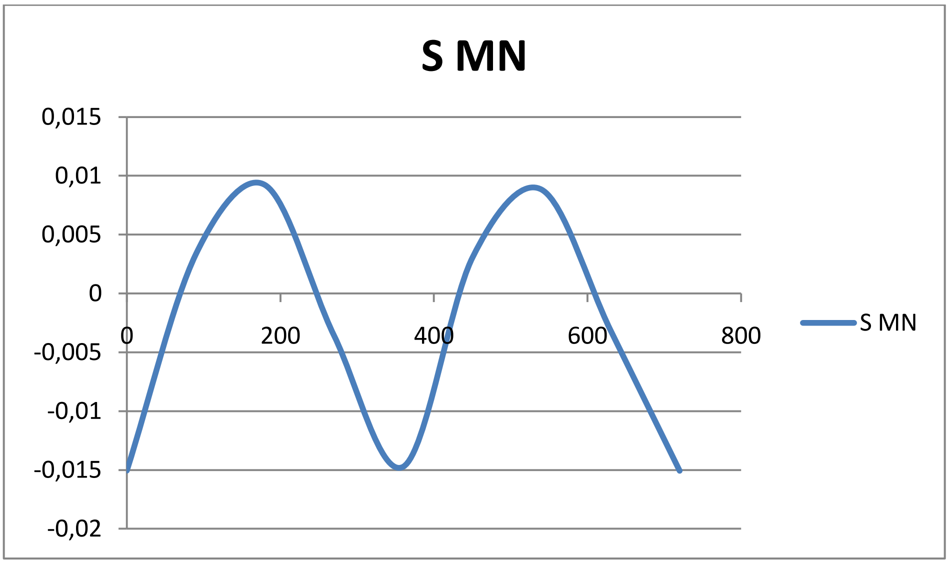
Graph N.5

φ	β	cos(φ+β)	sin(φ+β)	Pg	Pj	P_{Σ}	N	S	T	Z
deg.	deg.	cosβ	cosβ	MN	MN	MN	MN	MN	N	MN
0	0	1	0	0,00011538	-0,01517459	-0,01505921	0	-0,0150592	0	-0,01505921
90	14,45	-0,2576868	1	0,00011538	0,003128232	0,003243612	0,00083584	0,00334957	3243,612	-0,00083584
180	0	-1	1,2251E-16	0,00011538	0,009104753	0,009220133	0	0,00922013	1,1296E-12	-0,00922013
270	14,45	0,25768676	-1	-0,000352001	-0,00312823	-0,00348023	-0,0008968	-0,0035939	3480,234	-0,00089681
360	0	1	-2,45E-16	0,000449667	-0,01517459	-0,01472492	0	-0,0147249	3,60804E-12	-0,01472492
450	14,45	-0,2576868	1	-0,000190498	0,003128232	0,002937735	0,00075702	0,0030337	2937,735	-0,00075702
540	0	-1	3,6754E-16	-0,000267105	0,009104753	0,008837648	0	0,00883765	3,24823E-12	-0,00883765
630	14,45	0,25768676	-1	0,000099996	-0,00312823	-0,00302824	-0,0007803	-0,0031272	3028,236	-0,00078034
720	0	1	-4,901E-16	0,000099996	-0,01517459	-0,01507459	0	-0,0150746	7,38745E-12	-0,01507459

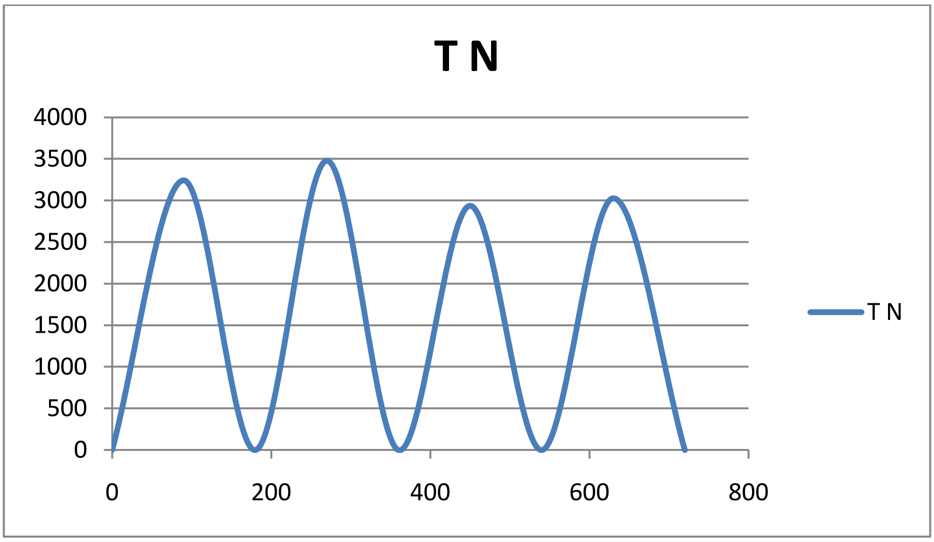
Table N.4



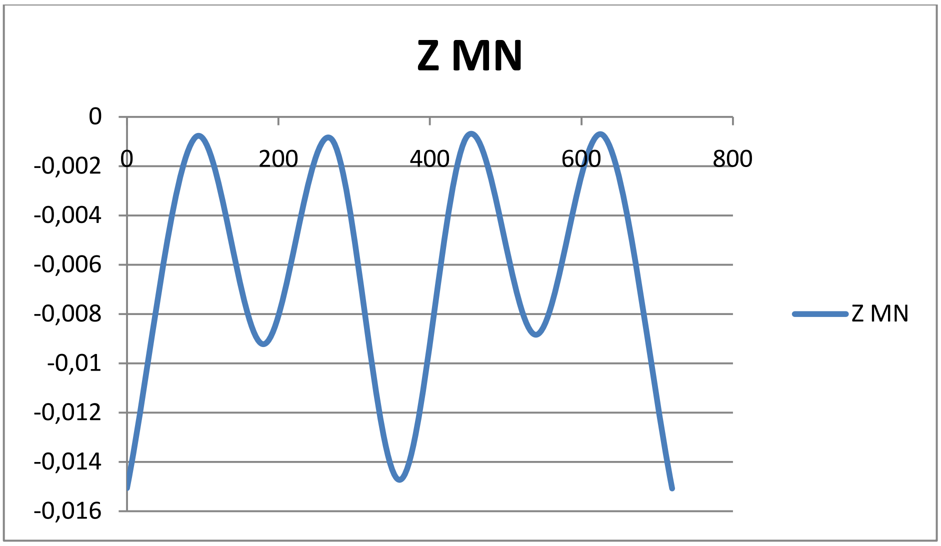
Graph N.6



Graph N.7



Graph N.8



Graph N.9

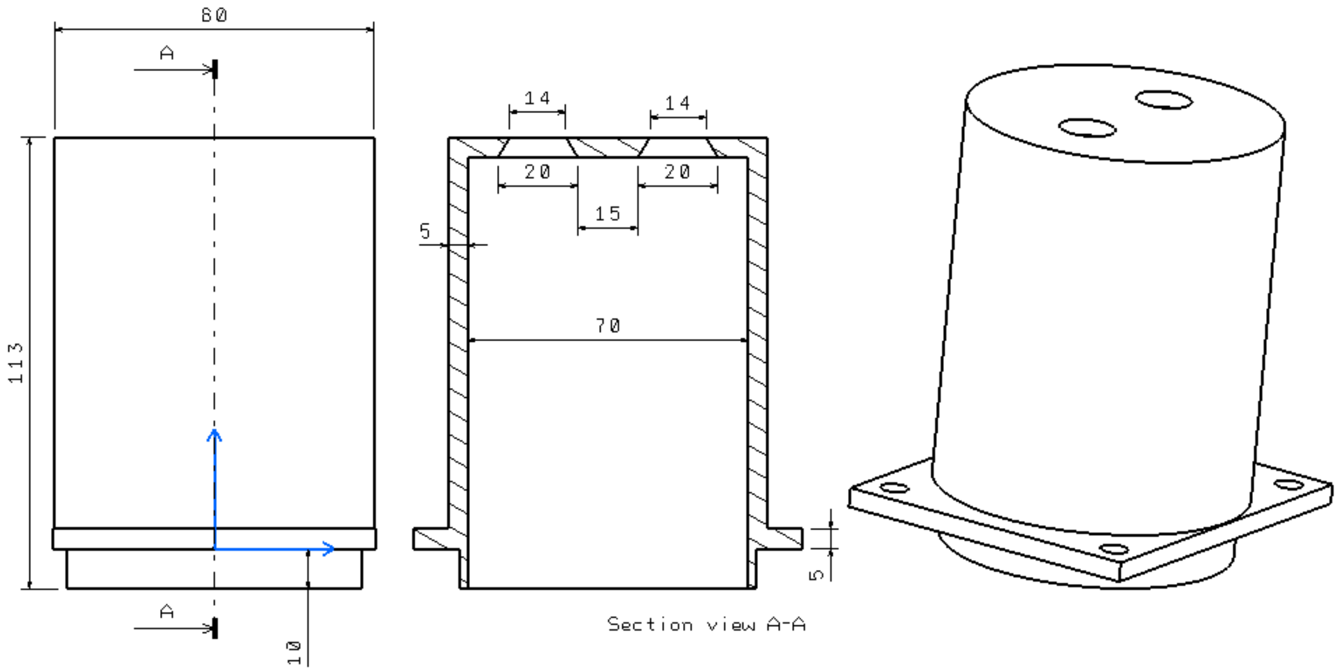
- Calculation of the outer diameter of the flywheel- it is determined by the following formula:

$$D_m = \frac{60}{\pi . n} . V_{per.} = \frac{60}{3{,}14.6000} .75 = 0{,}239m$$

Where: $V_{per.}$- the peripheral velocity which is in the following boundaries($V_{per.}$=(50÷100)m/s).

II. ***Strength calculations of some of the major parts of the engine***

1. Cylinders



Material: Stainless Steel 316L

Quantity: 3 cylinders

Tensile stress in the formation of the cylinder is determined by the formula for calculation of cylindrical pressure vessels, without taking into account variability of distribution of stress in the wall thickness, namely:

$$\sigma_{tz} = 0,5.p_z \frac{D}{\delta} \leq \sigma_{all.t}$$

Where: p_z=8,839 MPa – maximum pressure of the gases

δ=0,005m – wall thickness

D= 0,070m – cylinder diameter

$\sigma_{all.t}$=60 MPa

$$\sigma_{tz} = 0,5.8,839\frac{0,070}{0,005} = 60,374MPa$$

Under the effect of uneven heating of outer and inner surfaces of cylinder sleeves emerge temperature stress, which can be calculated by the formula:

$$\sigma_t = \frac{\alpha.E.\Delta T}{(1-\mu).2}, MPa$$

Where: α=11.10^{-5} K^{-1} – coefficient of linear expansion of cast iron

E=1,0.10^{5}Mpa- modulus of linear deformation of cast iron

ΔT=100K – temperature difference between outside and inside wall of cylinder sleeves

μ=0,25 – Poison`s coefficient

$$\sigma_t = \frac{11.10^{-6}.1{,}0.10^{5}.100}{(1-0{,}27).2} = 75{,}34MPa$$

Aggregate stress caused by the thermal load and the gas pressure will be equal to:

$$\sigma_{\Sigma} = \sigma_{tz} + \sigma_t = 34,374 + 75,34 = 109,71$$

2. Piston

When designing the piston we use constructive parameters of already existing engines. The elements of the piston are calculated without determining the variable way of loading. This reflects on the allowable stress.

Material: Aluminum ALLOY 4032-T6

Quantity: 3 Pistons

Main constructive dimensions of the piston:

Thickness of the piston crown “δ”- (0,05÷0,10).D δ=7mm

Height of the piston “H” – (0,8÷1,3).D H=70mm

Way between the crown and the axis

of the piston bolt: ”h_1” – (0,45÷0,47).D h_1=33mm

Diameter of the thickening of the

piston bolt: “d” – (0,3÷0,5).D d=28mm

Dimension: “b” – (0,3÷0,5).D b=22mm

Thickness of the walls of the

leading part: “δ_b“– (1,5÷4,5)mm δ_b=4mm

Thickness of the sealed part: “s” – (0,05÷0,10) s=6mm

Distance between the crown and the first

pitch: “e” – (0,06÷0,12).D e=4mm

Thickness of the walls between

the pitches: “h_n” – (0,03÷0,05).D h_n=3mm

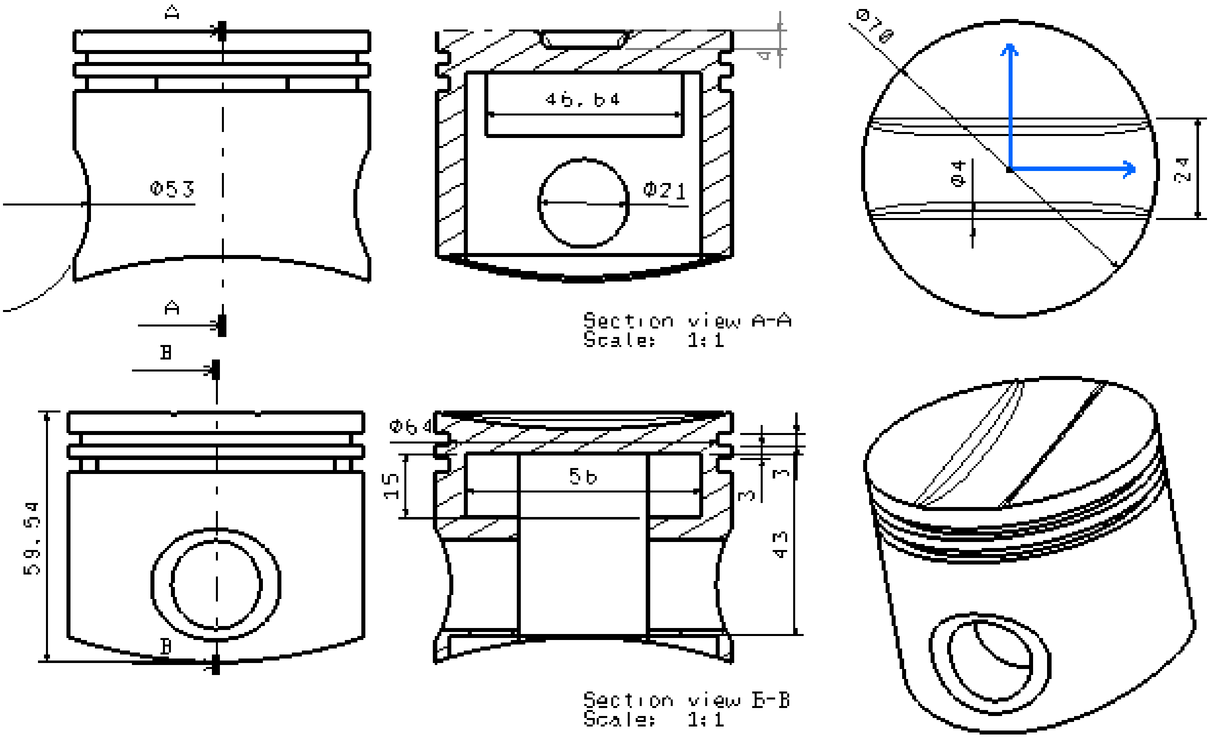
Number of the oil holes: “n_m” – (6÷12) n_m=8

Diameter of thee oil holes: “d_m” – (0,03÷0,05).d_β d_m=1mm

Diameter of the hole of the

piston bolt: “d_β” – (0,22÷0,28).D d_β=21mm

Height of the leading part: “h_b” – (0,6÷0,8).D h_b=42mm



Piston crown is calculated as the bending of circular plate resting on the cylinder and loaded with uniformly distributed load of gas pressure. The bending stress is defined under the formula:

$$\sigma_b = \frac{M_b}{W_b} = p_{z,\max} \cdot \left(\frac{r_i}{\delta} \right)^2 \leq \sigma_b^{all}, MPa$$

Where: $$r_i = \left[\frac{D}{2} - (s + t + \Delta t) \right]$$

$t = 0{,}005m$ - Radial thickness of the piston rings

Δt=0,0008m – Radial clearance between the piston ring and channels

$$r_i = \left[\frac{D}{2} - (s + t + \Delta t) \right] = \left[\frac{0,070}{2} - (0,006 + 0,005 + 0,0008) \right] = 0,0232m$$

$P_{z,max}$=8,839 MPa – maximum burning gases pressure in the cylinder

$$M_b = \frac{1}{3}.p_{z,\max}.r_i^3, MN.m$$ - Bending moment

$$W_b = \frac{1}{3}.r_i.\delta^2, m^3$$ - Moment of resistance

$$\sigma_b = p_{z,\max}.\left(\frac{r_i}{\delta}\right)^2 = 8,839.\left(\frac{0,0232}{0,007}\right)^2 = 97,1MPa < \sigma_b^{all} = 150MPa$$

The leading part of the piston in section x-x is weakened due to the outlet of the oil, so we should check the tension and compression.

The bending pressure is defined by the next formula:

$$\sigma_{b.pr.} = \frac{P_{z,\max}}{F_{x-x}} \le \sigma_{b.pr.}^{all}, MPa$$

Where $P_{z,max}$=$p_{z,max}$.F_b=8,839. 0,003846=0,033994 MN – maximum gas force acting on the crown of the piston

$$F_{x-x} = \frac{\pi}{4}.(d_k^2 - d_i^2) - n_м.F', m^2$$ - the area of the section x-x

$$d_k = D - 2.(t + \Delta t) = 0,070 - 2.(0,005 + 0,0008) = 0,0584m$$ - the internal diameter of the pitch for the oil

$$d_i = 2.r_i = 2.0,0232 = 0,0464m$$ - the internal diameter of the piston crown

$$F' = \frac{d_k - d_i}{2}.d_м = \frac{0,0584 - 0,0464}{2}.0,001 = 0,000006m^2$$

$$F_{x-x}=\frac{\pi}{4}.\left(d_k^2-d_i^2\right)-n_м.F^{'}=\frac{\pi}{4}.\left(0,0584^2-0,0464^2\right)-8.0,000006=0,0009411m^2$$

$$\sigma_b=\frac{P_{z,\max}}{F_{x-x}}=\frac{0,033994}{0,0009411}=36,125MPa<\sigma_b^{all}=40MPa$$

Tension strength in the cross-section x-x is defined by the next formula:

$$\sigma_{str}=\frac{P_{j,\max}}{F_{x-x}}\leq\sigma_t^{all},MPa$$

Where: $P_{j,\max}=10^{-6}.m_{x-x}.R.\omega_{nx,\max}^2.(1+\lambda),MN$ - inertia force of reciprocating motion of the mass of the piston over the section x-x set at a maximum speed of engine idling.

$n_{nx,\max}=(1,05\div1,35).n,\min^{-1}$ - Maximum speed of the crankshaft at idle.

We assume: $n_{nx,\max}=1,1.6000=6600\min^{-1}$

$$\omega_{nx,\max}=\frac{\pi.n_{nx,\max}}{30}=\frac{\pi.6600}{30}=690,8rad^{-1}$$

$\sigma_{str}^{all}=10MPa$ - Allowable tensile stress;

$m_{x-x}=\left(0,4\div0,6\right).m_{p,гр},kg$ -Mass of the piston group above section x-x

$m_{p.gr}=0,800kg$ - Mass of the piston group.

$$m_{x-x}=0,5.0,800=0,400kg$$

$$P_{j,\max} = 10^{-6}.m_{x-x}.R.\omega_{nx,\max}^2.(1+\lambda) = 10^{-6}.0,400.0,030.690,8^2.(1+0,250) = 0,007158MN$$

$$\sigma_{str} = \frac{P_{j,\max}}{F_{x-x}} = \frac{0,007518}{0,000941} = 7,607MPa < \sigma_{str}^{all} = 10MPa$$

The leading part of the piston is being tested under the maximum specific pressure by the formula :

$$q_{в} = \frac{N_{\max}}{h_b.D}, MPa$$

$N_{\max} = 0,008968MN$ - maximum normal force.

For already existing engines q_b is in the following boundaries: (0,3÷1,0)MPa

$$q_{в} = \frac{N_{\max}}{h_b.D} = \frac{0,0008968}{0,042.0,070} = 0,31MPa$$

Bending stress:

$$\sigma_{ben.} = 0,0045.p_{z,\max}.\left(\frac{D}{h_n}\right)^2 = 0,0045.8,839.\left(\frac{0,070}{0,003}\right)^2 = 21,6554MPa$$

Cutting stress:

$$\tau_{cut.} = 0,0314.p_{z,\max}.\frac{D}{h_n} = 0,0314.8,839.\frac{0,070}{0,003} = 6,476MPa$$

Sum of the stresses according to the third strength theory:

$$\sigma_{\Sigma} = \sqrt{\sigma_{ben}^2 + 4.\tau_{cut.}^2} = \sqrt{21,6554^2 + 4.6,476^2} = 25,233MPa < \sigma_{\Sigma}^{all} = 30MPa$$

To avoid jamming of the piston in the cylinder when the engine is working and it is heated, we must determine the diameter of the sealing part and the diameter of the leading part of the piston, hence the lateral clearances of the sealing and the leading part of the piston in the cold, when we know the necessary diametric clearances when it is hot.

For already existing engines the allowable clearances are in the following boundaries:

-for the sealing part: $\Delta_y^{'} = (0{,}002 \div 0{,}003).D, mm$

And we accept to be - $\Delta_y^{'} = 0{,}0025.86 = 0{,}215 mm$

-for leading part: $\Delta_в^{'} = (0{,}0005 \div 0{,}0015).D, mm$

And we accept it to be - $\Delta_в^{'} = 0{,}0015.86 = 0{,}129 mm$

Then we have:

$$D_s = \frac{D.\left[1 + \alpha_ц.\left(T_ц - T_0\right)\right] - \Delta_y^{'}}{1 - \alpha_б.\left(T_y - T_0\right)}, m$$

$$D_L = \frac{D.\left[1 + \alpha_ц.\left(T_ц - T_0\right)\right] - \Delta_в^{'}}{1 + \alpha_б.\left(T_B - T_0\right)}, m$$

Where: $\alpha_ц = 11.10^{-6} K^{-1}$ - coefficient of linear expansion of the material of the cylinder.

$\alpha_б = 11.10^{-6} K^{-1}$ - coefficient of linear expansion of the material of the piston.

$T_ц = 388K$ - temperature of the cylinder

$T_y = 593K$ - temperature of the sealing part.

$T_в = 473K$ - temperature of the leading part of the piston.

$T_0 = 293K$ - temperature of the engine when it is cold.

$$D_s = \frac{D.\left[1+\alpha_ц.\left(T_ц - T_0\right)\right] - \Delta_y^{'}}{1+\alpha_б.\left(T_y - T_0\right)} = \frac{0,070.\left[1+11.10^{-6}.\left(388-293\right)\right] - 0,215.10^{-3}}{1+11.10^{-6}.\left(593-293\right)} = 0,06763m$$

$$D_l = \frac{D.\left[1+\alpha_ц.\left(T_ц - T_0\right)\right] - \Delta_в^{'}}{1+\alpha_б.\left(T_в - T_0\right)} = \frac{0,070.\left[1+11.10^{-6}.\left(388-293\right)\right] - 0,129.10^{-3}}{1+11.10^{-6}.\left(473-293\right)} = 0,0698m$$

The diametrical clearances in the sealing and leading parts of the piston when the engine is will be as follow:

$$\Delta_s = D - D_s = 70 - 67,63 = 2,37mm$$

$$\Delta_l = D - D_l = 70 - 69,8 = 0,2mm$$

3. Piston Bolt.

Piston bolts are made of a Precision shaft and two little rivets that are connected to the shaft at the ends.

Precision shaft

Material: Steel 12L14

Tolerance: +0,0 mm to -0,0005 mm

Quantity: 3

Rivet

Material: Aluminum ALLOY 2117-T4 type Solid

Quantity: 6

Main Constructive elements:

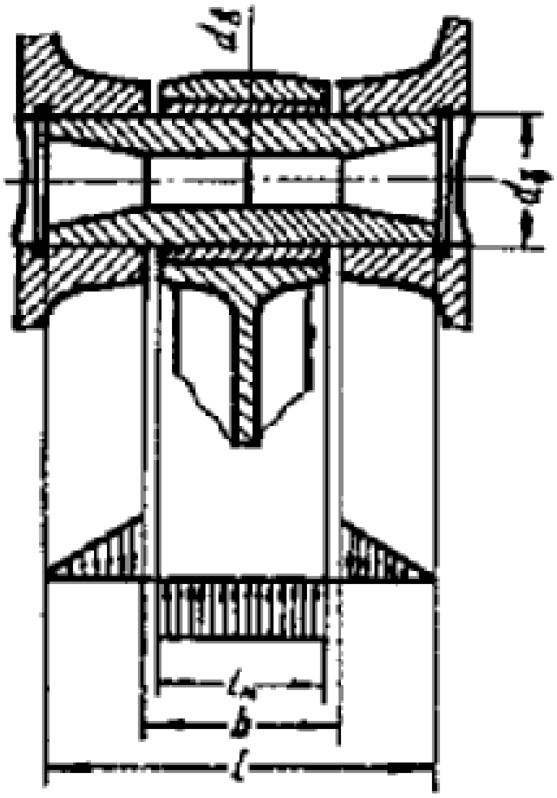
Outer diameter of the bolt: “d_o” – (0,22÷0,28).D d_o=21mm

Inner diameter of the bolt: “d_i” – (0,65÷0,75). d_o d_i=13mm

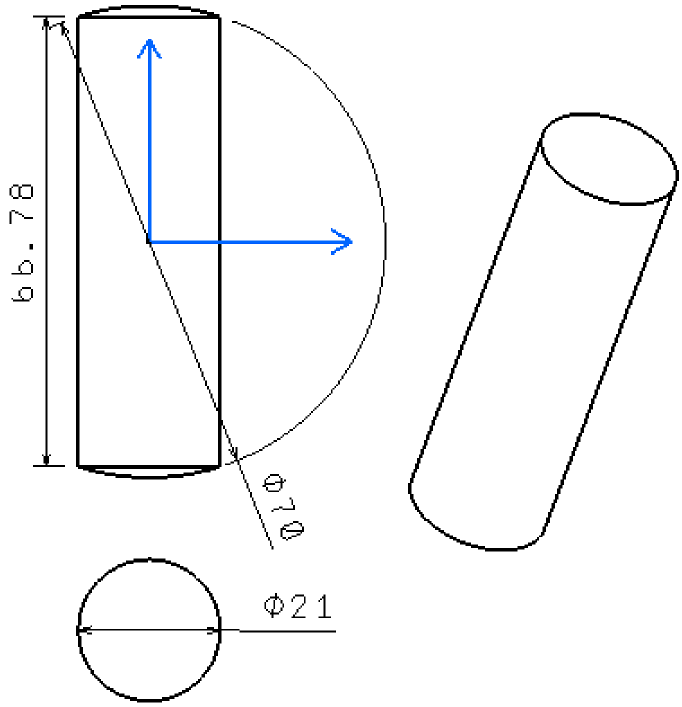
Length of the bolt: “l” – (0,88÷0,93).D l=63mm

Length of the upper part of the piston rod:

”l_m” – (0,28÷0,32).D l_m=22mm



(Scheme of the forces acting on the piston bolt)



Drawing of the Piston Bolt

The piston bolt is subjected to varying in size and direction load, causing surface tension pressure, bending and shear. We accept steel 18 XH3H as a making material.

Constructive dimensions about the piston bolt could be defined by the data we have from already existing engines.

The piston bolt is being calculated in the upper part of the piston rod “q_r“ and In the holes of the piston “q_p”.

$$q_r = \frac{P_{z,\max} + P_{j,\max}}{d_o.l_{м}} < q_r^{all}, MPa$$

$$q_p = \frac{P_{z,\max} + K.P_{j,\max}}{d_o.(l-b)} < q_r^{all}, MPa$$

Where:

$P_{z,max}$=0,033994, MN – maximum gas force acting on the crown of the piston

$P_{j,max}$=0,007158, MN – maximum inertia force of the piston group

b = 22mm – distance between the holes of the piston bolt

K=(0,78÷0,86) – coefficient that gives the mass of the piston bolt

$q_r^{all} = 60MPa$ and $q_p^{all} = 50MPa$ are the allowable values of q_p and q_r

$$q_r = \frac{0,033994 - 0,007158}{0,021.0,022} = 58,07 MPa < q_r^{all} = 60 MPa$$

$$q_p = \frac{0,033994 - 0,78.0,007158}{0,021.(0,063 - 0,022)} = 32,99 MPa < q_p^{all} = 50 MPa$$

The maximum load is equal to:

$$P = P_{z,\max} + K.P_{j,\max} = 0,033994 - 0,78.(0,007158) = 0,0284 MN$$

This load causes stresses of tension, cutting. When making the calculations we consider the piston bolt as a beam.

Stress of bending:

$$\sigma_{ben.} = \frac{P.(l + 2.b - 1,5.l_M)}{1,2.d_p^3.(1 - \alpha^4)} < \sigma_{ben.}^{all}, MPa$$

Where: $\alpha = \frac{d_i}{d_o} = \frac{13}{21} = 0,62$ - ratio between the inner and outer diameter.

$\sigma_{ben.}^{all} = 500 MPa$ - allowable bending stress.

$$\sigma_{ben} = \frac{P.(l + 2.b - 1,5.l_м)}{1,2.d_o^3.(1 - \alpha^4)} = \frac{0,0284.(0,063 + 2.0,022 - 1,5.0,022)}{1,2.0,021^3.(1 - 0,62^4)} = 222 MPa < \sigma_{ben.}^{all}$$

Maximum tangential stresses of cutting act in the cross sections of the bolt, placed between the ears of the piston and the piston rod ears, and is calculated by the formula:

$$\tau_{cp} = \frac{0,85.\left(1+\alpha+\alpha^2\right).P}{d_o^2.\left(1-\alpha^4\right)} \le \tau_{cp}^{all}, MPa$$

Where: $\tau_{cp}^{all} = 250MPa$ - Is the allowable stress.

$$\tau_{cp} = \frac{0,85.\left(1+\alpha+\alpha^2\right).P}{d_o^2.\left(1-\alpha^4\right)} = \frac{0,85.\left(1+0,62+0,62^2\right).0,0284}{0,021^2.\left(1-0,62^4\right)} = 128,75MPa < \tau_{cp}^{all}$$

Loading of the piston bolt on the surface is uneven so we assume that it has a sinusoidal distribution law, so when the engine works the diameter of the bolt increases.

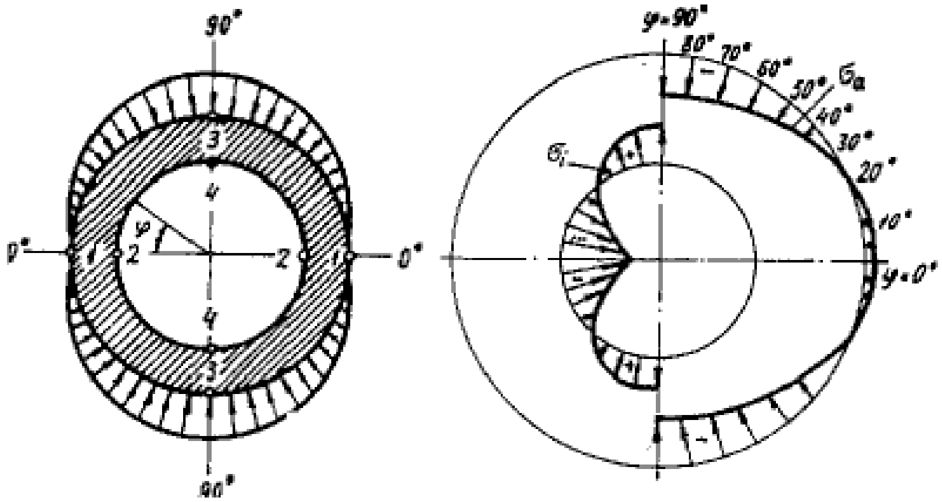
$$\Delta d_{o,\max} = \frac{0,09.P}{l.E}.\left(\frac{1+\alpha}{1-\alpha}\right)^3.k_1, m$$

Where: $k_1 = 1,5-15.\left(\alpha-0,4\right)^3 = 1,5-15.\left(0,62-0,4\right)^3 = 1,34028$

$E = 2,3.10^5\,MPa$ - model of deformation of the steel.

$$\Delta d_{o,\max} = \frac{0,09.P}{l.E}.\left(\frac{1+\alpha}{1-\alpha}\right)^3.k_1 = \frac{0,09.0,0284}{0,063.2,3.10^5}.\left(\frac{1+0,62}{1-0,62}\right)^3.1,34 = 0,000018m = 0,018mm$$

And $\Delta d_{o,\max} = \left(0,017 \div 0,05\right)mm.$



(Scheme of deformation of the piston bolt)

Under the deformation we have some stresses in the bolt which are maximal in the middle cross section of the bolt. Outer plane (point 1 and point 3), inner plane (point 2 and point 4).

When $\varphi = 0$:

Point 1:

$$\sigma_1 = \frac{P}{l.d_o}.\left[0{,}19.\frac{(2+\alpha).(1+\alpha)}{(1-\alpha)^2} - \frac{1}{1-\alpha}\right].k_1 =$$

$$= \frac{0{,}0284}{0{,}063.0{,}021}.\left[0{,}19.\frac{(2+0{,}62)(1+0{,}62)}{(1-0{,}62)^2} - \frac{1}{1-0{,}62}\right].1{,}34028 = 90{,}08MPa$$

Point 2:

$$\sigma_2 = -\frac{P}{l.d_o}.\left[0{,}19.\frac{(1+2.\alpha).(1+\alpha)}{(1-\alpha)^2.\alpha}+\frac{1}{1-\alpha}\right].k_1 =$$

$$= -\frac{0{,}0284}{0{,}063.0{,}021}.\left[0{,}19.\frac{(1+2.0{,}62)(1+0{,}62)}{(1-0{,}62)^2.0{,}62}+\frac{1}{1-0{,}56}\right].1{,}34028 = -286{,}97MPa$$

When $\varphi = 90$:

Point 3:

$$\sigma_3 = -\frac{P}{l.d_o}.\left[0{,}174.\frac{(2+\alpha).(1+\alpha)}{(1-\alpha)^2}+\frac{0{,}636}{1-\alpha}\right].k_1 =$$

$$= -\frac{0{,}0284}{0{,}063.0{,}021}.\left[0{,}174.\frac{(2+0{,}62)(1+0{,}62)}{(1-0{,}62)^2}+\frac{0{,}636}{1-0{,}62}\right].1{,}34028 = -195{,}20MPa$$

Point 4:

$$\sigma_4 = \frac{P}{l.d_o}.\left[0{,}174.\frac{(1+2\alpha).(1+\alpha)}{(1-\alpha)^2.\alpha}-\frac{0{,}636}{1-\alpha}\right].k_1 =$$

$$= \frac{0{,}0284}{0{,}063.0{,}021}.\left[0{,}174.\frac{(1+2.0{,}62)(1+0{,}62)}{(1-0{,}62)^2.0{,}62}-\frac{0{,}636}{1-0{,}62}\right].1{,}34028 = 154{,}81MPa$$

The biggest amount of stress is in point 2. The value of the stress should not be bigger than 350MPa.

The working clearance of the "piston bolt- piston" when it is hot should be:

$$\Delta L = 0,001.d_б = 0,001.21 = 0,021mm$$

The mounting clearance of the compound "piston bolt- piston" is different than the working one and is calculated in this formula:

$$\Delta_p = \Delta L + \left[\alpha_p \left(T_p - T_0\right) - \alpha_{pb}\left(T_{pb} - T_0\right)\right] d, mm$$

Where: $\alpha_{бб} = 11.10^{-6} K^{-1}$ - is coefficient of linear deformation of steel.

$\alpha_p = 25.10^{-6} K^{-1}$ - is coefficient of linear deformation of aluminum alloys.

$T_{pb} = T_p = 390K$ - the temperature of the piston bolt and the piston.

$T_0 = 293K$ - mounting temperature.

$$\Delta_б = 0,025 + [25.10^{-6}.(293-293) - 11.10^{-6}.(293-293)].25 = 0,021mm$$

For already existing engines it should be in the following boundaries:

$$0,2 \div 0,4mm$$

4. Piston Rings

Piston rings provide a tight cylinder space. They work at high temperature and variable loads. The main requirements about them is to have high elasticity, durability and low coefficient of friction with the cylinder walls. Materials that are used for this rings are alloy cast iron with chromium, nickel, copper titanium and other materials.

Main constructive elements could be used from data of already existing engines.

Radial thickness of sealing rigns: "t_s" – (0,039÷0,045).D t_y=3mm

Radial thickness of oil-collecting rings:"t_o" – (0,038÷0,043).D t_o=3mm

Height of the ring:"a" – (2÷4)mm a=3mm

Radial clearance of the sealing ring:"Δt_s" –(0,5÷0,95)mm Δt_s=0,8mm

Radial clearance of the oil-collecting

rings: "Δt_o" –(0,5÷0,95)mm Δt_o=0,8mm

Axial clearance of the ring:" Δa" –(0,04÷0,08)mm Δa=0,06mm

Clearance in the ring:"A_0" –(2,4÷4,0).t A_0=9mm

- Defining the middle radial pressure of the ring on the walls of the cylinder p_m:

$$p_m = 0{,}152.E.\frac{\frac{A_0}{t}}{\left(\frac{D}{t}-1\right)^3.\frac{D}{t}}, MPa$$

Where: $E = 1.10^5\,MPa$

$$p_m = 0{,}152.E.\frac{\frac{A_0}{t}}{\left(\frac{D}{t}-1\right)^3.\frac{D}{t}} = 0{,}152.1.10^5.\frac{\frac{0{,}0096}{0{,}004}}{\left(\frac{0{,}070}{0{,}004}-1\right)^3.\frac{0{,}070}{0{,}004}} = 0{,}386\,MPa$$

- Defining the stresses of bending if the ring

In working condition

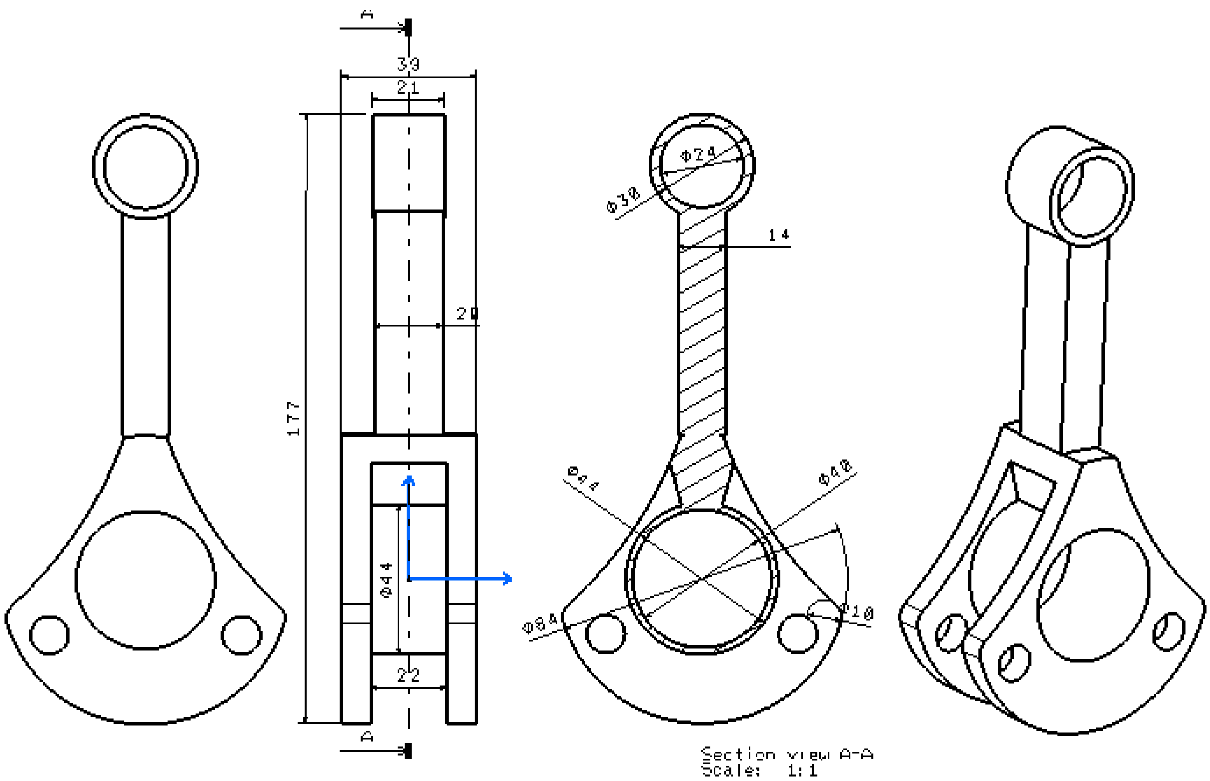
$$\sigma_{b,wc} = 2,61.p_m.\left(\frac{D}{t} - 1\right)^2 = 2,61.0,386.\left(\frac{0,070}{0,004} - 1\right)^2 = 274,24MPa$$

When mounting the rings on the pistons

$$\sigma_{b,m} = \frac{4.E.\left(1 - 0,114.\frac{A_0}{t}\right)}{m.\left(\frac{D}{t} - 1,4\right)^2} = \frac{4.1.10^5.\left(1 - 0,114.\frac{0,009}{0,004}\right)}{2.\left(\frac{0,070}{0,004} - 1,4\right)^2} = 323,6MPa$$

5. Master rod

When the engine is working the piston rod is under variable in size and direction gas forces and inertia forces. That is why it is made of high quality steel with high resistance of fatigue (40,45). The piston rod is contained by upper head, trunk, lower head. The dimensions of the rod could be determined over existing engines.



Drawing of the Master Rod

Material: Aluminum ALLOY 7075

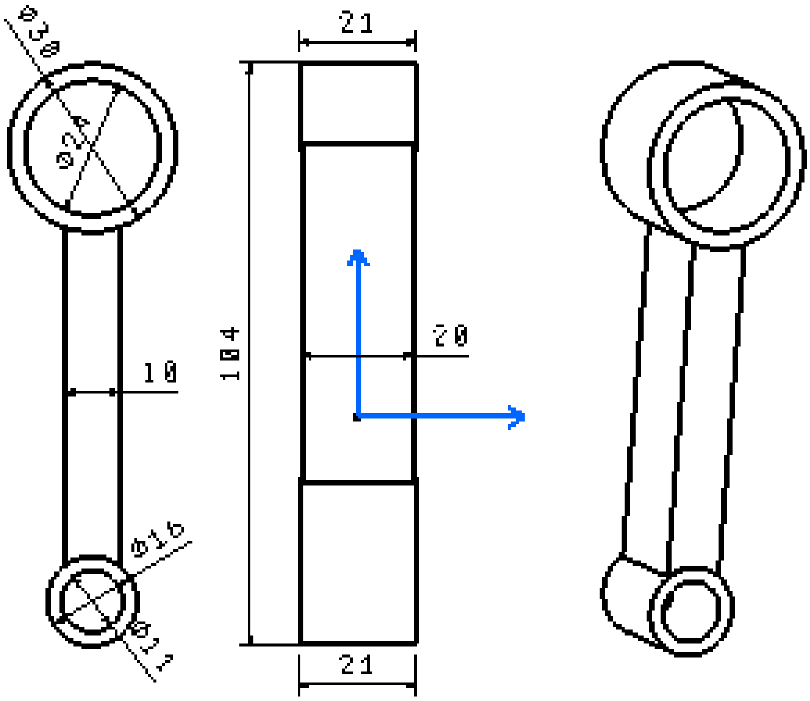
Quantity: 1

Main constructive elements:

Inside diameter of the upper head:"d" –($d=d_o$)	$d=21mm$
Outside diameter of the upper head:"d_g" –(1,25÷1,65) .d_o	$d_g=32mm$
Length of the upper head:"l_m" – (0,33÷0,45).D	$l_m=30mm$
Minimal radial thickness of the upper head: "h_g" – (0,16÷0,27).d	$h_g=4mm$
Minimum height of the profile: "$h_{m,min}$" –(0,5÷0,55).d_g	$h_{m,min}=17mm$
Dimensions: "h_m" – (1,2 ÷1,4).$h_{m,min}$	$h_m=20mm$
"$a_M \approx t_M$" -(2,5÷6)mm	$a_m=5mm$
Diameter of the lower head:"d_l" –(0,56÷0,75).D	$d_l=40mm$
Length of the lower head:"l_k" –(0,45÷0,95).d_l	$l_k=39mm$

6. Auxiliary Rod

The auxiliary rods are the connecting rods between the master rod and the other pistons of the radial engine.



Drawing of the Auxiliary Rod

Material: Aluminum ALLOY 7075

Quantity: 2

Main constructive elements:

Diameter of the upper hole of the Auxiliary Rod: d_u=24mm

Outside diameter of the upper hole of the Auxiliary Rod: $d_{u.o.}$=30mm

Diameter of the lower part of the Auxiliary Rod: d_l=11mm

Outside diameter of the lower part of the Auxiliary Rod: $d_{l.o.}$=16mm

Thickness of the "x-y" view of the Auxiliary Rod: Th_{x-y}=10mm

Thickness of the "x-z" view of the Auxiliary Rod: Th_{x-z}=20mm

Thickness of the upper hole of the Auxiliary Rod: Th_{u}=21mm

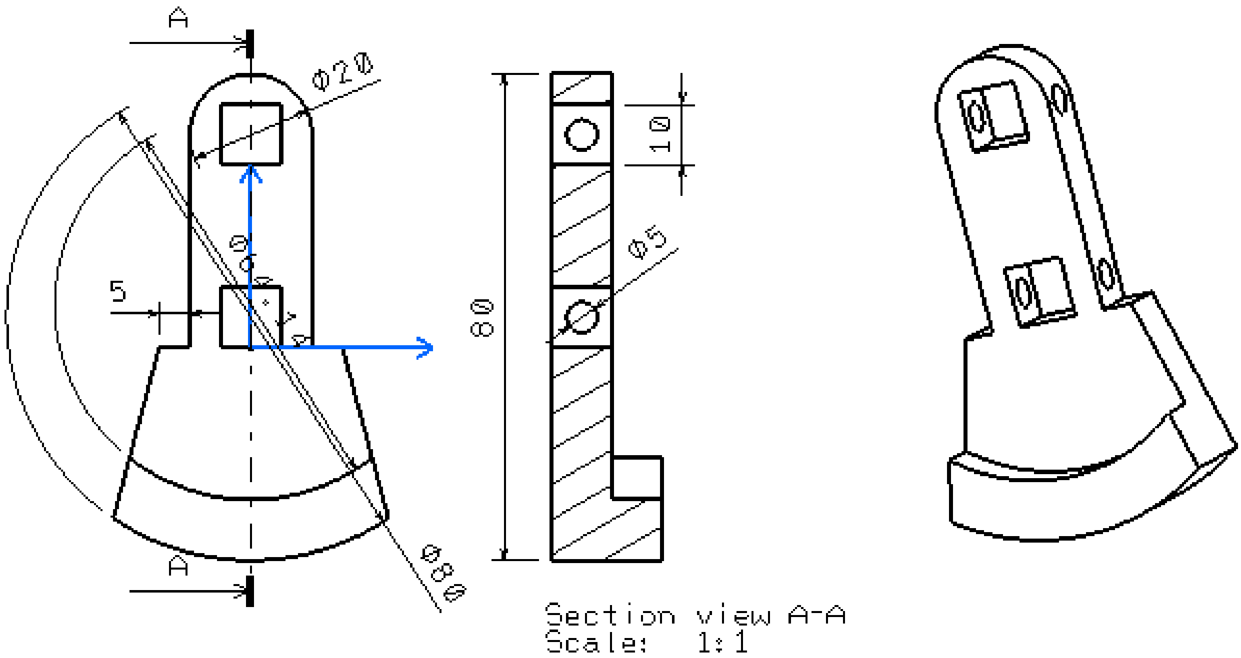
Thickness of the lower hole of the Auxiliary Rod: Th_{l}=21mm

Length of the Auxiliary Rod: $L_{a.r.}$=104mm

7. Crank-Shaft

The crank-shaft mechanism that I designed is constructed of two major part(front crank-shaft and rear crank-shaft), two crank cheeks, and one journal that connects the crank cheeks.

- Crank Cheeks



Drawing of the crank-cheeks

Material: Steel 416 Stainless

Quantity: 2

Main constructive elements:

Distance between the two centers

of the holes of the Crank Cheek: b=30mm

Diameter of the upper part of the Crank Cheek: $d_{u.cr}$=20mm

First diameter of the counter-weight: d^{1}_{cw}=64,14mm

Second diameter of the counter-weight: d^{2}_{cw}=80mm

Thickness between the upper part of the

Crank-Cheek and the lower part: f=5mm

Diameter of the two little holes: d_1=5mm

First side of the rectangular: a=10mm

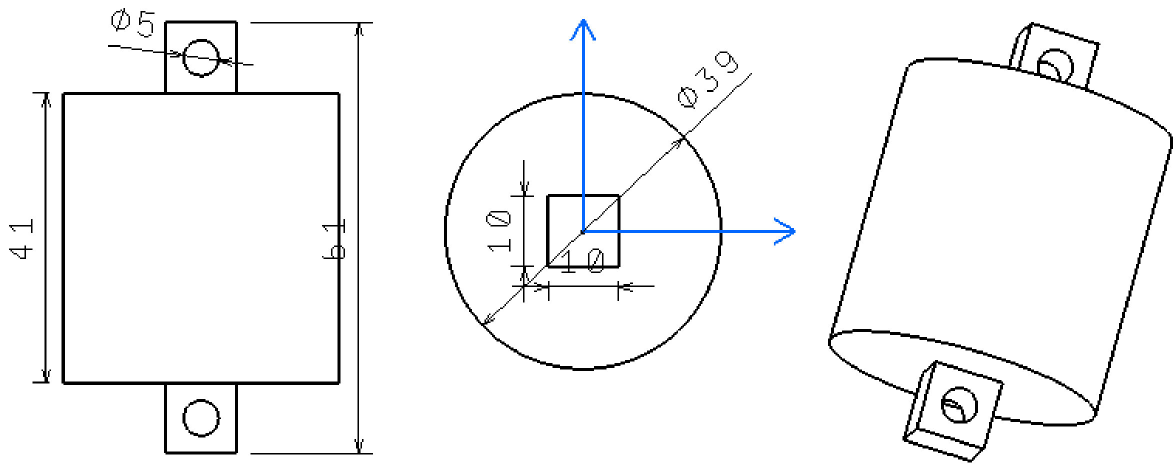
Second side of the rectangular: c=10mm

Length of the Crank-Cheek: L_{cr}=80mm

- Main Journal

Every Crank Shaft is made of the shafts and the different journals between them. The journals could be one more if we have a line engine (for example four-stroke, four cylinder in-line engine).

The whole mechanism should be fixed so we do not have any degree of freedom between the different parts. And as a whole it should rotate around an axis.



Drawing of the Main Journal

Material: Steel 4340 ALLOY

Quantity: 1

Main constructive elements:

Length of the Main Journal: L_{mj}=41mm

Length between the ends: L=61mm

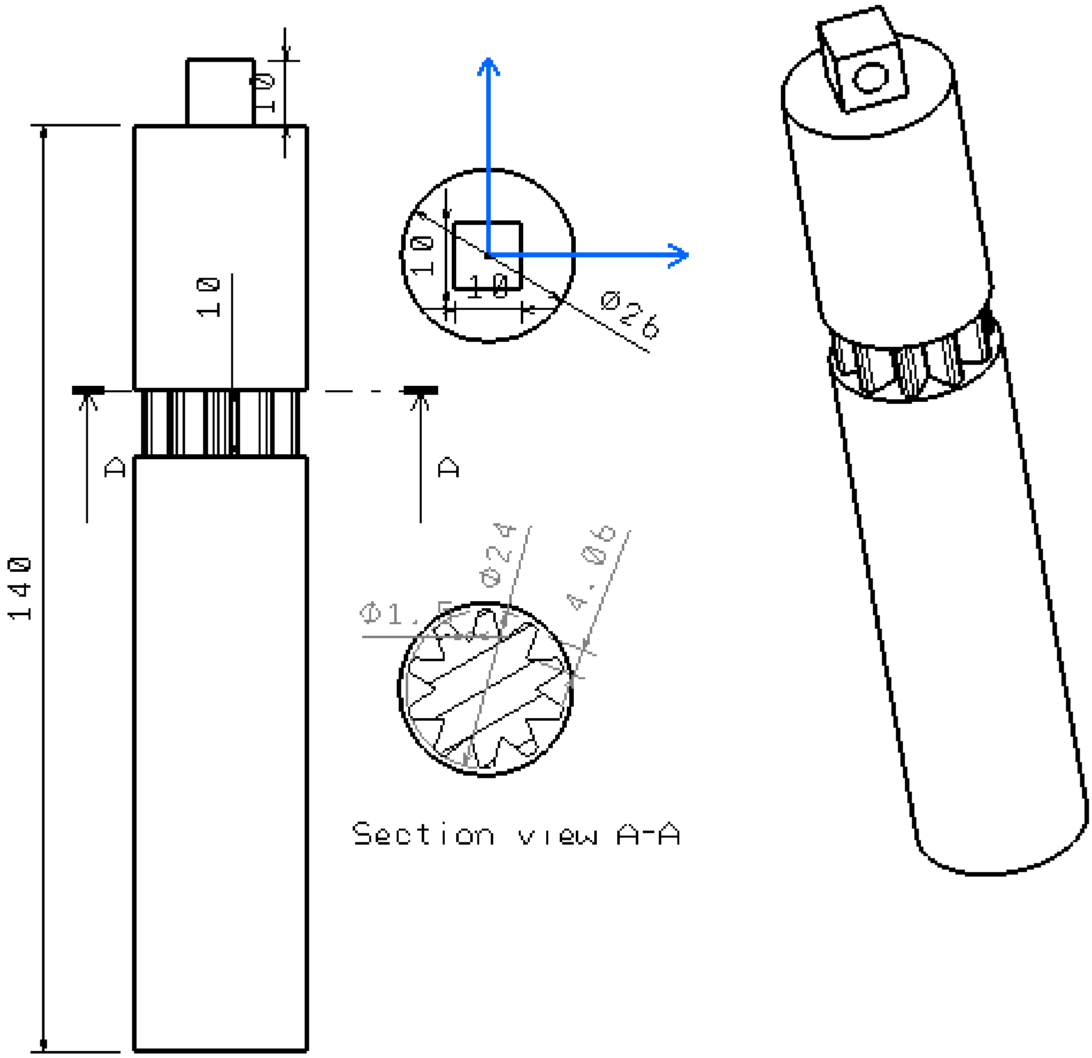
Diameter of the main journal: D_{mj}=39mm

Diameter of the holes: $d=d_1$=5mm

One side of the rectangular: a=10mm

Other side of the rectangular: c=10mm

- Crank Shaft (rear)



Material: Stainless Steel 4340 ALLOY

Quantity: 1

Main constructive elements:

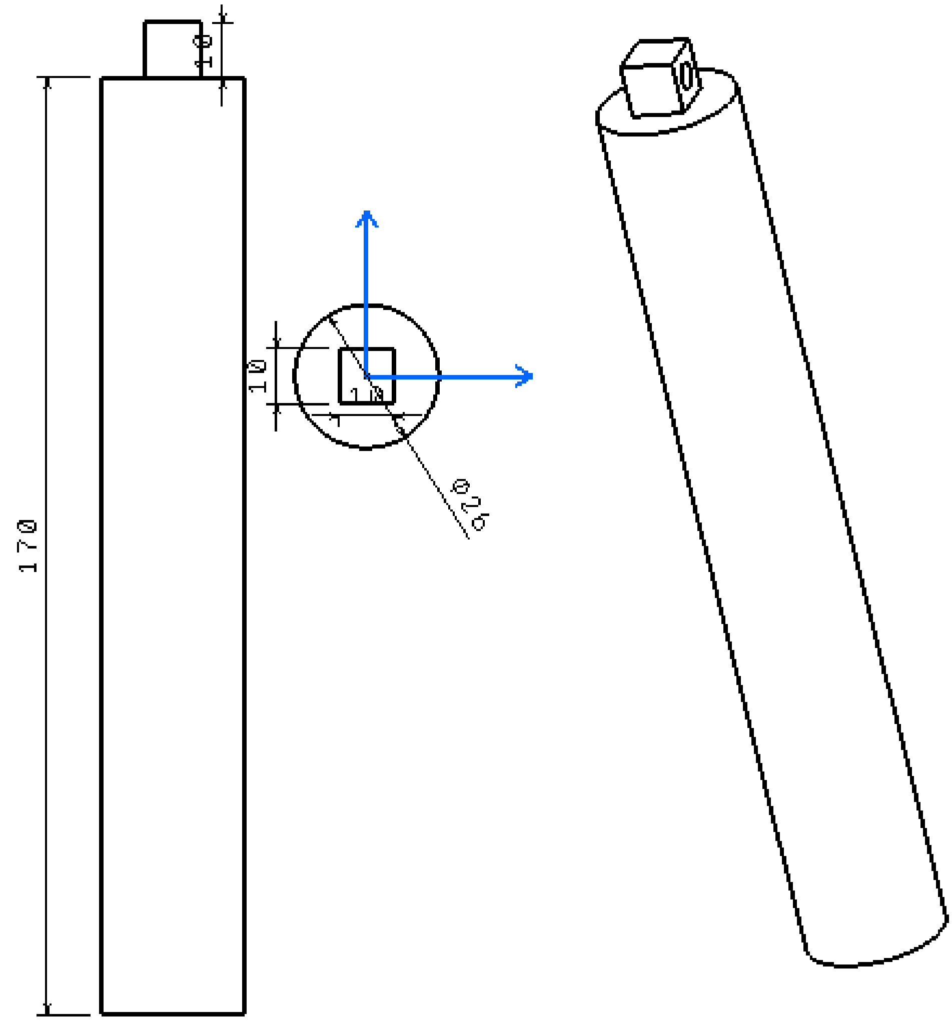
Length of the first part of the Crank Shaft: L_{cr1}=140mm

Diameter of the Crank Shaft: D_{cr}=26mm

Outside diameter of the Gear:d_{go1}=24mm

Diameter of the Gear: d_{g1}=20mm

- Crank Shaft (front).



Drawing of the second part of the Crank Shaft

Material: Stainless Steel 4340 ALLOY

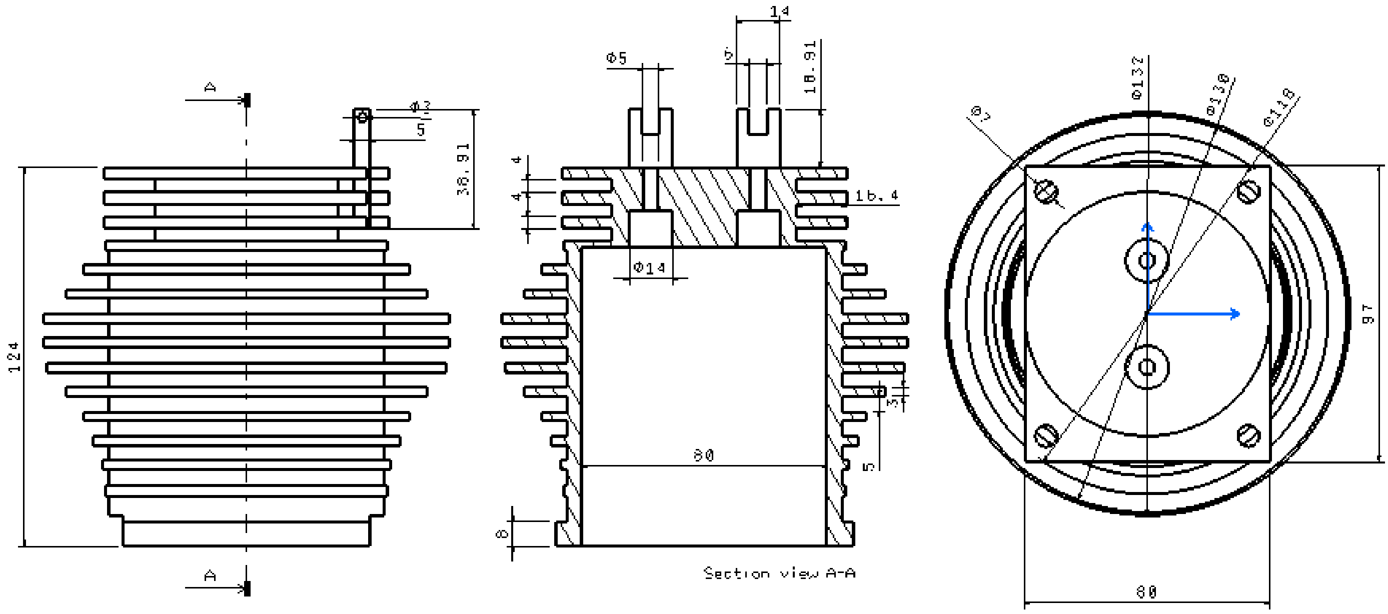
Quantity:1

Main constructive elements:

Length of the second part of the Crank Shaft: L_{cr2}=170mm

Diameter of the second Crank Shaft: D_{cr}=26mm

8. Cylinder Head



Drawing of the Cylinder Head

Material: Aluminum ALLOY 7075

Quantity: 3

Main constructive elements:

Inside diameter of the Cylinder Head: $D_{c.h}$=80mm

Length of the Cylinder Head: $L_{c.h}$=124mm

Lower diameter of the valve hole is = 14mm

Upper diameter of the valve hole is =5mm

Thickness between rings in the lower part is =5mm

Thickness of the rings in the lower part is =3mm

Thickness of the rings in the upper part is =4mm

Thickness between the rings in the upper part is =4mm

Thickness of the vertical walls is =5mm

Thickness of the horizontal walls is =8mm

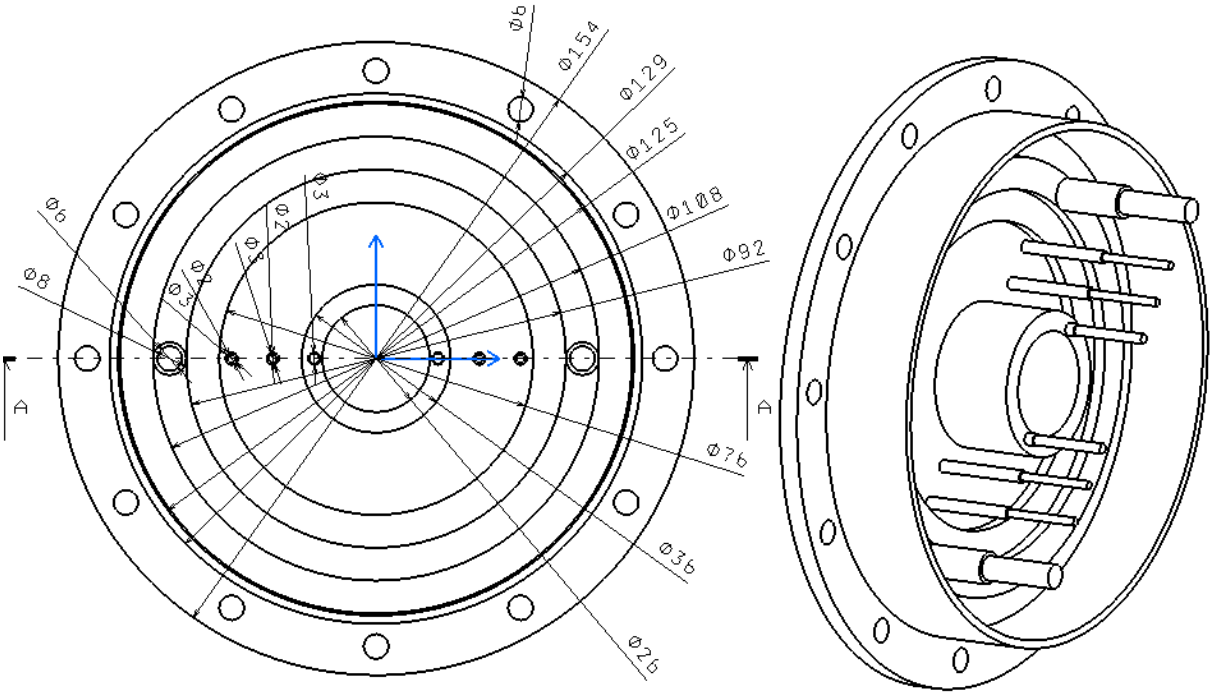
9. Bearings.

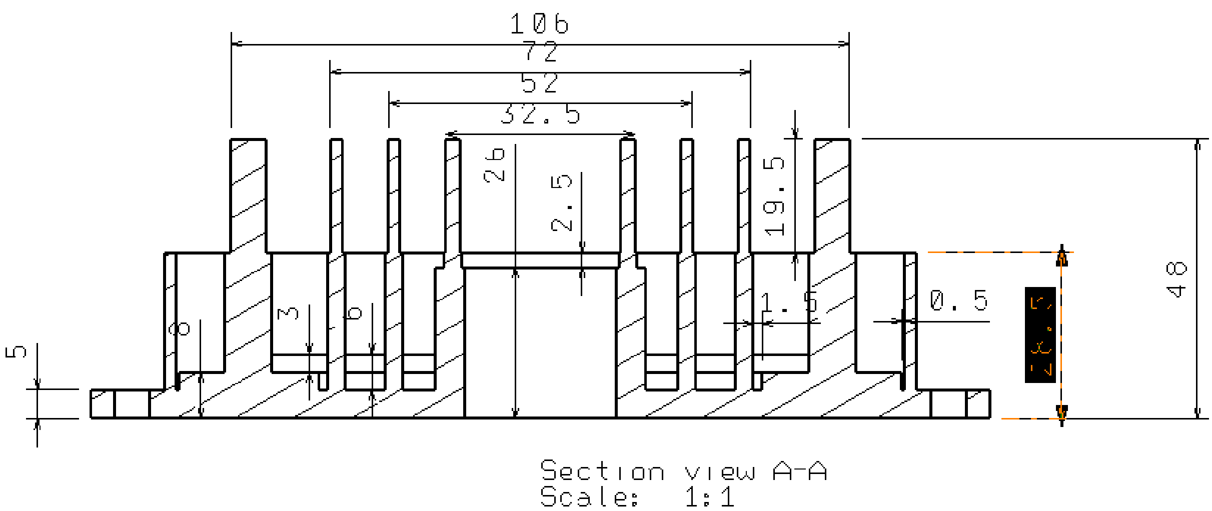
The main bearings that the crank shaft is rotated about are these two bearings.

- Rear Bearing

Material: C63000 Nickel Bronze Aluminum

Quantity: 1

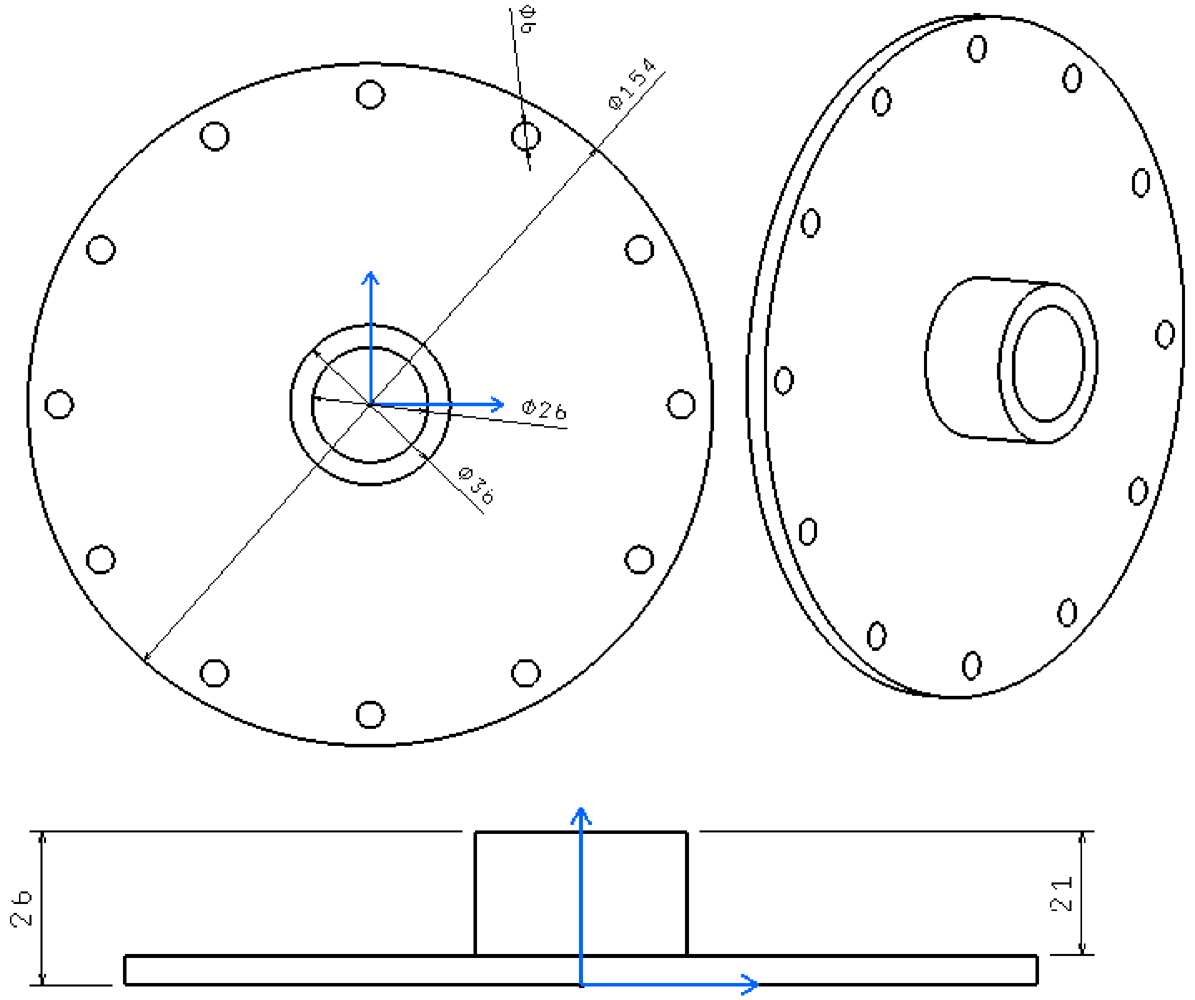




- Front Bearing

Material:C6300 Nickel Bronze Aluminum

Quantity: 1



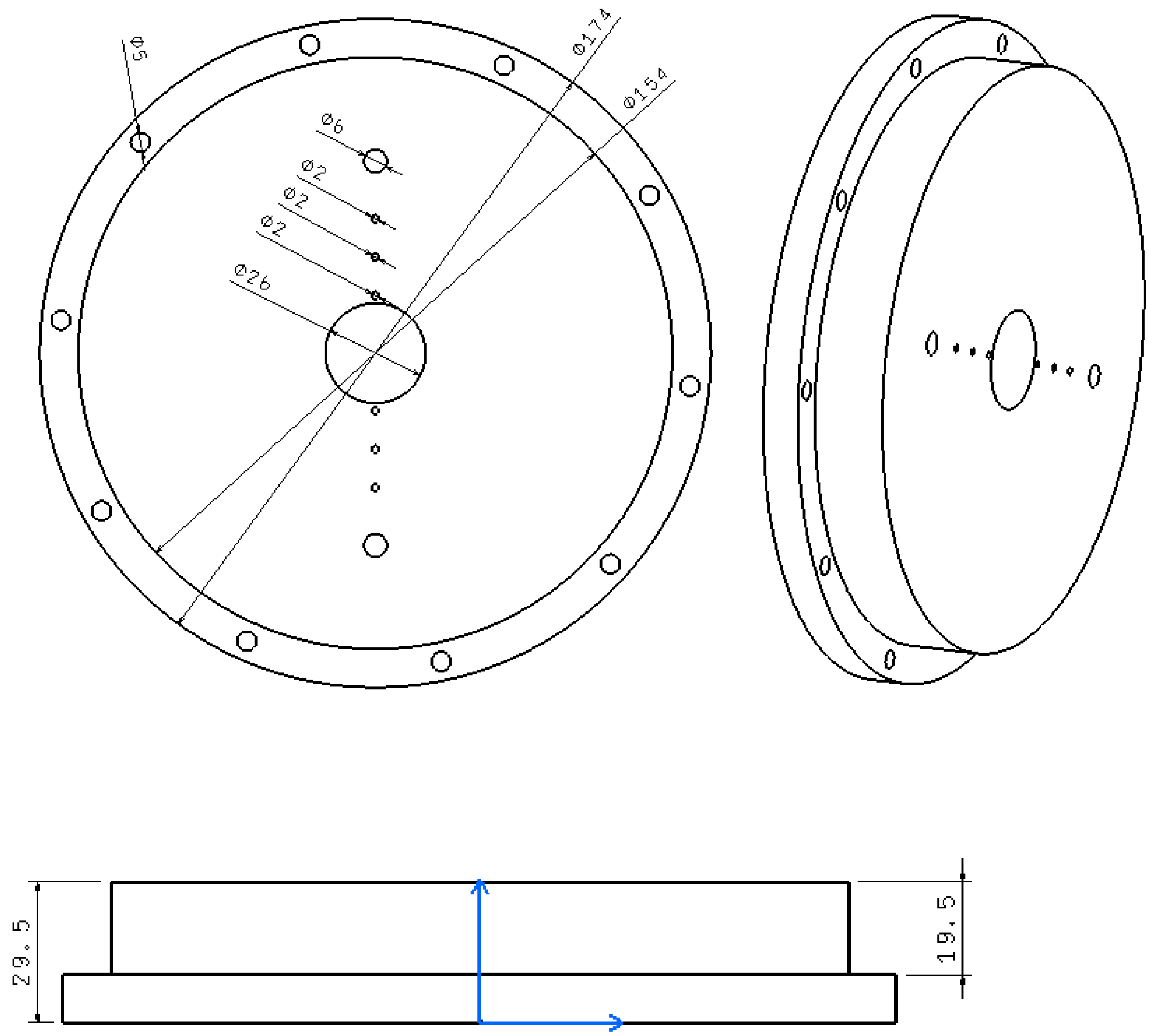
Drawing of the Front Bearing

10. Gear Box.

Gear Box is the cover of the whole gear drive mechanism so it is very important part of the Design although there is not so much stresses acting on it.

Material: Aluminum Alloy 7075

Quantity: 1



Drawing of the Gear Box

11. Gear drives mechanism:

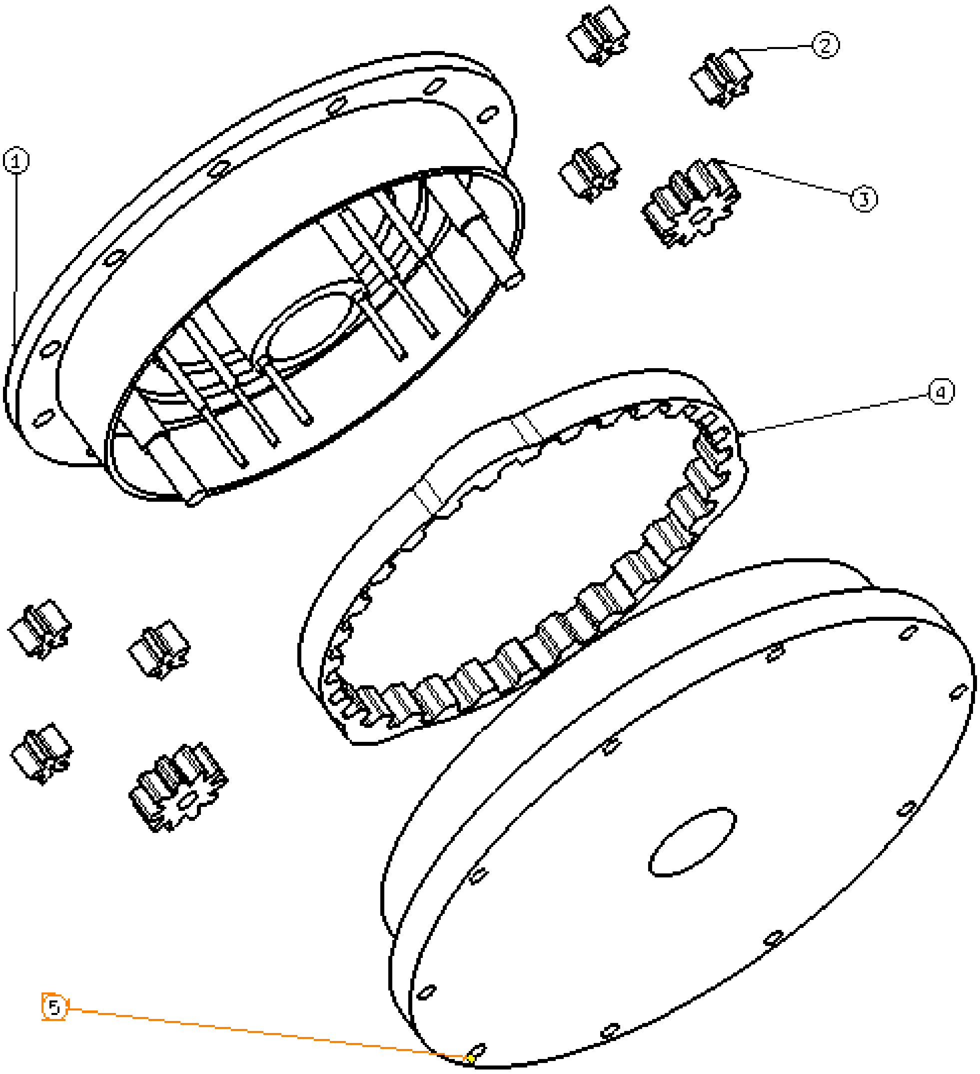
The gear drive mechanism is located on the rear side of the Radial engine in the so called of myself gear box. The base of this gear box to which base rely all of the gears is the rear bearing. The gear mechanism is being used to reduce the input torque and to give an appropriate torque to the cam mechanism.

It is a compound mechanism and is composed of several gears, all of the gears with different diameters and number of teeth.

It is a difficult mechanism to calculate and takes time. It is like that due to the small dimensions of the engine and the small amount of space in which the gears should be located.

The mechanism I thought out is composed of one base (rear bearing also used as a gear box base), six gears with 10mm diameters, 2 gears with 20mm diameters, one gear that is engraved in the rear Crank Shaft and is 20mm diameter and one internal gear which center of rotation is coincident with the center of rotation of the Crank Shaft and is 120mm diameter. So all of the gear connected transforms the torque that comes from the Crank Shaft into six times less torque and by the external side of the largest gear is transferred to the cam mechanism.

The number of teeth is calculated by various formulas and the angular velocity of a gear is to other gear angular velocity as the second ones diameter is to the first one.



Bill of Material: Gear Drive Mechanism

Part Name	Quantity	Material
Bearing	1	C6300 Nickel Aluminum Bronze
Gear “10”	6	Steel Alloy 4340
Gear “20”	2	Steel Alloy 4340
Gear “120”	1	Steel Alloy 4340
Gear Box	1	Aluminum Alloy 7075

- Calculation of the Gear Drive Mechanism

To calculate the mechanism we start from the beginning which is the gear engraved in the shaft with diameter 20 mm and the gear next to it with diameter 10 mm.

$$\frac{\omega_1}{\omega_2} = \frac{d_2}{d_1} = \frac{20}{10} = 2:1$$

It means that the angular velocity in the second gear with a smaller diameter is twice bigger than the angular velocity in the first gear.

Second calculation is between the second and the third gears which are with same diameters (10 mm).

$$\frac{\omega_2}{\omega_3} = \frac{d_3}{d_2} = \frac{10}{10} = 1:1$$

It means that the angular velocities in both gears are the same.

The ratio between the third and the fourth gears are the same because of the diameters they have (10mm).

Next calculation is between gear number 4 that is with 10 mm diameter and gear number 5 that is with 20 mm diameter.

$$\frac{\omega_4}{\omega_5} = \frac{d_5}{d_4} = \frac{10}{20} = 1:2$$

It means that the angular velocity of the fifth gear (20mm) is twice smaller than the angular velocity of the fourth gear. So we have the same velocity as we had in the first shaft.

Now we have to calculate the angular velocity of the last gear the biggest one with diameter 120 mm.

$$\frac{\omega_5}{\omega_6} = \frac{d_6}{d_5} = \frac{20}{120} = 1:6$$

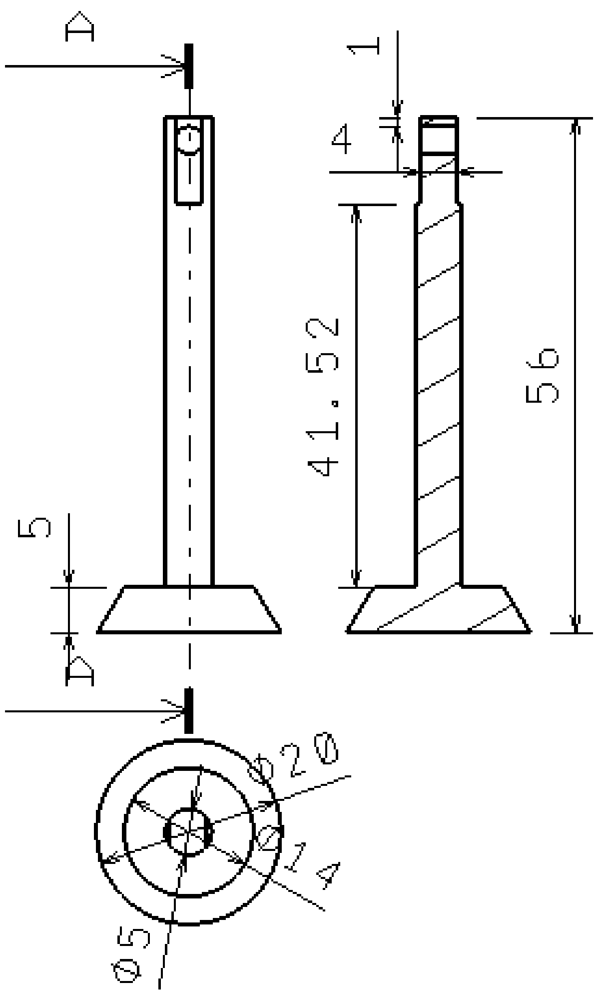
It means that the output torque will be six times smaller than the input and that is exactly what we needed of.

12. Valves.

The Valves are located under 90 degrees on the top of the cylinder. They cross-section the cylinder and the cylinder head. There are two valves connected with each cylinder. The first one is called intake valve and is used to let fuel go into the cylinder. The other one is called exhaust valve and is used to let the gases after the process go out of the engine.

Material: Stainless Steel 316L

Quantity: 6



Drawing of the Valve

13. Cam Mechanism

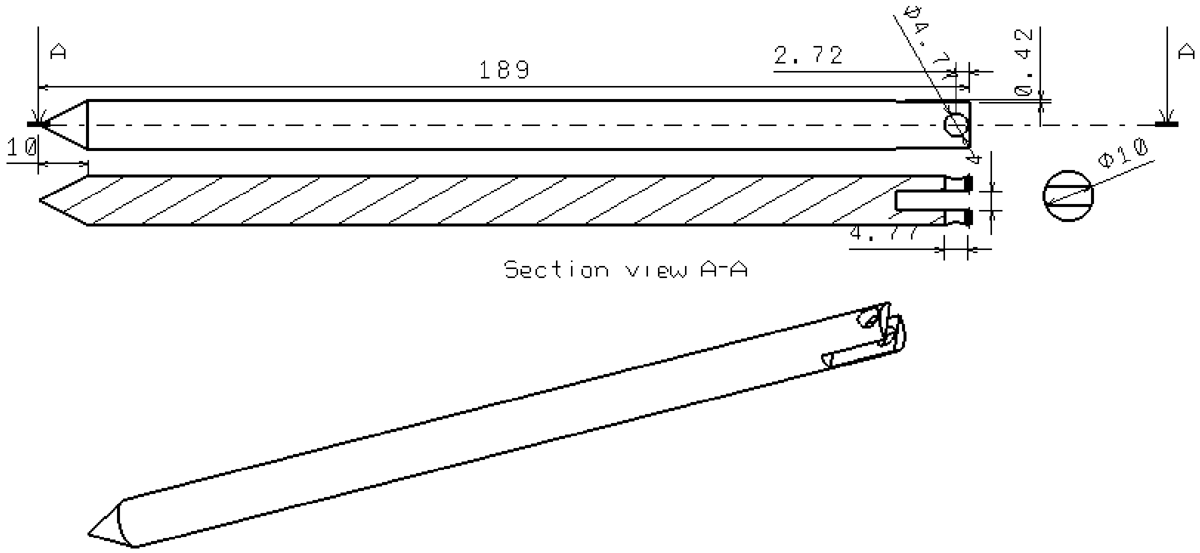
The Cam mechanism is one the most important mechanism. It takes plenty of time until calculations are made and all of the parts are designed. It is a delicate mechanism because of its sizes. Every size should be really precise. In the cam mechanism we have four important parts. First one is the pushing rod that is connected to the largest gear (120mm diameter) through a hole in the crank case, and in the other side is connected to an arm located at the top of the cylinder head. And both mechanisms are connected by socket. On the other side the arm of the cam is connected to the valve by a socket too. On every cylinder is located two cam mechanism.

- Pushing Rod

It is moving up and down and the main purpose of this rod is to push the arm that pushes the valve down.

Material: Stainless Steel 416

Quantity: 6



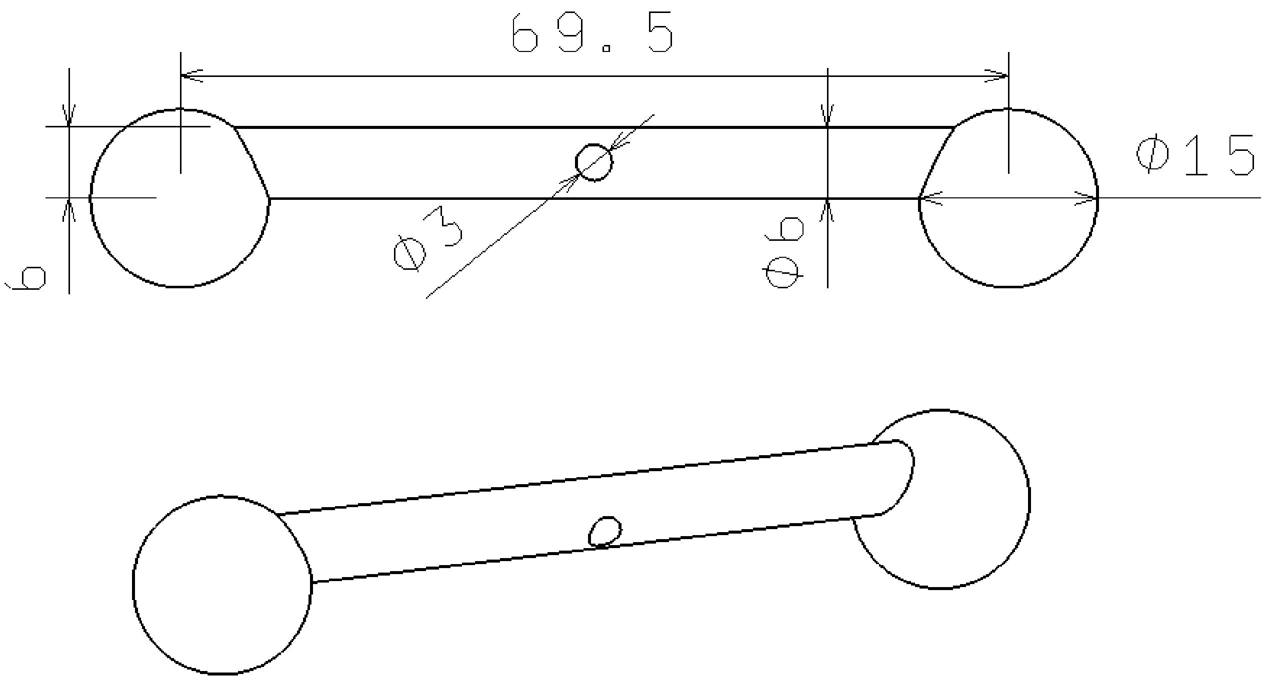
Drawing of the Pushing Rod

- Arm of the Cam mechanism

It is moving about an axis fixed to the cylinder head. The arm is being constantly pushed by the pushing rod and acts the force of the rod to the valve through a socket.

Material: Stainless Steel 416

Quantity: 6



Drawing of the Arm

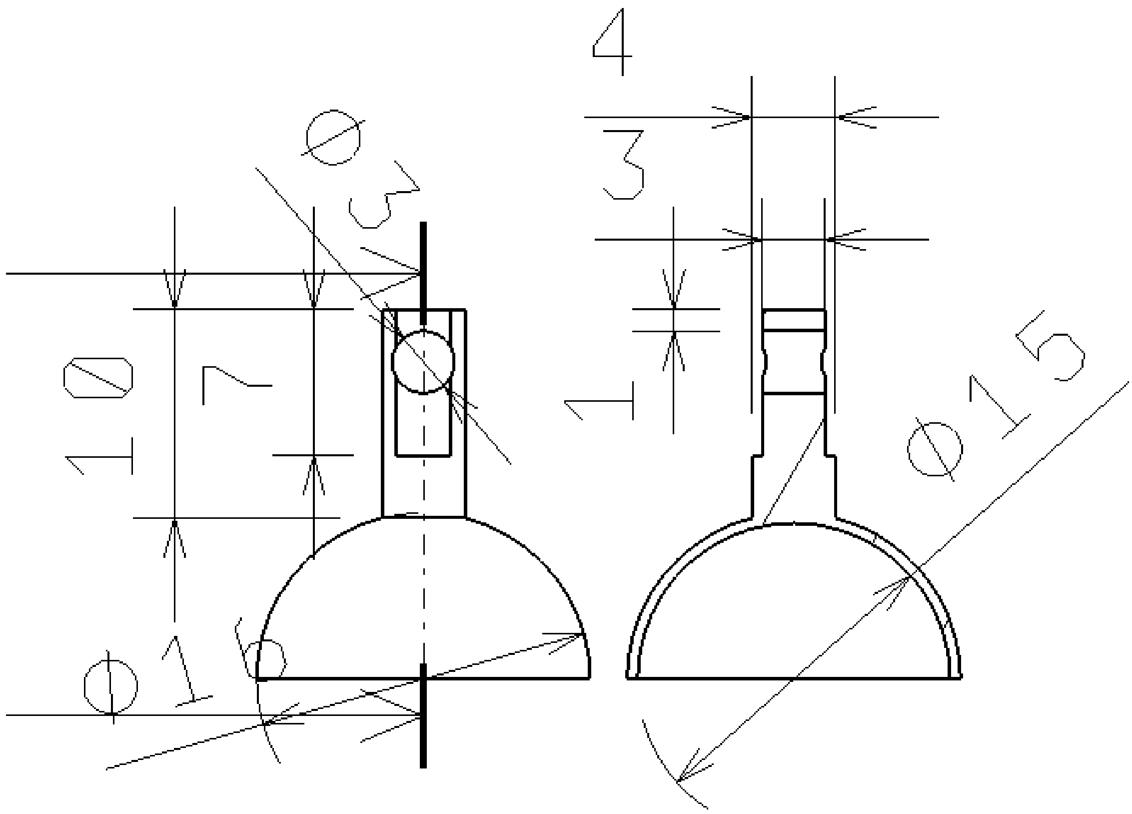
- Sockets

There are two type s of socket. The first one is the one that connects the pushing rod to the arm. The other one is the one that connects the arm to the valve. The only difference between the two of them is at the way they are connected not to the arm but to the pushing rod and the valve.

- Socket connecting the Pushing rod and the Arm

Material: Stainless Steel 416

Quantity: 6

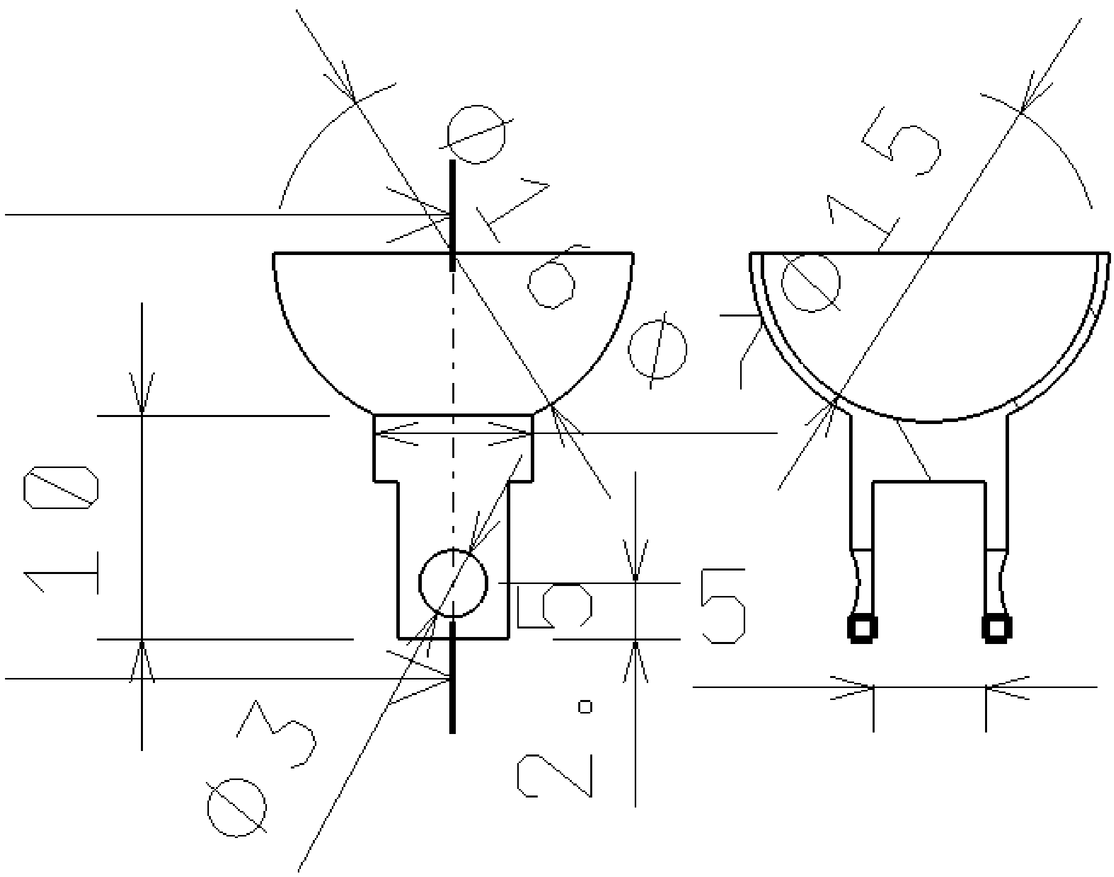


Drawing of the Socket connected to the Pushing Rod

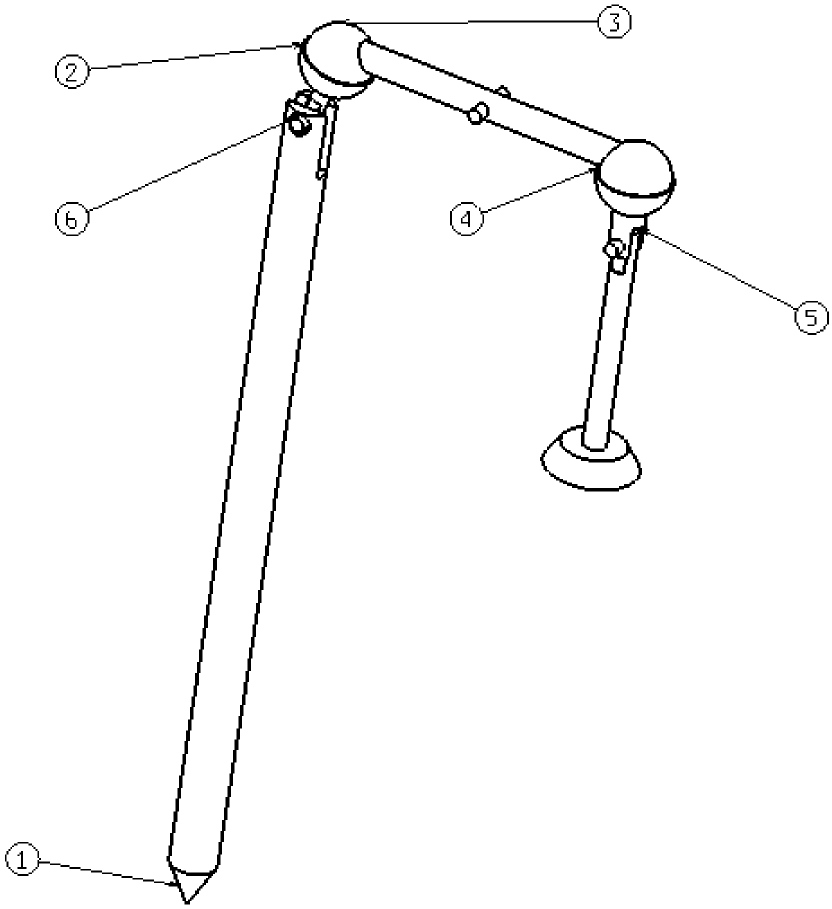
- Socket connecting the Arm with the Valve

Material: Stainless steel 416

Quantity: 6



Drawing of the Socket connected to the Valve



Assembly of Cam Mechanism

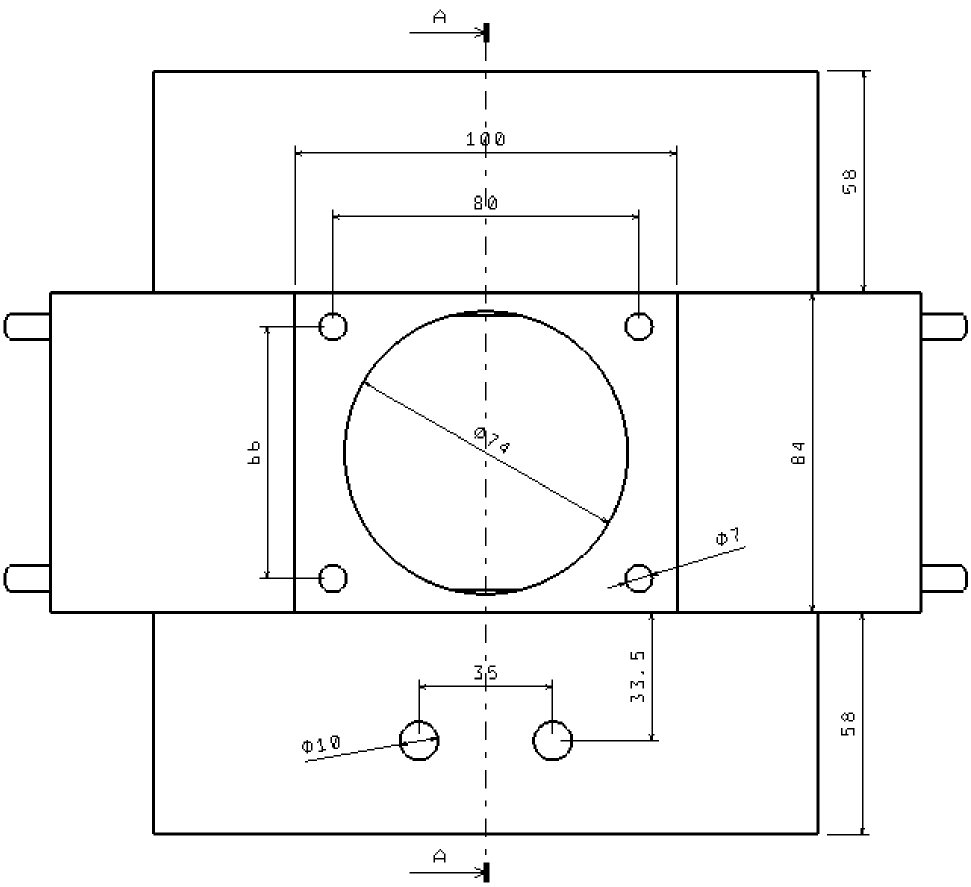
N:	Part Name	Quantity	Material
1.	Pushing Rod	6	Stainless Steel 416
2.	Socket Rod	6	Stainless Steel 416
3.	Arm	6	Stainless Steel 416
4.	Socket Valve	6	Stainless Steel 416
5.	Valve	6	Steel Stainless 316L
6.	Bolt 3mm10mm	18	Stainless Steel

14. Crank Case

The Crank Case is the external part of the whole engine. It is difficult to be designed and difficult to be manufactured. When designing such a part is important to know almost all of the dimensions of the Engine so you could fit every single part of it. Inside the Crank Case are located the Auxiliary rods, the Master rod, the Crank Cheeks, The main journal and small parts of the Crank Shafts. Also in the outside of the crank case on the walls are located the cylinders. They are connected to the crank case by special bolts. On the front and the rear side of the crank case are located also the gear box mechanism, and the rear and front bearing.

Material: Aluminum ALLOY 7075

Quantity: 1



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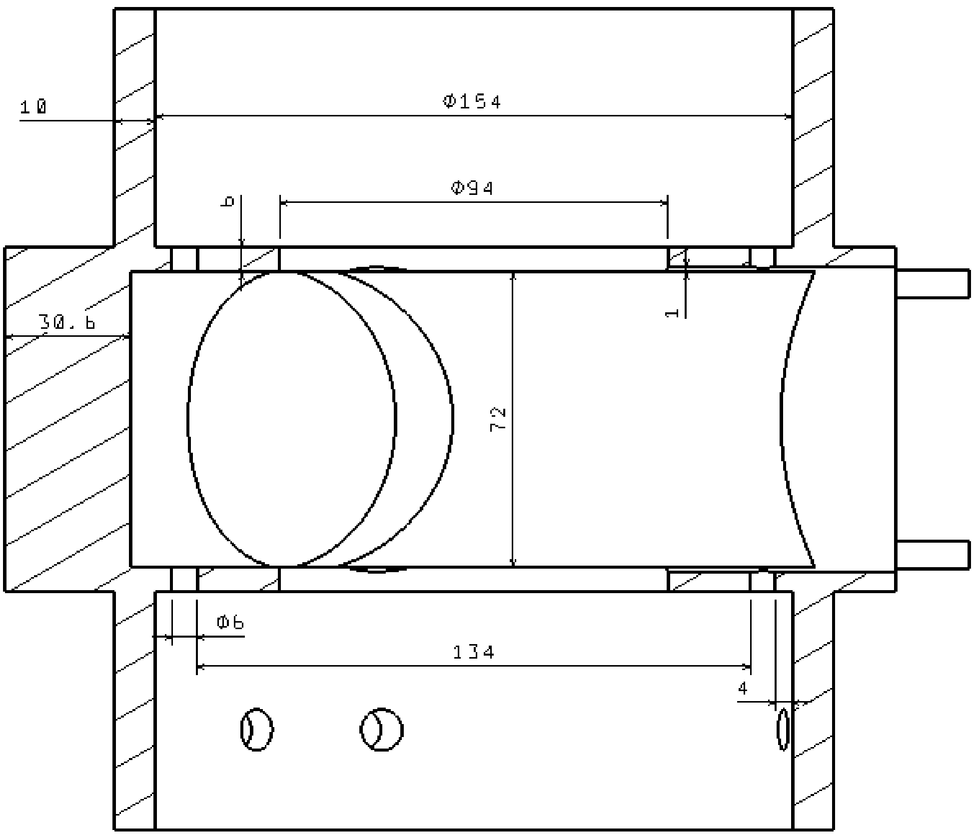
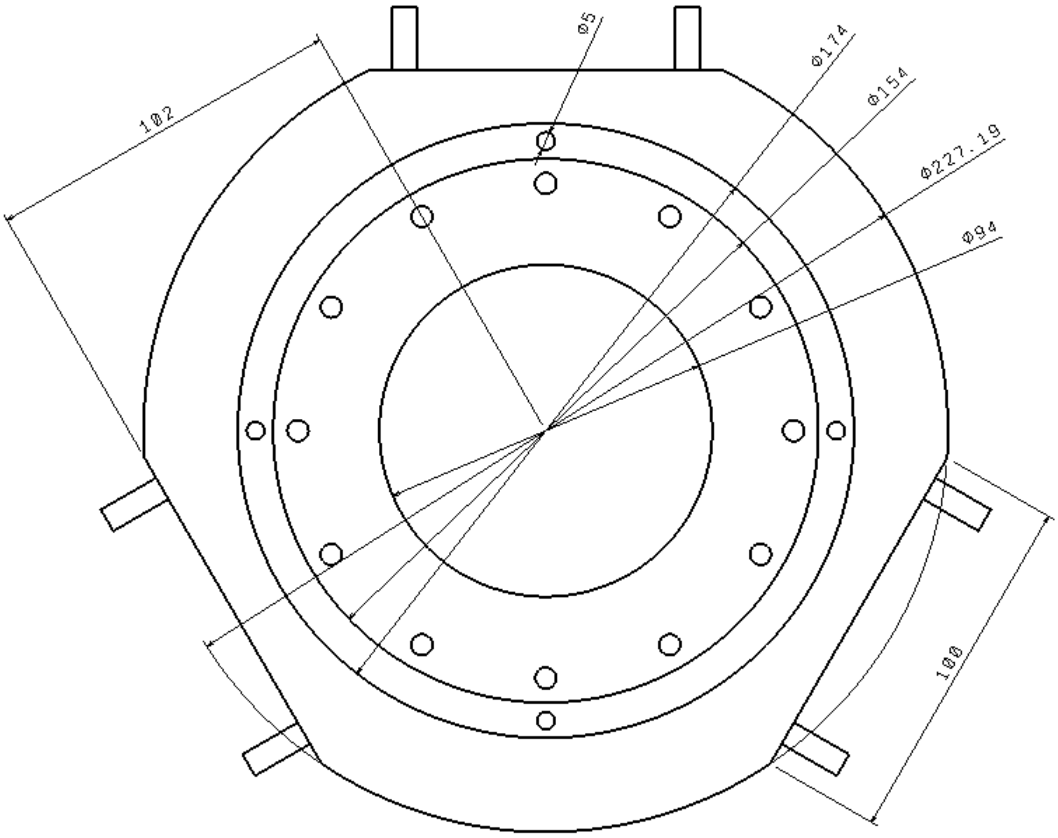


Section view A-A

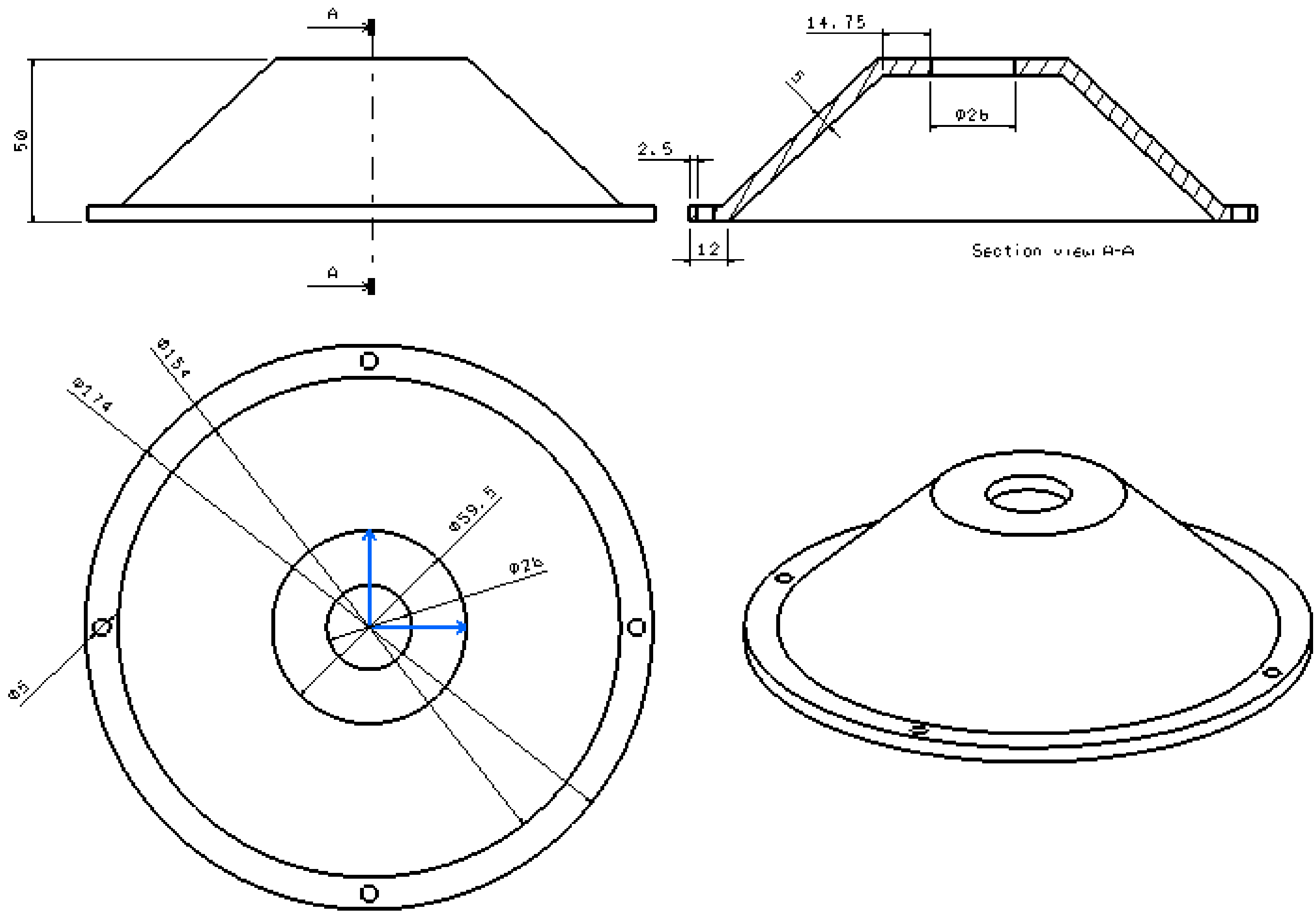


15. Front Cover

This is the cover of the front part of the engine.

Material: Aluminum ALLOY 7075

Quantity: 1

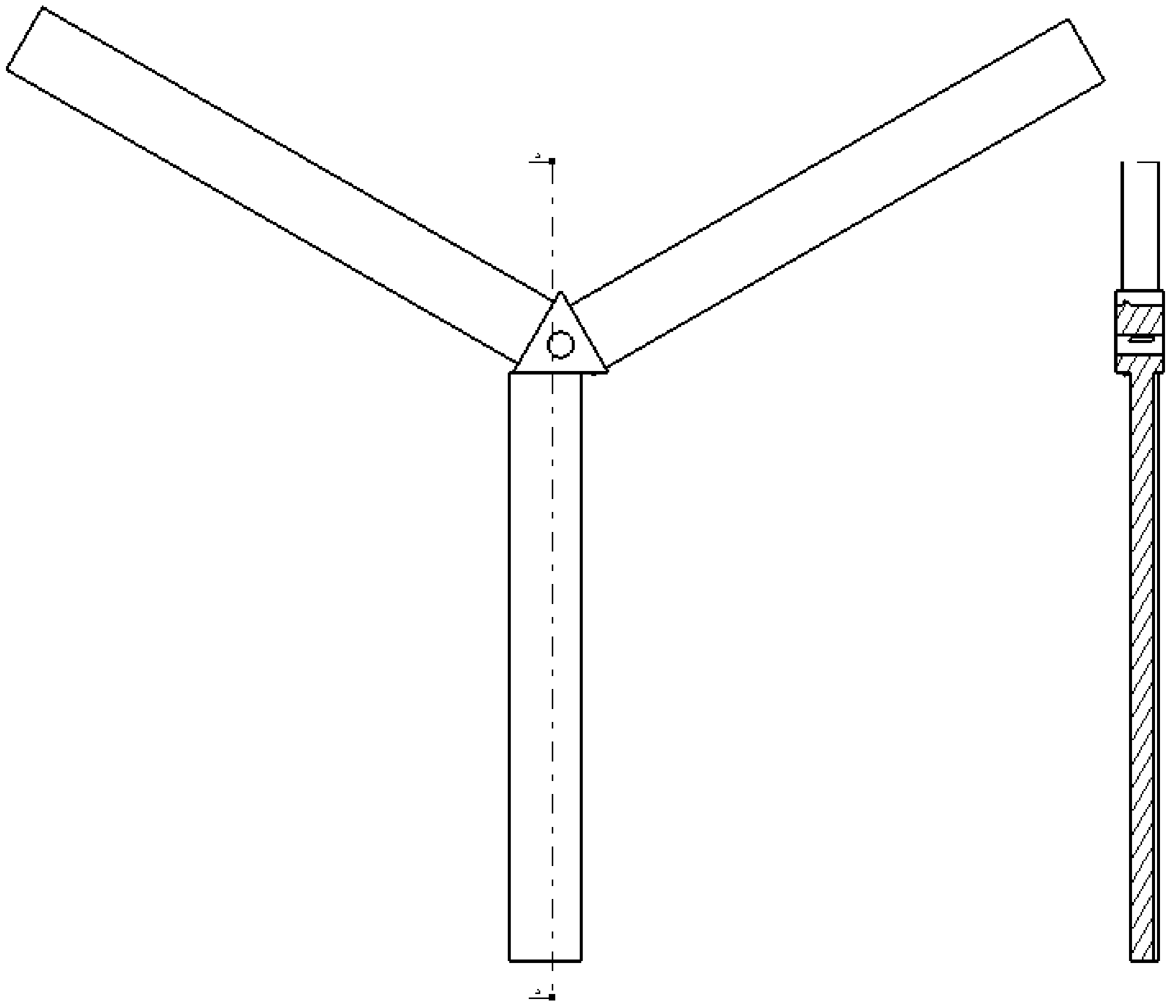


Drawing of the Front Cover

16. Propeller

Material: Plastic

Quantity: 3



17. Materials used in the parts of the Radial Engine

- Aluminum ALLOY 7075- a cold finished aluminum wrought product, has the highest strength of all aluminum machine alloys. The –T6 and –T651 tempers have the typical tensile strength, which is higher than many mild steels. Due to its highly strength, ALLOY 7075 is used for highly stressed structural parts. Applications include aircraft fittings, gears and shafts, fuse parts, meter shafts and gears, missile parts, regulating valve parts, worm gears, keys and various others commercial aircraft, defense and aerospace equipment.

- Steel alloy 4340- it is heat treatable, low alloy steel containing nickel, chromium and molybdenum. It is known for its toughness and capability of developing high strength in the heat treated condition while retaining good fatigue strength. Typical applications are for structural use such as aircraft landing gear, power transmission gears and shafts and other structural parts.

- Steel 12L14- it gives a smooth, machined surface and because of its low friction components allows for increased tool life. It is used extensively in automatic screw machines for manufacturing numerous parts requiring considerable machining and close tolerances, along with a smooth finish. It can be used for maximum advantage where considerable machining is required, such as bushings, inserts, couplings, and hydraulic hose fittings. With good ductility, 12L14 is suitable for parts involving crimping bending or riveting.

- Aluminum alloy 4032-T6- a cold finished aluminum wrought product is suggested for applications requiring wear and abrasion resistance. It eliminates the need for hard coat anodizing commonly required in applications using 6061 and 6262 alloys. Superior wear and abrasion resistance of this alloy is achieved through high silicon and nickel content. Applications include brake master cylinders, transmission valves, copier parts, bushings for rack and pinion steering systems, sound recording devices, bearings hydraulic applications and forged pistons.

- Aluminum alloy 2117-T4- This material is commonly used I aeronautical applications for riveting operations.

- Stainless Steel 316L- It is an austenitic chromium-nickel stainless steel containing molybdenum. This additions increases general corrosion resistance, improves resistance to pitting from chloride ion solutions and provides increased strength at elevated temperatures. Typically uses include exhaust manifolds, furnace parts, pharmaceutical and photographic equipment, valve and pump trim, chemical equipment, digesters, tanks, evaporators, pulp, paper and textile processing equipment, parts exposed to marine atmospheres and tubing. Type 316L is used extensively for weldments where its immunity to carbide precipitation due to welding assures optimum corrosion resistance.

- C6300 Nickel Aluminum Bronze- It combines high strength and wear resistance under severe loading conditions. The addition of nickel increases the alloy strength without diminishing its ductility, toughness and corrosion resistance.

- Stainless Steel 416- it was the first machining stainless steel. It is a heat treatable chromium steel with excellent machinability and non galling characteristics. The alloy is magnetic in all conditions. It uses include a wide variety of screw machine parts including nuts, bolts, screws, gears and pinions, valve trim, shafts and axles.

18. Parts specifications` table

N.	Part Name	Material	Quantity
1.	Crank Case	Aluminum Alloy 7075	1
2.	Crank Shaft (Rear)	Steel Alloy 4340	1
3.	Crank Shaft (Front)	Steel Alloy 4340	1
4.	Crank Cheek	Steel Alloy 4340	2
5.	Main Journal	Steel Alloy 4340	1
6.	Master Rod	Aluminum Alloy 7075	1
7.	Auxiliary Rod	Aluminum Alloy 7075	2
8.	Piston	Aluminum Alloy 4032-T6	3
9.	Cylinder	Stainless Steel 316L	3
10.	Cylinder Head	Aluminum Alloy 7075	3
11.	Front Bearing	C6300 Nickel Aluminum Bronze	1
12.	Rear Bearing	C6300 Nickel Aluminum Bronze	1
13.	Gear “120”	Steel Alloy 4340	1
14.	Gear “20”	Steel Alloy 4340	2
15.	Gear “10”	Steel Alloy 4340	6
16.	Gear Box	Aluminum Alloy 7075	1
17.	Pushing Rod	Stainless Steel 416	6
18.	Socket(Rod)	Stainless Steel 416	6
19.	Socket Valve	Stainless Steel 416	6
20.	Arm	Stainless Steel 416	6
21.	Valve	Steel Stainless 316L	6
22.	Front Cover	Aluminum Alloy 7075	1
23.	Propeller	Aluminum Alloy 7075	1
24.	Piston Bolt	Stainless Steel 12L14	3
25.	Master Rod Bolt	Stainless Steel 12L14	2
26.	Bolt 3mm*10mm	Stainless Steel 12L14	18
27.	Bolt 5mm*10mm	Stainless Steel 12L14	4
28.	Bolt 5mm*15mm	Stainless Steel 12L14	10
29.	Bolt 6mm*10mm	Stainless Steel 12L14	24

III. Conclusion.

Designing of an engine at all is a very complicated process which involves serious of other processes that are hard to be designed. It takes a very long time of thinking of the proper dimensions, proper material and even the proper shape of all different parts. The design of the radial engine is not less complicated process than any other engine. It has crank case that is a little bit different than the other engines but although that is hard to be figured out too. It also has pistons, cylinders, rods, it has a cam mechanism that in the ordinary engines is called cam shaft, and lets not forget about the gear mechanism that is some ways different than the gear mechanism in the ordinary engines but the way it works is the same in both engines. And the crank shaft- the same as in an ordinary engine but with less journals, easier to be manufactured. The crank shaft is a really amazing part of the engine, torque is transmitted into distance is exciting.

Although not that used nowadays radial engine is a cute little engine that was in the beginning of the aerospace transportation. It was very helpful and mainly used because of its small weight and size. That makes it comfortable and suitable for any machine that is close of space. Despite of its small size and weight it does not make it less powerful than other engines. The same way the in-line engines are more powerful with more cylinders it is more powerful with more rows of cylinders. It was also comfortable for the World War II airplanes when the engine was in its peak. When it is war you need more space for fuel, power, weapons and bullets than any other things and than any other time.

There are several different stages that I passed until the final engine was completely designed. In the way to the end I met lots of difficulties and problem about the way every single part is done. Had lots of problems also with the program I used to make the whole assembly and simulations- CATIA.

The first stage is about orientation of what the radial engine looks like and how it performs. As I said earlier it was mainly used long time ago and although there are plenty of engines like this nowadays it is really hard to find any information about it. I needed information also about the various parts and mechanisms used in it for it was difficult for me to imagine how it works.

Second part was meeting CATIA and start drawing on it. In the beginning it was new for me although has lots of stuff similar to the other Cad / Cam programs. After a week it became easier to me and I could begin with the main part of the project.

Next part was start thinking about the shape first after that about the dimensions so each part could fit perfectly on its spot. It takes long time until every part is last dimensioned. I passed through lots of problems about that for it is difficult to think about all dimensions and shapes. And for me it was very important the engine to be as good from outside as it is from inside.

The third and the major step was about designing the two major mechanisms that are in the radial engine. The gear drive mechanism took me about a week to design it. First it was difficult to calculate the gears and to make them that they could fit in the gear box and the same way they will reduce and transform the torque from the crank shaft six times. The cam mechanism was also very hard to be designed, even harder. It took me about two weeks. The difficulties here came not that much from the complication of the mechanism as a whole but from the program and the knowledge I had about it and that was still not enough.

After all difficulties about dimensions, shape, different mechanism and nevertheless calculations about the stress and the strength of the materials the radial engine came to an end.

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