Czech Technical University in Prague

Faculty of Mechanical Engineering

Department of Automotive, Combustion Engine and Railway Engineering

Master's thesis

Design of a four-stroke single-cylinder motocross engine

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CTU CZECH TECHNICAL UNIVERSITY IN PRAGUE

MASTER'S THESIS ASSIGNMENT

I. Personal and study details

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Master's thesis title in English:

Design of a four-stroke single-cylinder motocross engine

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Návrh čtyřdobého jednoválcového motokrosového motoru

Guidelines:

-Make a review of contemporary motocross single-cylinder engines up to 250ccm
-Design a conceptual design of the motocross engine and build a thermodynamic model
-In CAD software, design the power unit with a focus on the crank mechanism, valve train and balance shaft; Only process the gearbox shaft and clutch schematically
-Make a strength calculation of the crank mechanism
-Summarize the advantages and disadvantages of this concept

Bibliography / sources:

Name and workplace of master's thesis supervisor:

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Date of master's thesis assignment: **30.10.2020** Deadline for master's thesis submission: **06.01.2021**

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Supervisor's signature	Head of department's signature	Dean's signature

III. Assignment receipt

The student acknowledges that the master's thesis is an individual work. The student must produce his thesis without the assistance of others, with the exception of provided consultations. Within the master's thesis, the author must state the names of consultants and include a list of references.

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Annotation

Author:	Bc. Adam Bureš
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Supervisors:	Ing. Jindřich Hořenín (CTU in Prague) Satrio Wicaksono ST, M.Eng., Ph.D. (ITB)
Abstract:	This master's thesis deals with the design of a four-stroke single-cylinder motocross engine up to 250ccm. The content of the theoretical part is a review of three contemporary motocross engines. The practical part deals with the conceptual design of the engine itself, including the design in CAD software.
Keywords:	Engine, motocross, motorcycle, single-cylinder, four-stroke, design, Creo Parametric
Number of pages:	104
Number of pictures:	95
Number of tables:	20

Declaration

I hereby declare that I have completed this thesis independently and that I have listed all the literature and publication used in accordance with the methodological guidelines about adhering to ethical principles in the preparation of the final thesis.

In Prague, January 6, 2020

Bc. Adam Bureš

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I would like to thank my supervisor Ing. Jinřich Hořenín for his professional guidance, advice and patience with this diploma thesis. I would also like to thank Ricardo Prague s.r.o, especially Ing. Tomáš Zemek, for enabling the processing of this thesis. And finally, I would like to thank my family and friends for their support throughout my studies.

Table of contents

1 INTRODUCTION

The topic of my diploma thesis was assigned by Ricardo Prague s.r.o. This company deals with projects in the field of construction of internal combustion engines, transmissions, vehicles, hybrid and electrical systems, as well as in the field of environmental impact analysis. The aim of this thesis is a conceptual design of a four-stroke single-cylinder motocross engine with an engine displacement of up to 250ccm.

In the theoretical part, basic information about motocross will be given first. Then I will describe the basic difference between a two-stroke and a four-stroke engine, because motocross bikes are made with both types of engines. The main focus of the theoretical part will consist of reviews of three engines of motocross bikes, which are currently manufactured, namely: KTM 250 SX-F, Yamaha YZ250F and Honda CRF 250R. At the end of this section, a comparison of these three engines will be made.

In the practical part I will deal with the conceptual design of the motocross engine. First, I will draft the basic parameters of the designed engine such as bore, stroke, compression ratio, engine displacement and basic parameters of the crank mechanism. Then I will compile a simplified thermodynamic model in the GT-Power software, which will give me the required information about the designed engine. Then, in the Creo Parametric software, I will design the individual parts of the engine, namely: the crank mechanism, valvetrain, balancer shaft, cylinder head, cylinder, engine cases, simplified clutch and simplified transmission. In the same software, I then perform a strength analysis of the crank mechanism using the FEM method. Finally, the advantages and disadvantages of this conceptual design will be described.

2 THE THEORETICAL PART

2.1 MOTOCROSS

2.1.1 ABOUT MOTOCROSS

Motocross is a type of motorcycle racing (sometimes called as an off-road motorcycle race) and it is one of the most popular motorsports in the world. Motocross originated in the United Kingdom and the first known motocross racing took place in 1924 in Camberley. Motocross races are held on a closed outdoor circuit with several obstacles, ramps and turns. The track consists of grass, gravel, mud and sand. In each race, about 25 riders can participate. The competitors line up in front of the gate and the race will begin after the gate has fallen. Motocross races take the form of two races (in most cases within one day) and each race is scored as a separate race. The points earned in each race add to the overall ranking. Each race takes a certain period of time. For example, in the World Championship races it is 35 minutes + 2 laps, the International Championship of the Czech Republic takes 30 minutes +2 laps and in regional championships it is 20 minutes + 2 laps. [1] [2] [3]

FIG. 1 MOTOCROSS RACING [4]

2.1.2 MOTOCROSS BIKES

FIG. 2 ILLUSTRATIVE PICTURE OF A MOTOCROSS BIKE [5]

The motocross bike is sometimes called a dirt bike. These bikes are constructed mainly for the track. The main goal is to design and build a motorcycle as fast and light as possible. These motorbikes are designed to reach maximum speed very quickly. They are extremely aggressive, and the size of motocross bike is smaller than a usual street motorcycle. The largest manufacturers of motocross bikes are KTM, Honda, Kawasaki, Suzuki, Yamaha, Husqvarna, Husaberg and Sherco.

In order to reduce the weight of the dirt bike as much as possible, a lot of common components that are not needed on the track are missing. At the first sight we can see that parts such as mirrors, speedometers, kickstands, electric starter and stereo systems are missing. Other important missing parts are the front and rear lights, so it is impossible to use a motocross bike at night without proper lighting on a track. Even the small fuel tanks are used to achieve less weight. The frame of a motocross bike is usually made of steel or aluminum.

Motocross bikes are designed and built to be used on a special track during a race, which last about 30 minutes, so they are not comfortable, and they are not suitable for long rides.

The suspension system consists of spring shocks which are stiff. The suspension system must be able to handle the jumps, the rough roads and the rapid acceleration. The dirt bike usually has a long, flat, narrow seat with a hard foam which allows the rider to move well for a better traction, although riders often stand during the race.

The dirt bikes apply single-cylinder engines designed for high speeds and they feature a lighter flywheel which makes for more aggressive power delivery. The gear ratios in a gearbox are very close for better acceleration. The exhaust system does not contain a spark arrestor and there is only a little silencing.

Motocross bikes can have an engine displacement from 50ccm (a motorcycle for children from 4 to 10 years) up to 450ccm. KTM company manufactures two-stroke and four-stroke motocross engines with the following displacements, which are shown in the Tab. 1 below.

2 stroke engines	50ccm, 65ccm, 85ccm, 125ccm, 150ccm ,250ccm
4 stroke engines	250ccm, 350ccm, 450ccm

TAB. 1 OVERVIEW OF ENGINES BY DISPLACEMENT [5]

A power output of engines with a displacement of 450ccm ranges from approximately 50 HP to 55 HP (37-41 kW). A power output of engines with a displacement of 250ccm ranges from approximately 35 HP to 40 HP (26-30 kW) and a power output of engines with a displacement of 50ccm is approximately 5 kW.

The tires are suitable for off-road racing only because they are greatly grippy on the dirt. They ensure only a little traction on the classic road and therefore it is illegal to use them on the road. [1] [6] [7]

2.1.3 MAJOR MOTOCROSS COMPETITIONS

FIM Motocross World Championship

The FIM (Fédération Internationale de Motocyclisme) was established in 1904. It is the main governing body for motorcycle sports. The main events include Motocross, MotoGP, Superbike, Endurance, Supercross, etc. Motocross is divided into 2 main classes. The first class is called MXGP. In the MXGP class, the minimum age of riders is 16 years. The second class is called MX2. In the MX2 class, the minimum age of riders is 15 years. There are 40 riders in each group who must have an FIM World Championship license. Any rider can use 2 motorcycles during the weekend and the maximum limit of the sound level of the motorcycle is 114 dB/A before the race and 115 dB/A after the race. These two classes are distinguished based on engine type and engine volume. Both parameters are shown in the Tab. 2 below. [8]

MXGP	2-stroke engines	+175ccm up to 250ccm
	4-stroke engines	+290ccm up to 450ccm
MX2	2-stroke engines	+100ccm up to 125ccm
	4-stroke engines	+175ccm up to 250ccm

TAB. 2 CLASSES OF FIM MOTOCROSS WORLD CHAMPIONSHIP [8]

AMA Motocross Championship

The AMA (American Motorcyclist Association) Motocross Championship takes place at twelve major tracks all over the continent of the United Stated. There are 2 classes according to engine displacement. The first class is called 250 Motocross Class and for this class, the rule of limiting the engine volume is 0-125ccm for a two-stroke engine and 150–250ccm for a four-stroke engine. The second class is called 450 Motocross Class and this class includes engines with volume 150–250ccm for two-stroke engine and 251–450ccm for four-stroke engine. [3] [9]

Motocross des Nations

The competition involves teams of three riders of the same nationality. Each nationality can enroll one team only. Each rider of the team must use a different class of motorcycle. The first one drives an MXGP class motorcycle, the second one drives a MX2 motorcycle and the third one drives an *Open* class motorcycle, which fulfills the requirements of the MXGP or the MX2 class. [10]

British Motocross Championship

MXGB is the official British Motocross Championship of the Auto Cycle Union and it is one of the oldest and most respected off-road motorcycle championships in the world. The ACU British Motocross Championship is the only UK body associated to the FIM. [11]

2.1.4 SPORTS DERIVED FROM MOTOCROSS

Enduro

The Enduro motorcycles must have a legal street license. The motorcycle must be equipped with the required equipment for obtaining the license, such as lights, an electric starter, a large fuel tank, an exhaust system, mirrors, etc. The most popular Enduro bikes are 250 ccm and 450ccm and the suspension of Enduro bikes is usually softer than motocross bikes. The Enduro race tests the endurance of a driver on an off-road terrain combined with street racing. A racing circuit consists of water obstacles, jumps, fallen trunks, stones, etc. [12]

FIG. 3 ENDURO MOTORCYCLE YAMAHA WR 450F [13]

Supercross

The biggest difference between motocross and supercross is the race location. While motocross races take place outdoor, the supercross races take place indoors. Because supercross races are organized in large stadiums, spectators can usually see the whole track from their seats. The Supercross track is usually narrower than a motocross track because of less space. [14]

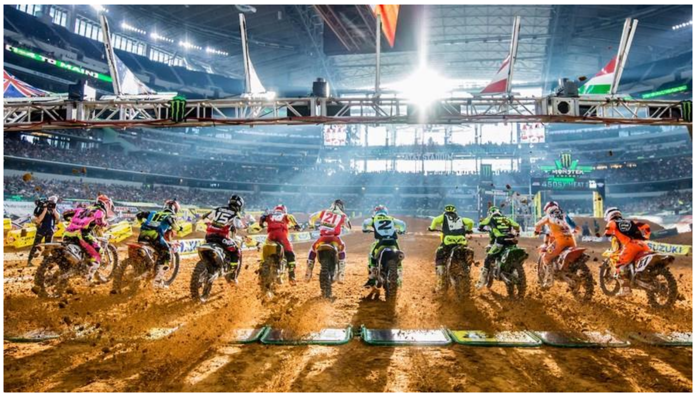


FIG. 4 SUPERCROSS RACE [15]

Supermoto

Supermoto motorcycles are the new hit in Europe, but the price of these motorbikes is very high, and their maintenance is also very expensive. Therefore, some companies stopped producing these bikes due to low demand. A Supermoto track consists of a clean asphalt with the addition of an off-road section with jumps. The Supermoto drivers use 450ccm bikes. [6]

FIG. 5 SUPERMOTO RACE [6]

Freestyle Motocross

Freestyle motocross is one of the most popular variations of motocross. Freestyle motocross is also called as FMX. The riders try to present the best jump and stunt. The goal is to get as many points from referees to win the competition. [16]

FIG. 6 FREESTYLE MOTOCROSS [17]

2.2 THE DIFFERENCE BETWEEN A TWO-STROKE AND A FOUR-STROKE ENGINE

This chapter describes the basic difference between a four-stroke and a two-stroke engine. The motocross riders can race either with a motorcycle with a four-stroke engine or with a motorcycle with a two-stroke engine.

2.2.1 FOUR-STROKE ENGINE

The four-stroke engine is a reciprocating internal combustion engine and its total operating cycle consists of four separate strokes. During these 4 strokes, the crankshaft rotates twice and, based on this, one power stroke is produced. The illustrative four-stroke engine is shown below in the Fig. 7. The four separate strokes are called:

1. **An intake stroke**: This stroke begins with the piston at top dead center (TDC) and ends with the piston at bottom dead center (BDC). During this movement of the piston, a vacuum is created in the cylinder, the intake valve is opened, and the fresh air-fuel mixture is drawn into the cylinder.
2. **A compression stroke**: This stroke begins with the piston at bottom dead center and ends with the piston at top dead center. At this stage, the piston compresses the mixture in the cylinder and thus the pressure in the cylinder increases. Both valves (intake and exhaust) are closed. At the end of this stroke, combustion begins.
3. **A power stroke**: This stroke begins with the piston at top dead center and ends with the piston at bottom dead center. The compressed air-fuel mixture is ignited by a spark plug (in a gasoline engine) when the piston is at top dead center. The piston is pushed down due to high temperature and high-pressure gases which are generated during combustion. Because of this force, the crankshaft starts to rotate. At the end of this phase, the exhaust valve opens.
4. **An exhaust stroke**: This stroke begins with the piston at bottom dead center and ends with the piston at top dead center. The exhaust valve is opened, and the burned exhaust gases leave the cylinder. This happens for two reasons. Firstly, the piston moves toward top dead center and thus sweeps out the burned gases from the cylinder. Secondly, the pressure in the cylinder can be significantly

higher than the pressure in the exhaust. The inlet valve opens before top dead center, the exhaust valve closes after top dead center and the whole work cycle begins again. [18]

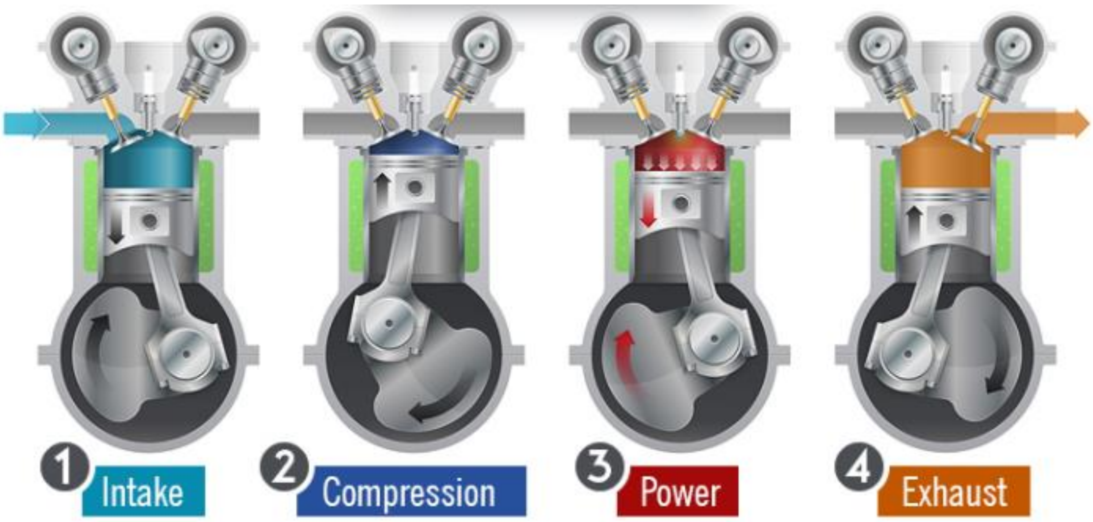


FIG. 7 FOUR-STROKE ENGINE [19]

2.2.2 TWO-STROKE ENGINE

A two-stroke engine is a reciprocating internal combustion engine which accomplishes a work cycle with two strokes of the piston. During this engine cycle, the one power stroke is completed in one crankshaft revolution. The two-stroke engine was created to gain a higher power output from a given engine size. The intake and exhaust ports open and close by the piston motion and the basic construction of the two-stroke engine is shown in the Fig. 8. These two strokes are named:

1. **A compression stroke**: This stroke starts when the intake and exhaust ports are closed by the piston. As the piston moves upwards, it compresses the mixture in the cylinder and at the same time the inlet port opens, through which fresh charge is drawn into the crankcase. When the piston gets closer to the top dead center, combustion starts.
2. **A power stroke**: The high-temperature and high-pressure gases produced during combustion push the piston down to the bottom dead center. When the piston moves down, the exhaust port opens first and then the transfer port opens. The burnt gases leave the cylinder through the exhaust due to blowdown

process. The fresh charge compressed in the crankcase flows into the cylinder through the transfer port and then it all starts again. The two-stroke engines are designed to deflect the fresh charge from the transfer port directly into the exhaust port. [18]

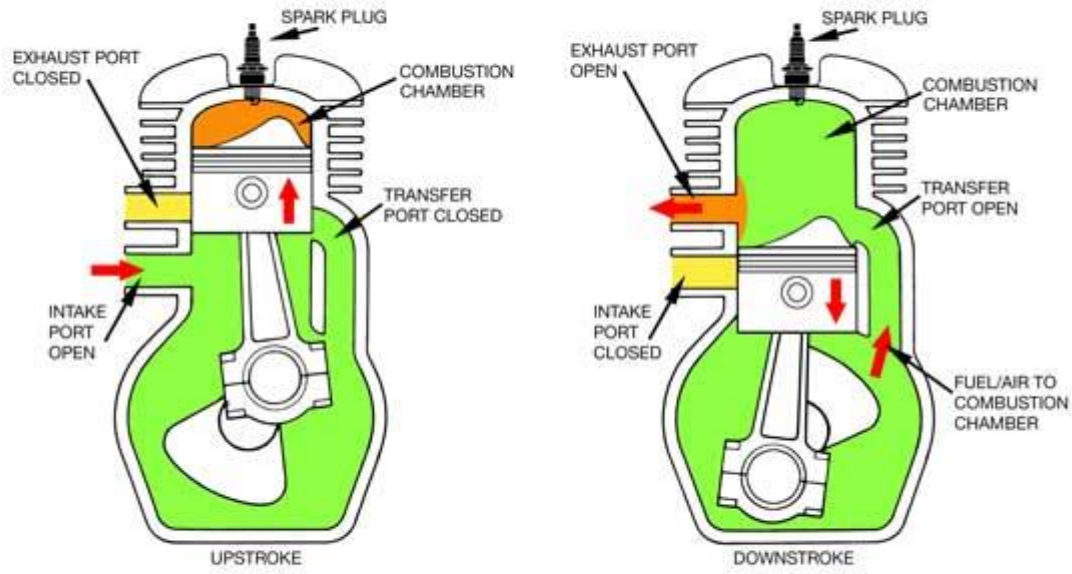


FIG. 8 TWO-STROKE ENGINE [20]

2.2.3 ADVANTAGES AND DISADVANTAGES OF FOUR STROKE OVER TWO STROKE CYCLE ENGINE

Advantages:

- A four stroke engines produce less pollution than two stroke engines because the four-stroke engine has separate strokes where the mixture is used more efficiently. (no mixing of unburned and burnt mixture). Another reason is that the fuel in two stroke engines contains an oil or a lubricant and the burnt oil (burnt lubricant) gets released through the exhaust into to air. Plus motocross bikes with two stroke engines also tend to smoke during startup.
- The four stroke engines produce more torque at a lower RPM and because the combustion takes place every two revolutions of the crankshaft, the output of power is more stable and more easily managed.

- An oil consumption of a four-stroke engine is lower because an oil is only needed for moving parts. Compared to the two stroke engines, there is no need to add oil to the fuel.
- The two stroke engines tend to wear out faster and therefore the four stroke engines are generally more durable.
- The four stroke engines have better fuel efficiency because fuel is consumed only once every four strokes and the incoming fuel does not have such a chance to get into the exhaust.
- Because of the way a four-stroke engine is designed, the four stroke engines have a wider range of speeds at which it can run effectively, and the rider does not have to shift so often.
- The four stroke engines do not require a maintenance as often as two stroke engines, because the two stroke engines have fewer parts which do more work and therefore each part wears out more. [21]

Disadvantages:

- The four stroke engines are less powerful because a power is delivered only once every two rotations of the crankshaft.
- An initial cost of a four-stroke motocross bike is more expensive because they are more complicated, and they have more parts. Maintenance is generally also more expensive, since there are a lot of parts used to keep the engine running and the maintenance is obviously more difficult.
- The four stroke engines are generally heavier and bigger because they contain more parts than two stroke engines and a flywheel of four stroke engine is usually heavier as well.
- Due to the complex construction, it is more difficult to clean a four-stroke engine than a two-stroke engine. [21]

2.3 BENCHMARKING OF CURRENT MOTOCROSS ENGINE MANUFACTURERS

In this section, several contemporary motorcycles that appear on the market are analyzed. The main parameters and components of each engine are shown below.

2.3.1 KTM 250 SX-F

KTM is an Austrian company and is one of the largest manufacturers of off-road motorcycles in the world. The motorcycle KTM 250 SX-F is the lightest motorcycle in its category and offers a great power delivery.

FIG. 9 MOTORCYCLE KTM 250 SX-F [22]

The motorcycle has the most powerful engine in the MX2 category and weighs only 25.9 kg. The engine is very compact and therefore, thanks to the centralization of weight, the motorcycle is well controlled. [22]

FIG. 10 ENGINE OF KTM 250 SX-F [22]

Engine Specifications:

Displacement	249.9ccm
Design	1-cylinder, 4-stroke engine, DOHC
Bore	78mm
Stroke	52.3mm
Compression ratio	14.4:1
Starter	Electric starter
Transmission	5-speed
Clutch	Multi-plate clutch, Brembo hydraulics
Cooling system	Liquid
Fuel delivery	Electronic fuel injection

TAB. 3 KTM 250 SX-F ENGINE SPECIFICATIONS [22]

Cylinder head

A part of the cylinder head in the KTM 250 SX-F motorcycle is a valve train in the form of a DOHC (double overhead camshaft). The first camshaft controls two titanium intake valves with a diameter of 32.5mm. The second camshaft controls two titanium exhaust valves with a diameter of 26.5mm. All valves are activated by finger followers, which are super-light and DLC (Diamond-like carbon) coated. The camshaft has an extremely smooth surface finish that minimizes friction. [22]

FIG. 11 CYLINDER HEAD OF KTM 250 SX-F [22]

Crankshaft

The KTM 250 SX-F motorcycle has a split crankshaft. A connecting rod is equipped with a plain bearing with two force-fitted bearing shells which are connected directly to the crank pin. This design is necessary for the higher engine speeds and increases the durability of the engine which is great for crankshaft service. [22]

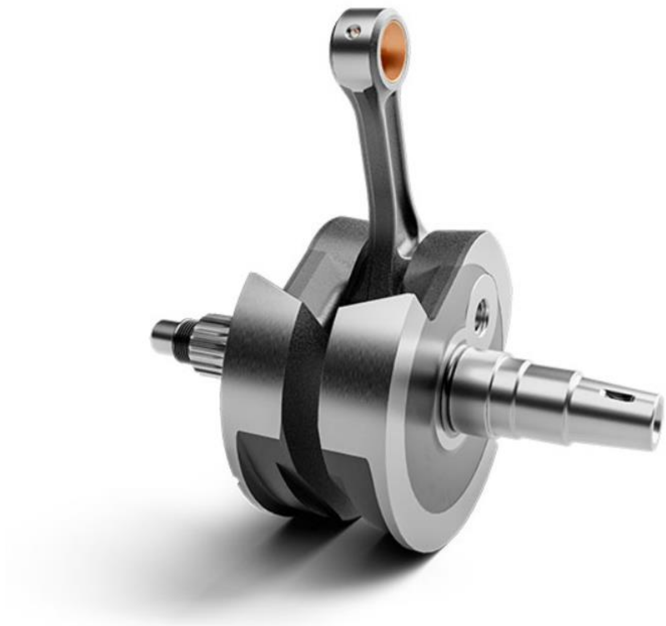


FIG. 12 CRANKSHAFT OF KTM 250 SX-F [22]

Cylinder and piston

In the cylinder with a bore of 78mm there is a lightweight, forged, bridged, box-type piston. The compression ratio is 14.4:1 and therefore there is a high-compression piston, which has a solid construction and has a unique crown shape. [22]

FIG. 14 CYLINDER OF KTM 250 SX-F [22]

FIG. 13 PISTON OF KTM 250 SX-F [23]

Transmission

The KTM 250 SX-F motorcycle has a 5 speed transmission. The gear ratios are the same as for older model and are shown in the table below. [22]

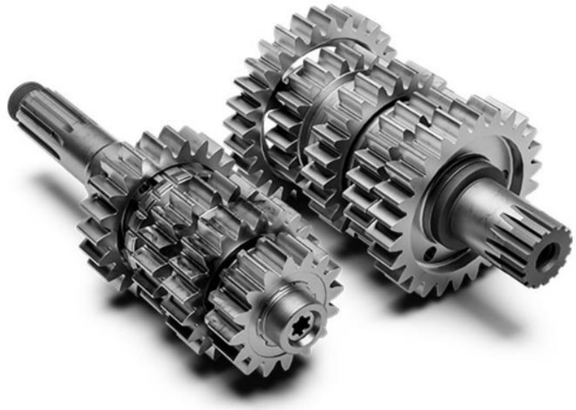


FIG. 15 TRANSMISSION OF KTM 250 SX-F [22]

Primary drive	24:73
1st gear	13:32
2nd gear	16:32
3rd gear	17:28
4th gear	19:26
5th gear	21:25
Final drive	14:51

TAB. 4 GEAR RATIOS OF KTM 250 SX-F [22]

Clutch

The KTM 250 SX-F motorcycle is equipped with a DS (diaphragm steel) clutch. The clutch consists of a solid, one-piece clutch basket and primary gear. It is made of high-strength steel. Furthermore, the clutch is equipped with extreme heat-resistant steel carrier friction disc, which are pretensioned by a diaphragm spring. Because of this design and the Brembo hydraulic system, the most efficient clutch disengagement is ensured. [22]

FIG. 16 CLUTCH OF KTM 250 SX-F [22]

Balancer shaft

The engine of the KTM 250 SX-F motorcycle is equipped with a multifunctional counter balancer shaft to balance the mass forces. This balancer shaft reduces vibrations, drives the water pump and timing chain. [22]

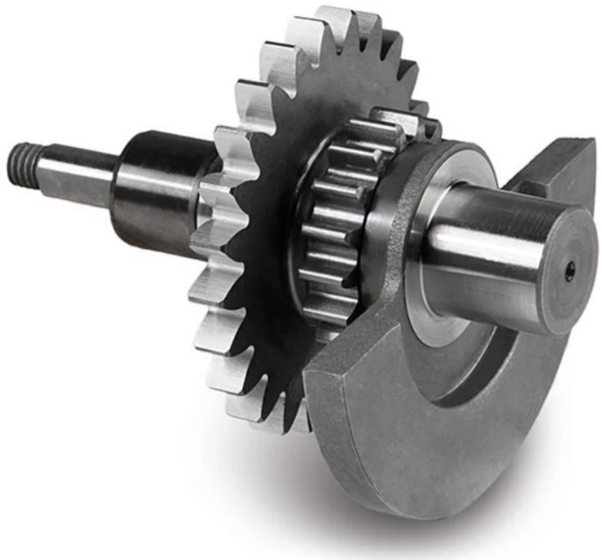


FIG. 17 BALANCER SHAFT OF KTM 250 SX-F [22]

E-Starter

The engine of KTM 250 SX-F uses an electric starter from a Japanese company Mitsuba. Because time is one of the most important factors in motocross racing, it is essential that the starter is reliable and efficient. [22]

FIG. 18 E-STARTER OF KTM 250 SX-F [22]

Engine management system

An engine control is provided by an engine management system from Keihin Corporation. It includes an electric fuel injection with a 44mm throttle body. The position of the injector ensures ideal vaporization of the air and fuel. There is also a traction control system, a launch control, a system for cold starting or a system for idle adjustment. The handlebars are also equipped with a map select switch that allows the driver to select a power map (standard - advanced). [22]

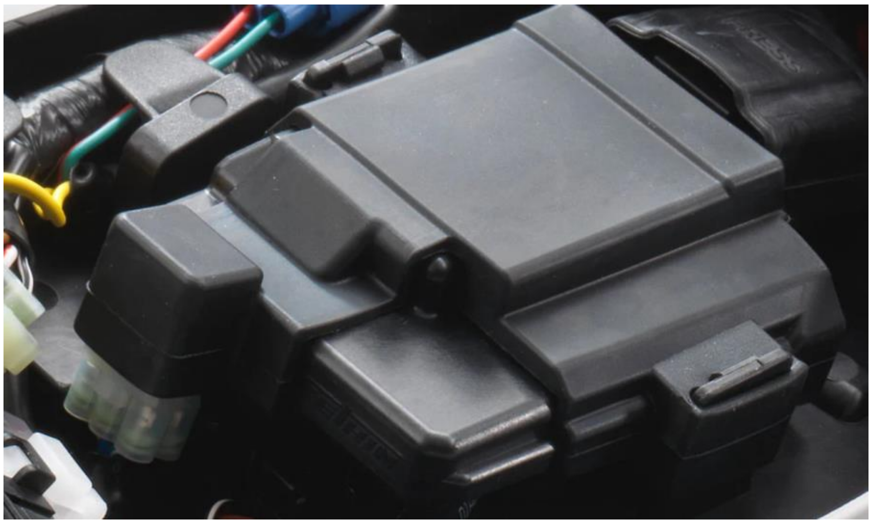


FIG. 19 ENGINE MANAGEMENT SYSTEM OF KTM 250 SX-F [22]

Engine Layout

The axis of the cylinder of the KTM 250 SX-F motorcycle is tilted forward. From the pictures below it can be seen that the exhaust is led forward and the intake is led from behind. The engine has a conventional cylinder, which means that the cylinder offset is equal to 0mm. The engine does not have a kick start lever and the electric starter is located between the gearbox and the intake device. The timing chain is located on the right side of the engine and is driven via the crankshaft and the balancer shaft. The chain guide is provided by a plastic timing chain guide rail, a plastic timing chain tensioning rail and a timing chain tensioner. The cooling system consists of a water pump, hoses, and two radiators. The pump is located in the right engine cover and is driven by a balancer shaft. The water is distributed from the radiators to the crankcase, from where it is pushed by a water pump into the cylinder, the head and back into the radiators. The lubrication system consists of two trochoid pumps, the first one is called a force pump and the second one is called a suction pump. The same oil is used to lubricate the gearbox and engine. Both trochoid pumps are on the same shaft and driven via the clutch by means of gearing. The force pump pumps the oil through the oil filter to the crankshaft, conrod, cylinder head and sprays it onto the piston and alternator. From there, it is sucked into the gearbox by a suction pump and then the cycle is repeated. [22]

FIG. 20 ENGINE KTM 250 SX-F INSIDE LEFT [24]

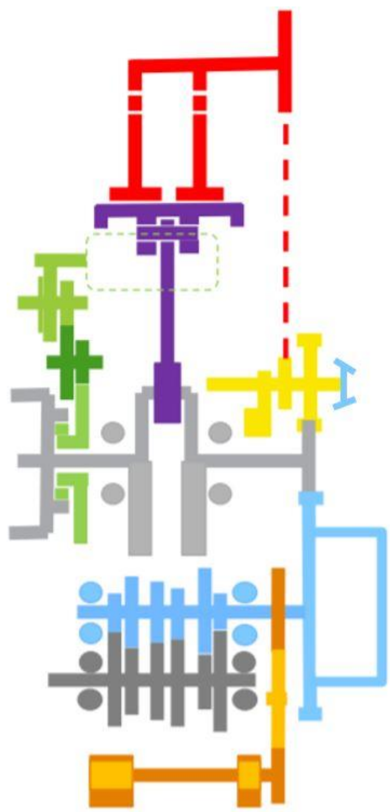


FIG. 21 ENGINE LAYOUT KTM 250 SX-F [25]

2.3.2 YAMAHA YZ250F

Yamaha Corporation is a Japanese corporation that is the world's largest manufacturer of pianos and, among other things, also manufactures motorcycles. The engine of the Yamaha YZ250F motorcycle is the most powerful series-produced four-stroke power unit, which the Yamaha corporation has ever produced, with a displacement of 250ccm.

FIG. 22 MOTORCYCLE YAMAHA YZ250F [26]

The Yamaha YZ250F motorcycle has a liquid-cooled engine with a DOHC (double overhead cam) valvetrain. For 2019, the engine is equipped with a new, high-compression piston, and the cylinder is also lighter. The axis of the cylinder is tilted more vertical to improve centralization of mass. [26]

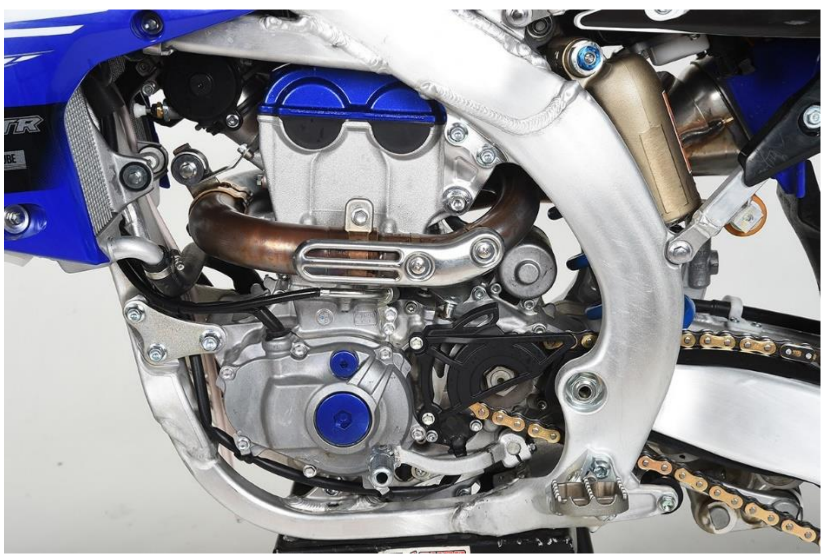


FIG. 23 ENGINE OF YAMAHA YZ250F [27]

Engine Specifications:

Displacement	250ccm
Design	1-cylinder, 4-stroke engine, DOHC
Bore	77mm
Stroke	53.6mm
Compression ratio	13.8:1
Starter	Electric starter
Transmission	5-speed
Clutch	Multi-plate clutch
Cooling system	Liquid
Fuel delivery	Mikuni electronic fuel injection

TAB. 5 YAMAHA YZ250F ENGINE SPECIFICATION [26]

Cylinder head

The valve train of the Yamaha YZ250F motorcycle is in the form of a DOHC (double overhead camshaft), which is the same as in the KTM 250 SX-F motorcycle. There are two intake valves with a diameter of 31mm which are controlled by a camshaft. On the other side there are two exhaust valves with a diameter of 25mm which are controlled by another camshaft. All valves are made of titanium, because titanium valves are about 40 % lighter than steel valves. The Yamaha YZ250F motorcycle does not use the finger followers design as a KTM 250 SX-F but uses valve-actuation buckets. The diameter of the exhaust lifter buckets is 24.5 mm and the diameter of the intake lifter bucket is 27mm. [28]

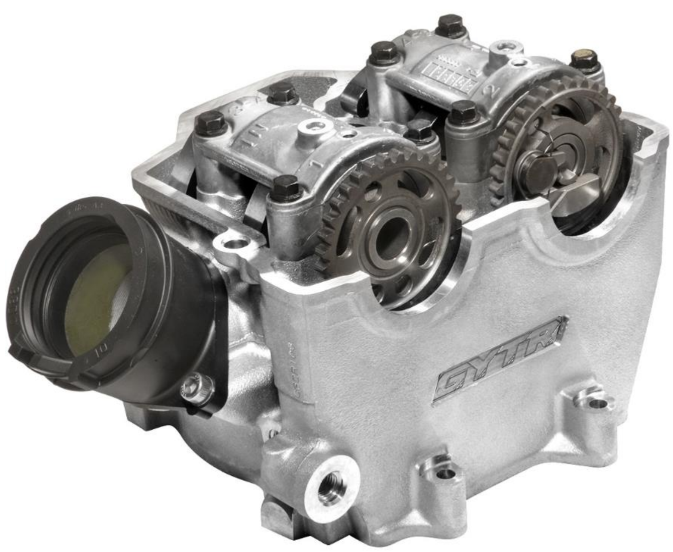


FIG. 24 CYLINDER HEAD OF YAMAHA YZ250F [29]

Crankshaft

The crankshaft of the Yamaha YZ250F motorcycle is also split, and is designed similarly to the KTM 250 SX-F motorcycle. The crankshaft is mounted on ball bearings and is heat-treated for great strength and durability. Part of the crankshaft is an undivided nickel-chromium-molybdenum connecting rod. [28]

FIG. 25 CRANKSHAFT OF YAMAHA YZ250F [30]

Cylinder and piston

The engine of Yamaha YZ250F is equipped with a lightweight forged aluminum piston with a bridge-box design. The piston has only two rings (a compression ring and oil control ring) which reduce a sliding resistance. The piston is connected to the connecting rod by means of a DLC (diamond like coating) piston pin, which reduces internal friction. The engine of Yamaha YZ250F does not have a liner, so there is only a lightweight aluminum cylinder. The cylinder features a ceramic composite coating which reduces friction and improves a thin layer of oil between the piston and the cylinder. [28]

FIG. 26 PISTON OF YAMAHA YZ250F [31]

E-Starter

The Yamaha YZ250F motorcycle no longer utilizes a kick starter, but uses an electric starter to reduce weight and minimize restart delays during the race. The electric starter drives the clutch basket and is located behind the cylinder. It is powered by a four-cell lithium-ion battery with following parameters: 2.4Ah/13.2volt. The weight of the battery is only 713.5g. [32]

Transmission

The transmission of the Yamaha YZ250F motorcycle has 5 speed just like a KTM 250 SX-F motorcycle. The whole shift mechanism (a selector drum, gears, dogs, shift shaft, etc.) is designed for smooth gear changes and for great durability. [26]

Clutch

The design of the clutch on the Yamaha YZ250F motorcycle differs from the clutch on the KTM 250 SX-F motorcycle. It uses an aluminum housing, which is more common and lighter, but requires more space, thus increasing the width of the engine. The diameter of the wet multi-disc clutch plate is 144mm, which is 7mm more than the previous model. The clutch includes six clutch springs and eight friction plates. The thickness of the steel plate is 1.6 mm. [32]

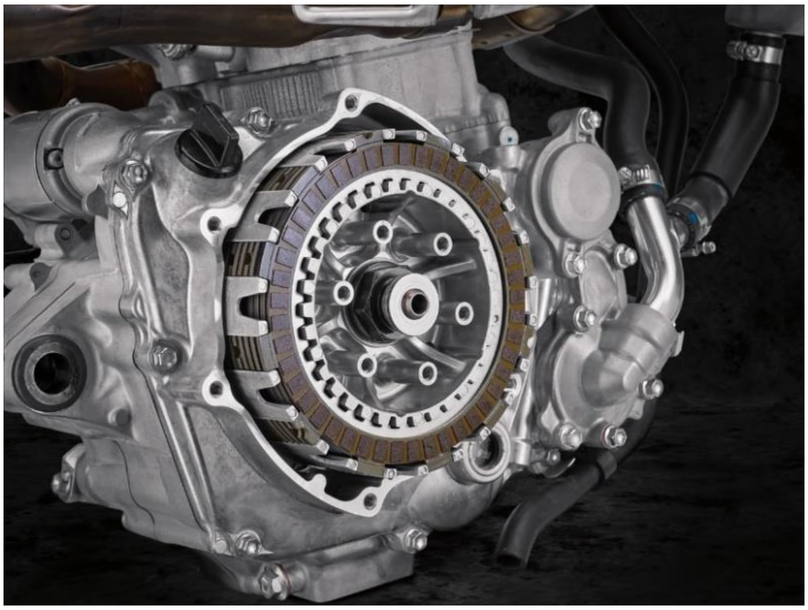


FIG. 27 CLUTCH OF YAMAHA YZ250F [33]

Engine management system

The Yamaha YZ250F motorcycle used to use a throttle body from Keihin Corporation (as well as the KTM 250 SX-F motorcycle), but now it uses a throttle body from Mikuni Corporation with a diameter of 44mm. The throttle body from Mikuni Corporation is 12 percent lighter than the previous unit. The injection system also consists of a high-pressure electric fuel pump, and a new Denso® injector with twelve holes. The ignition system features a dual-electrode spark plug. The rider of the Yamaha YZ250F can choose from two-mode engine maps by pushing a button on the handlebars. The rider can change an engine character depending on the weather or change of track. The Yamaha company came up with a new Power Tuner application that allows to adjust the ignition timing, and the air/fuel mixture based on track conditions using the iOS® or Android® device via Wi-Fi. [26]

Engine Layout

The axis of the cylinder is tilted backwards but compared to the previous model it is tilted forwards by 1 degree to improve mass-centralization. The cylinder is thus tilted backwards by 5.2 degrees. The engine has a cylinder offset of 4mm towards the exhaust to reduce mechanical friction against the cylinder wall. The intake of the air and injection system are located at the front of the engine and provide a direct intake path. The fuel is cooler because it is not greatly affected by the heat of the engine. The flow rate of the air is increased due to a smoother intake port shape. The engine is liquid cooled. The liquid is driven by a water pump into two radiators, which are larger and more angled to the incoming air flow compared to the previous model. The water pump is located on the right side of the engine. The exhaust is led from behind the cylinder and the exhaust port is straighter to allow free outflow of exhaust gasses. The exhaust has a wraparound construction which means that the exhaust circles the cylinder to achieve the optimal length for the pulse effect, which results in improved power output and improved mass concentration. The exhaust system includes a new lightweight, aluminum, re-buildable muffler. The chain that drives both camshafts in the cylinder head is located on the left side of the engine and, is driven by gearing on the crankshaft. A part of the engine is also a counter balancer shaft, which has the task of reducing vibrations. As with KTM 250 SX-F motorcycle, the same oil is used to lubricate the engine and gearbox. [28] [32]

FIG. 28 ENGINE YAMAHA YZ250F LEFT VIEW [33]

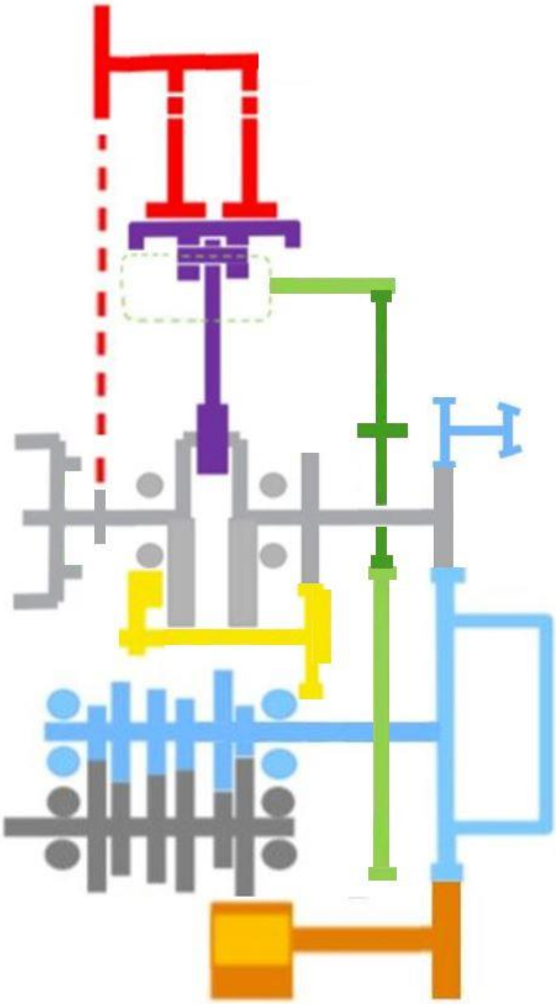


FIG. 29 ENGINE LAYOUT YAMAHA YZ250F

2.3.3 HONDA CRF250R

Honda motor company is a corporation that mainly produces cars, motorcycles, and power units. The company is based in Japan and is the largest manufacturer of motorcycles in the world. A motocross motorcycle CRF250R was introduced in 2004 and nowadays it is one of the most powerful motorcycles in its category (MX2 class).

FIG. 30 MOTORCYCLE HONDA CRF250R [34]

The Honda CRF250R motorcycle has a liquid-cooled four-stroke engine with a DOHC (double overhead cam) valvetrain, although it was equipped with a unicam single overhead camshaft (SOHC) for several years. At the first sight, we can see that it differs from previous motorcycles (KTM 250 SX-F and Yamaha YZ250F) by containing a dual exhaust system. Specific engine parameters are shown in the table below. [34]

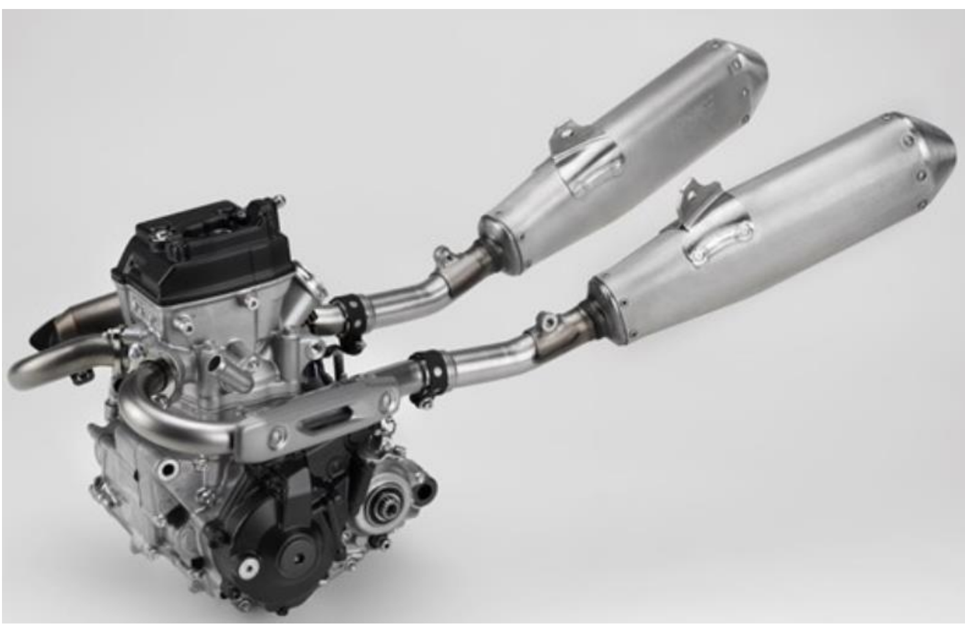


FIG. 31 ENGINE OF HONDA CRF250R [35]

Engine Specifications:

Displacement	249.4ccm
Design	1-cylinder, 4-stroke engine, DOHC
Bore	79mm
Stroke	50.9mm
Compression ratio	13.9:1
Starter	Electric starter
Transmission	5-speed
Clutch	Multi-plate clutch
Cooling system	Liquid
Fuel delivery	Electronic fuel injection

TAB. 6 HONDA CRF250R ENGINE SPECIFICATION [34]

Cylinder head

The Honda CRF250R motorcycle uses a DOHC (double-overhead-cam) cylinder head design. The valvetrain includes ovalized valve springs and finger follower rockers that activate the valves. The finger follower rockers are DLC (Diamond-like carbon) coated. There are two intake valves with a diameter of 33mm, and two exhaust valves with a diameter of 26mm. The lift of the valve reaches 10.5mm for intake and 9.5mm for exhaust. The valve angle is 20.5 degrees. The intake channels are short and almost straight. [35]

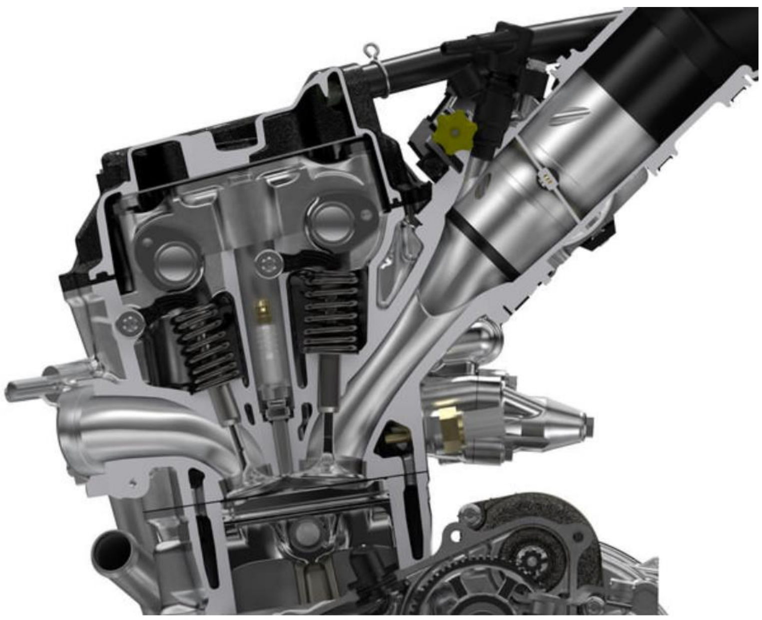


FIG. 32 CYLINDER HEAD OF HONDA CRF250R [35]

Crankshaft

The engine of Honda CRF250R is also equipped with a split crankshaft. The crankshaft is mounted on roller bearings and the undivided connecting rod is connected to the crank by means of a crank pin. Although the crankshaft is 350 grams lighter than the previous unit, a rigidity and an internal mass have been retained. The design of the crankshaft is shown in the Fig. 33 below. [36]

FIG. 33 CRANKSHAFT OF HONDA CRF250R [36]

Cylinder and piston

The Honda CRF250R motorcycle features a bridged-box piston design just like the two previous motorcycles (KTM 250 SX-F and Yamaha YZ250F). The compression ratio increased from 13.8:1 to 13.9:1 compared to the previous Honda model. A bore is increased to 79mm and the stroke is decreased to 50.9mm. Due to the larger bore and shorter stroke, there is a higher rev limit. The engine uses a 5-hole piston oil jet (instead of the previous 4-hole design) to improve piston cooling and reduce knocking. [34]

FIG. 34 PISTON AND PISTON PIN OF HONDA CRF250R [37]

FIG. 35 PISTON OF HONDA CRF250R [38]

Transmission

The engine of Honda CRF250R features a constant-mesh 5-speed transmission as well as previous motorcycles (KTM 250 SX-F and Yamaha YZ250F). The transmission includes a front sprocket with 13 teeth and a rear sprocket with 48 teeth. To keep the engine speed when shifting from second to third gear, the second gear ratio is closer to the third and the gap between the two gears is also closer. A gear position sensor is added which allows the use of three specific engine maps. [35] [39]

Clutch

The engine of Honda CRF250R includes a multi-plate wet clutch with five coil springs and eight friction plates. The plates use two different friction materials instead of one and because of that the clutch can operate with a greater power without being bigger. The rate of the clutch springs has been increased and thus the power also increased compared to the previous model. [39] [40]

FIG. 36 CLUTCH OF HONDA CRF250R [36]

E-Starter

The Honda CRF250R motorcycle is equipped with an electric starter as well as KTM 250 SX-F and Yamaha YZ250F. Although the kick starter was removed, the addition of an electric starter increased the weight of the engine by 1 kilogram. The electric starter is powered by a compact, lightweight Lithium-ion battery that weighs 0.65kg. The electric start is suitable for start/restart in cold conditions because the lithium-ion battery can withstand temperatures of minus 10 degrees Celsius. [38]

Engine management system

The inlet valves are fed by a programmed fuel injection system (PGM-FI) with a throttle body from Keihin Corporation. The throttle body has a diameter of 44mm, which is 2mm less than the previous body, which improves the flow of intake air at low speed. The Honda CRF250R motorcycle uses the same throttle body with the same diameter as KTM 250 SX-F. The fuel injector sprays twice per intake cycle, which provides a better fuel atomization. The motorcycle is equipped with an HRC launch control system that helps the rider to achieve the fastest start. The rider pushes the button located on the left handlebar and selects one of the following three modes:

- Level 3 – 8,250rpm, muddy conditions/beginner
- Level 2 – 8,500rpm, dry conditions/basic
- Level 1 – 9,500rpm, dry conditions/expert.

The engine management system also includes an engine mode select button (EMSB) that can change engine characteristics. Depending on the track conditions, the driver can choose from three maps (Standard, Smooth, Aggressive) that are activated by pressing the button on the left side of the handlebars. The ignition system is fully transistorized. [35]

Engine Layout

The cylinder of the engine of Honda CRF250R is tilted forward, as we can see in the Fig. 30. The exhaust system is led from the front of the cylinder and consists of two exhausts. The injection system and intake of the air are therefore located behind the cylinder. The engine has a cylinder offset of 4.5mm towards the exhaust to reduce mechanical friction against the cylinder wall. The cam chain is located on the right side of the engine, and is driven by the crankshaft and drives both camshafts in the cylinder head. To reduce vibrations, the engine is equipped with a balancer shaft, which is also driven by a crankshaft. The engine is liquid cooled. The liquid is distributed by a water pump located on the right side of the engine and driven by a balancer shaft. There is a shared oil lubrication system for the gearbox and the engine. The combination of these two systems reduces weight of the engine and helps increase the compactness of the engine. The multi-plate wet clutch is located on the right side, and the alternator, which is driven by the crankshaft, is located on the left side of the engine. [34] [35]

FIG. 37 ENGINE HONDA CRF250R INSIDE LEFT [35]

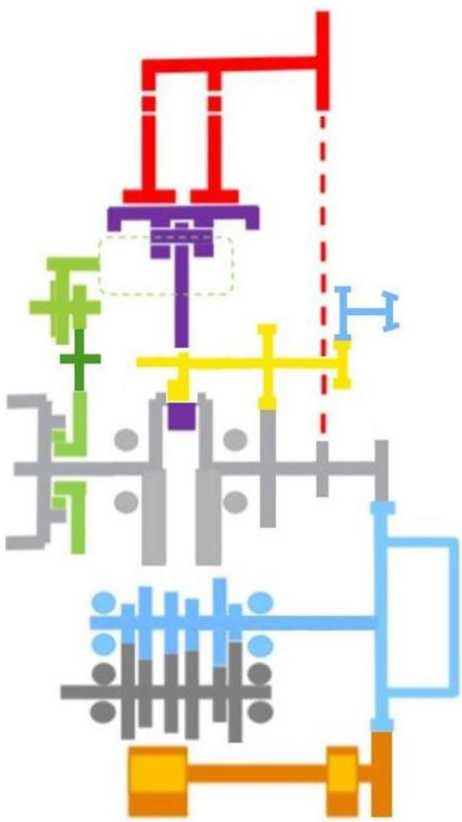
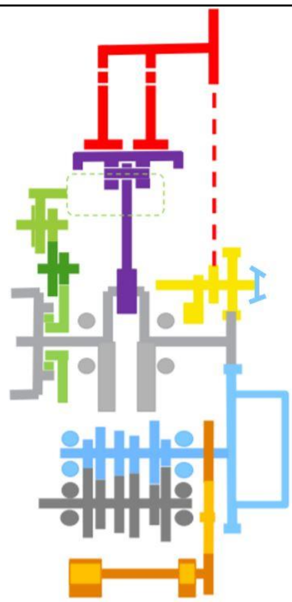
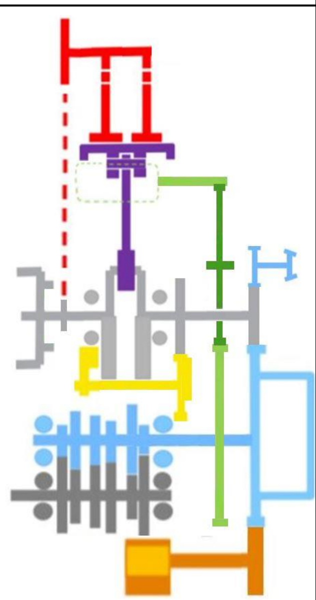
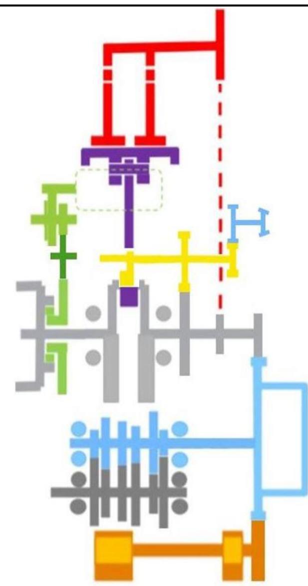


FIG. 38 ENGINE LAYOUT HONDA CRF250R

2.4 COMPARISON OF MOTOCROSS ENGINES

	KTM 250 SX-F	Yamaha YZ250F	Honda CRF250R
Displacement	249.9ccm	250ccm	249.4ccm
Design	1-cyl, 4-stroke, DOHC	1-cyl, 4-stroke, DOHC	1-cyl, 4-stroke, DOHC
Bore	78mm	77mm	79mm
Stroke	52.3mm	53.6mm	50.9mm
Comp. ratio	14.4:1	13.8:1	13.9:1
Cam chain	Right side	Left side	Right side
Intake	Behind	Forward	Behind
Exhaust	Forward	Behind	Forward
Piston	Bridge-box design	Bridge-box design	Bridge-box design
Comp. rings	1	1	1
Oil rings	1	1	1
Intake valves	32.5mm	31mm	33mm
Exhaust valves	26.5mm	25mm	26mm
Valve actuation	Finger followers	Buckets	Finger followers
Layout			

TAB. 7 COMPARISON OF MOTOCROSS ENGINES

Based on the benchmarking in the previous chapter, I compiled a clear table where the basic parameters of all three motorcycles are compared. We can see that the values of displacement, bore, and stroke are very similar. The KTM motorcycle has a slightly higher compression ratio than the other two. The design of the pistons is the same for all, including the number of rings. The diameters of the exhaust and intake valves are also very similar. The difference occurs in the valve actuation, where the motorcycles of KTM and Honda use finger followers, and the motorcycle from Yamaha uses buckets. The simplified schemes of the layout of components in the engines are shown at the end of the table, where we can see that the layout differs for all three engines.

In 2019, the Dirt Rider magazine compared six motocross bikes on a dynamometer. The compared motorcycles include: 2019 Yamaha YZ250F, 2019 Suzuki RM-Z250, 2019 KTM 250 SX-F, 2019 Kawasaki KX250, 2019 Husqvarna FC 250 and 2019 Honda CRF250R. The resulting characteristics of powers and torques are shown in the Fig. 39 below. [41]

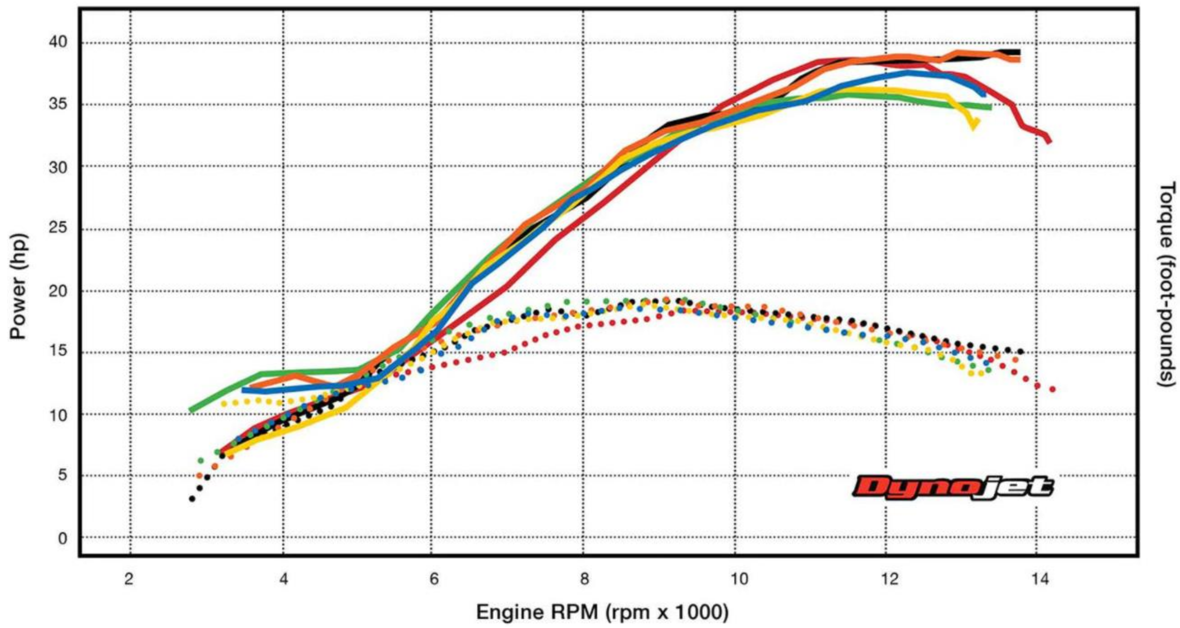


FIG. 39 COMPARISON OF POWER AND TORQUE CHARACTERISTICS [41]

When evaluating the graph, I focused only on the motorcycles that I compared in the previous chapter. The graph shows that the highest values of power and torque occur in the KTM motorcycle. The values of the highest powers and torques are shown in the Tab. 8 below.

	Max power	Max torque
• 2019 Honda CRF250R	38.4hp(28.6kW) at 11730rpm	18.3 lb-ft(24.8N.m) at 9760rpm
• 2019 KTM 250 SX-F	39.2hp(29.2kW) at 13210rpm	19.1 lb-ft(25.9N.m) at 8900rpm
• 2019 Yamaha YZ250F	37.5hp(27.9kW) at 12390rpm	18.5 lb-ft(25.1N.m) at 8440rpm

TAB. 8 COMPARISON OF MAX POWER AND MAX TORQUE [41]

3 THE PRACTICAL PART

3.1 THE TECHNICAL PARAMETERS OF THE DESIGNED ENGINE

3.1.1 BORE, STROKE, AND ENGINE DISPLACEMENT

At first, we need to design the bore and stroke from which we get the engine displacement. These parameters define the basic geometry of a designed engine. Based on the theoretical part, we know that motocross engines have a bore greater than the stroke, which means that the ratio of the cylinder bore to piston stroke will be greater than 1.

Ratio of cylinder bore to piston stroke:

$$R_{bs} = \frac{B}{L}\ [-] \tag{1}$$

where: B – bore

L – stroke.

For motocross engines, it is better to use an engine with a higher ratio of cylinder bore to piston stroke than one, because the engine can reach higher speeds, the friction losses are smaller, and the height of the engine can be reduced. I chose a bore with a diameter of 79mm and a stroke with a length of 51mm. In my case the ratio of cylinder bore to piston stroke is equal to:

$$R_{bs} = \frac{B}{L} = \frac{79}{51} \cong 1.55 \tag{2}$$

The displacement of the engine (swept) volume is calculated from the following equation:

$$V_d = \frac{\pi \times B^2}{4} \times L = \frac{\pi \times 79^2}{4} \times 51 = 249\ 985.17\ mm^3 = 249.958\ cm^3 \tag{3}$$

The displacement of the engine is therefore less than 250cm^3 and thus the volume condition is met. The designed parameters are shown in the Tab. 9 below.

Bore	79mm
Stroke	51mm
Engine displacement	249.958cm^3

TAB. 9 BORE, STROKE, ENGINE DISPLACEMENT

3.1.2 COMPRESSION RATIO

Another important parameter is the compression ratio. The compression ratio is defined as the ratio between the maximum cylinder volume and the minimum cylinder volume. The compression ratio is therefore equal to:

$$r_c = \frac{V_d+V_c}{V_c}\ [-] \tag{4}$$

where: V_d – engine displacement or swept volume

V_c – clearance volume.

The compression ratio mainly affects the thermal efficiency. The larger the compression ratio, the greater the efficiency. However, the larger the ratio, the greater the engine wear. Since in my case it is a gasoline engine, the compression ratio is also limited by knocking. To calculate the compression ratio, you need to know the clearance volume. To determine the clearance volume, it is necessary to design a complete combustion chamber (piston, cylinder, cylinder head, valves, etc.). I dealt with these designs in Chapter 2.3. From the resulting model in Creo Parametric which showed that the clearance volume is equal to 19,300.78 mm^3. After substituting into the formula (4) above, it proves that the compression ratio is equal to:

$$r_c = \frac{V_d+V_c}{V_c} = \frac{249{,}958+19{,}300.78}{19{,}300.78} = 13.95\ [-] \tag{5}$$

3.1.3 KINEMATICS OF THE CRANK MECHANISM

I used the diagram below to calculate the kinematics quantities of the crank mechanism. First, I had to design the length of the connecting rod *l* and the length of the crank radius *r*.

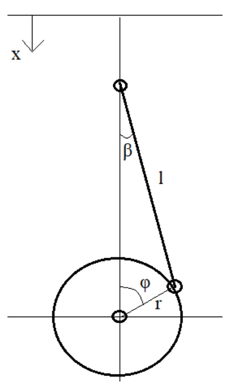


FIG. 40 DIAGRAM OF THE CRANK MECHANISM

The designed parameters are shown in the following table:

Length of connecting rod l	92mm
Length of crank radius r	25.5mm

TAB. 10 CONNECTING ROD AND CRANK RADIUS

Piston position

I calculated the distance of the piston from TDC from the equation:

$$x = r + l - r \times cos\varphi - l \times cos\beta \text{ [mm]} \tag{6}$$

$$\sin\beta = \lambda_s \times sin\varphi \text{ [-]} \tag{7}$$

$$\lambda_s = \frac{r}{l} \text{ [-]} \tag{8}$$

and then there is the final equation:

$$x = r \times (l - \cos\varphi + \frac{\lambda_s}{2} \times sin^2\varphi) \text{ [mm]} \tag{9}$$

where: φ – crank angle

λ_s – con rod ratio.

The calculation was performed in the MATLAB software, when the crank angle was set in the range of 0 to 360 degrees and in Fig. 41, we can see a graphical result.

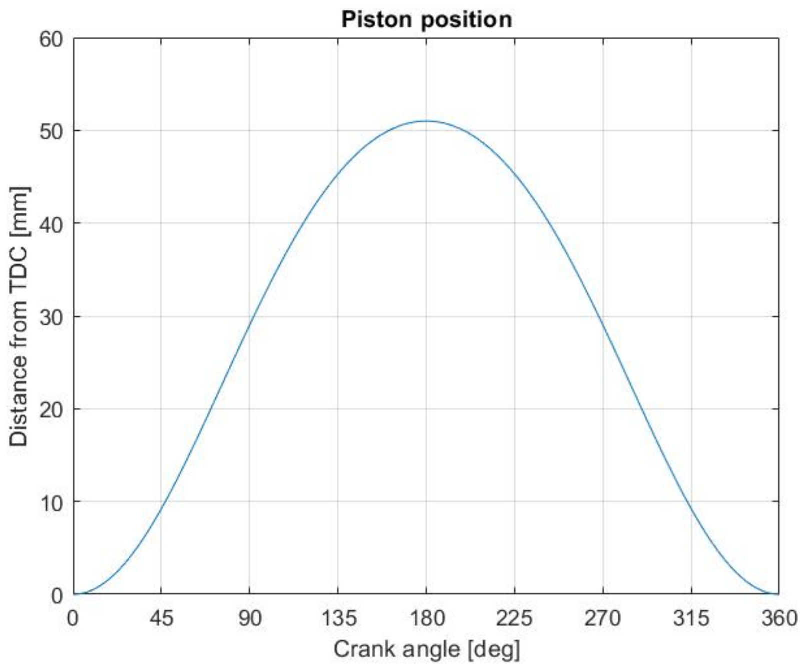


FIG. 41 PISTON POSITION

The greatest distance from the TDC is 51mm, which is of course the total stroke of the engine.

Piston velocity

The velocity of the piston is obtained from the derivative of its path, so I derive the equation (9) according to time. First, we must express the crank position using the following equation:

$$\varphi = \omega \times \tau \text{ [rad]} \tag{10}$$

$$\omega = 2 \times \pi \times n \text{ [rad.s}^{-1}\text{]} \tag{11}$$

where: ω – angular velocity

τ - time

n – engine speed (crank rotational speed)

I chose a maximum engine speed of 13,000 rpm and the resulting piston velocity is shown at that rotations. The calculation was performed in the MATLAB software according to the following modified equation:

$$v_p = \frac{dx}{dt} = r \times \omega \times (sin\varphi + \frac{\lambda_s}{2} \times \sin(2\varphi)) \text{ [m/s]} \tag{12}$$

and the resulting graphical representation of the piston velocity is shown in the Fig. 42.

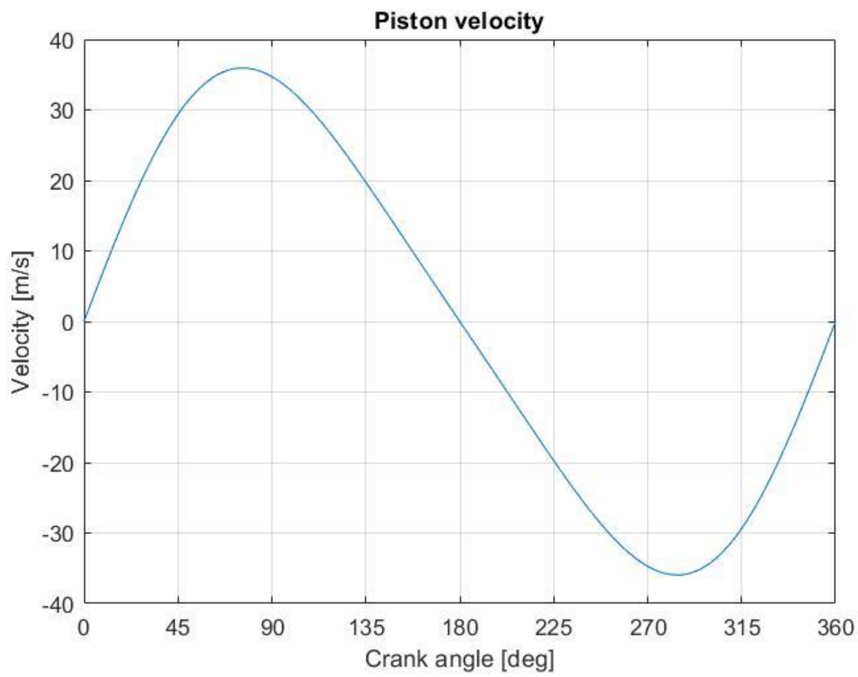


FIG. 42 PISTON VELOCITY

The graph shows that the maximum piston velocity is about 36 m/s.

Mean piston speed

A more suitable parameter for engine comparison and engine behavior is a mean piston speed. A piston speed affects friction work, inertia forces, engine noise and wear, etc. The mean piston speed is calculated according to the equation:

$$v_m = 2 \times L \times n = 2 \times 0.051 \times \frac{13000}{60} = 22.1\, m/s \tag{13}$$

The mean piston speed for 13000 revolutions per minutes is therefore equal to 22.1 m/s.

Piston acceleration

The acceleration of the piston is obtained from the derivative of its velocity, so I derive the equation (12) according to time. The calculation was performed in the MATLAB software according to the following modified equation:

$$a_p = \frac{dv_p}{dt} = r \times \omega^2 \times (cos\varphi + \lambda_s \times \cos(2\varphi))\ [\mathrm{m/s^2}] \tag{14}$$

The resulting graphical representation is shown in the Fig. 43, where we can see that the acceleration of the piston reaches high values.

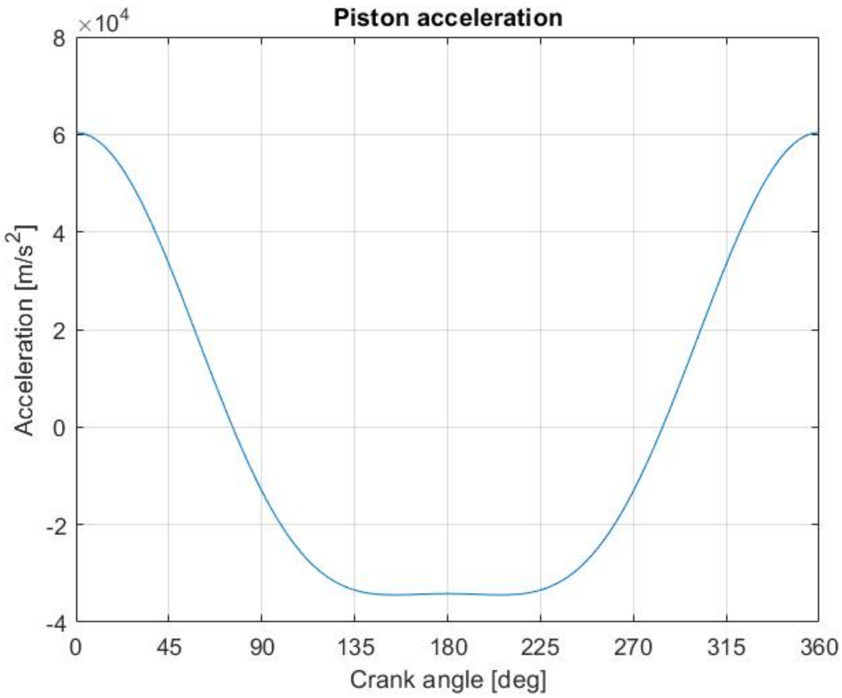


FIG. 43 PISTON ACCELERATION

3.2 THERMODYNAMIC MODEL

A thermodynamic model of the designed engine was compiled in the GT-SUITE program, which I could use on a university computer using a remote desktop. GT-SUITE is a simulation tool that can be used for fast conceptual design, detailed analysis, optimization, etc. The diagram of my thermodynamic model is shown in the Fig. 44.

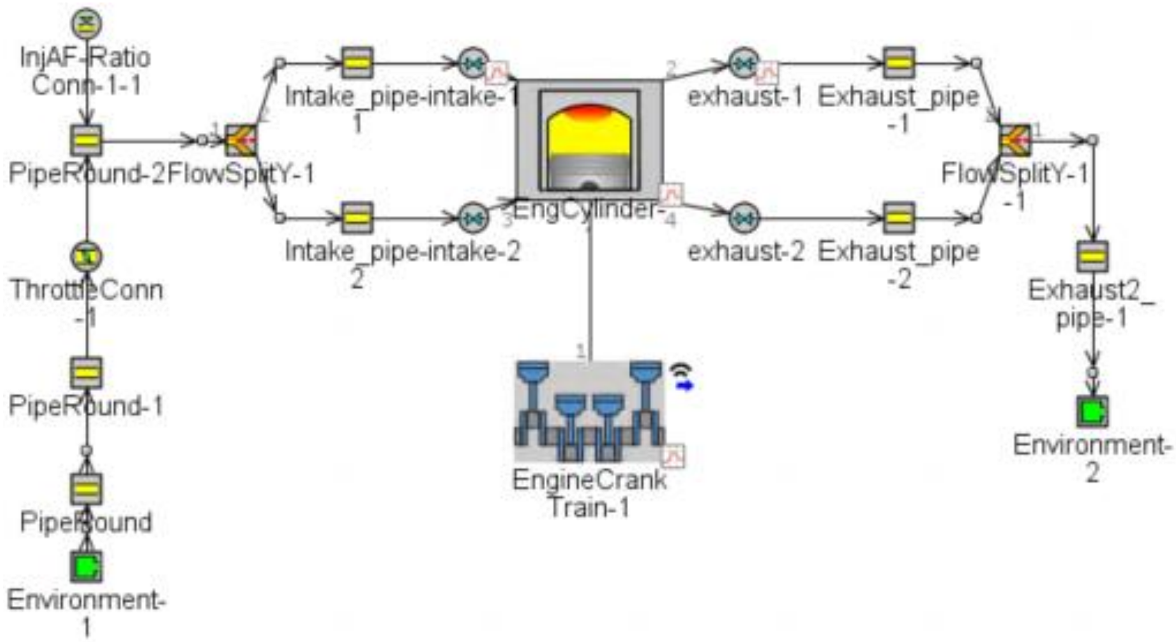


FIG. 44 DIAGRAM OF THERMODYNAMIC MODEL

The inlet air temperature is 25 degrees Celsius (298 K) and the air pressure is 1 bar. It is therefore atmospheric pressure, as the use of supercharging or turbocharging is prohibited for motocross engines. An intake system is simplified to a system of plastic pipes. The model is equipped with a throttle with a diameter of 44mm and the simulation takes place only when the throttle is fully open (full load). There is also an injector where I set the air–fuel equivalence ratio λ to 0.9, which means it will be a slightly rich mixture. The injected fluid temperature is 27 degrees Celsius (300 K). Furthermore, the intake system is divided into two branches. This is because the designed engine includes a DOHC (double overhead camshaft) valvetrain with two intake and two exhaust valves. The diameter of the intake valve is set to 32mm and its maximum lift is 10.8mm. The diameter of the exhaust valve is set to 26,5mm and its maximum lift is 9.8mm. The dependence of the intake and exhaust valve lift on the crank angle was provided to me by Ricardo. The company also provided me with a flow coefficient for intake and exhaust ports. In the Engine Crank Train section, it was necessary to set that it is a four-stroke engine. Furthermore, it was necessary to set the geometry of the cylinder. The cylinder has a bore with a diameter of 79mm and a stroke with a diameter of 51mm. The length of the connecting rod is 92mm and the compression ratio has been set to 13.95. All these parameters correspond to the design carried out in chapter 2.1. The combustion rate in the cylinder is based on the Wiebe function and the heat transfer is based on the Woschni model. Behind the exhaust valves are two branches led back to one, and the exhaust system is simplified to a system of cast-iron pipes. The maximum speed was set at 13,000rpm.

The GT-SUITE program can calculate countless quantities and can draw dozens of graphs, and in my case, I will show only selected ones.

The Fig. 45 shows the dependence of the brake torque on the engine speed. The simulation took place with the fully open throttle (full load). As we can see, the greatest torque occurs at 9 000 rpm and has a value of 26.95 N.m.

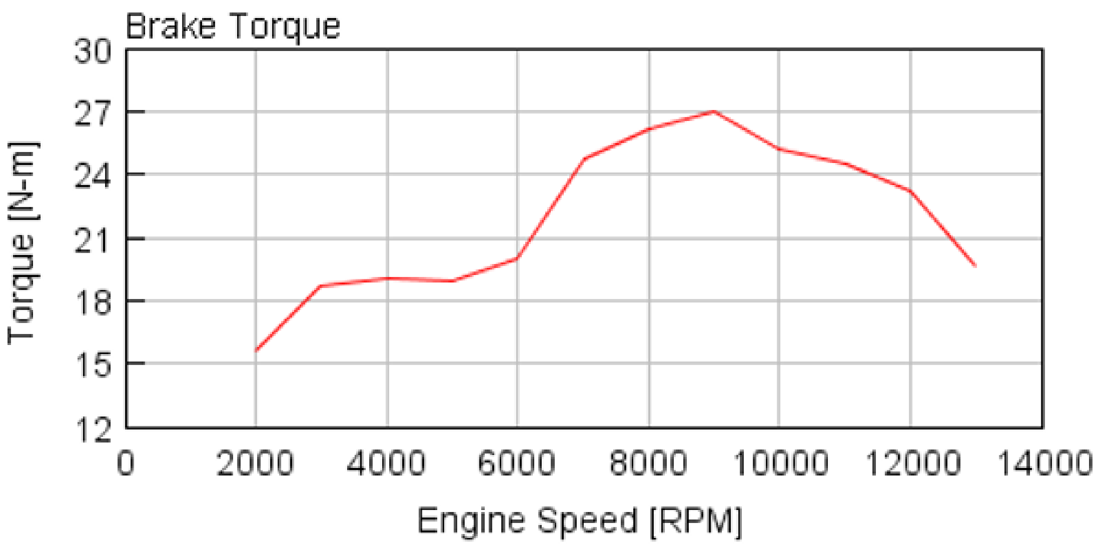


FIG. 45 ENGINE BRAKE TORQUE CHARACTERISTICS

Another important characteristic is an engine brake power characteristic. The dependence of the brake power on engine speed can be seen in the Fig. 46. The simulation took place with the fully open throttle (full load) as well. The engine has a maximum brake power of 29.1kW, which occurs at 12,000rpm. If we compare these two characteristics with the Fig. 39 where there is a comparison of power and torque characteristics of contemporary motocross engines, then we can see that the values of brake power and brake torque are quite similar.

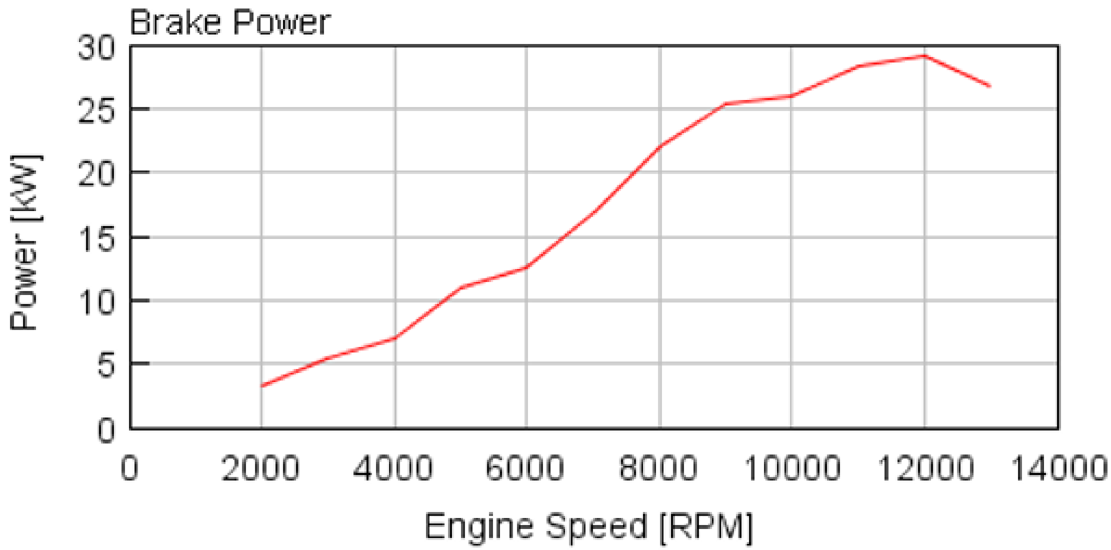


FIG. 46 ENGINE BRAKE POWER CHARACTERISTICS

The Fig. 47 shows the dependence of the cylinder pressure on the crank angle. The simulation was performed at 12,000rpm, which is the engine speed when the engine has the greatest power. The highest pressure in the cylinder occurs at 13.2 degrees after TDC (top dead center) during a power stroke. The greatest pressure value is 82.7 bars (8.27 MPa).

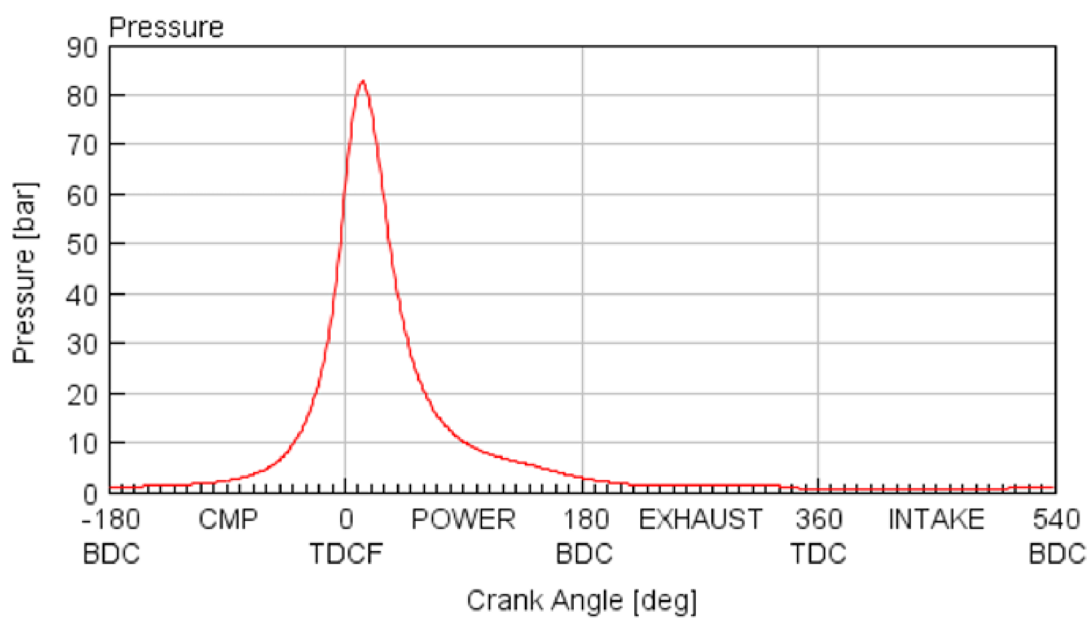


FIG. 47 CYLINDER PRESSURE VERSUS CRANK ANGLE

The dependence of cylinder temperature on crank angle was also simulated at 12,000rpm. The resulting graph is shown in the Fig. 48 below. The highest temperature occurs at 24.4 degrees after TDC during a power stroke and has a value of 2,609 Kelvins (2,339 degrees Celsius).

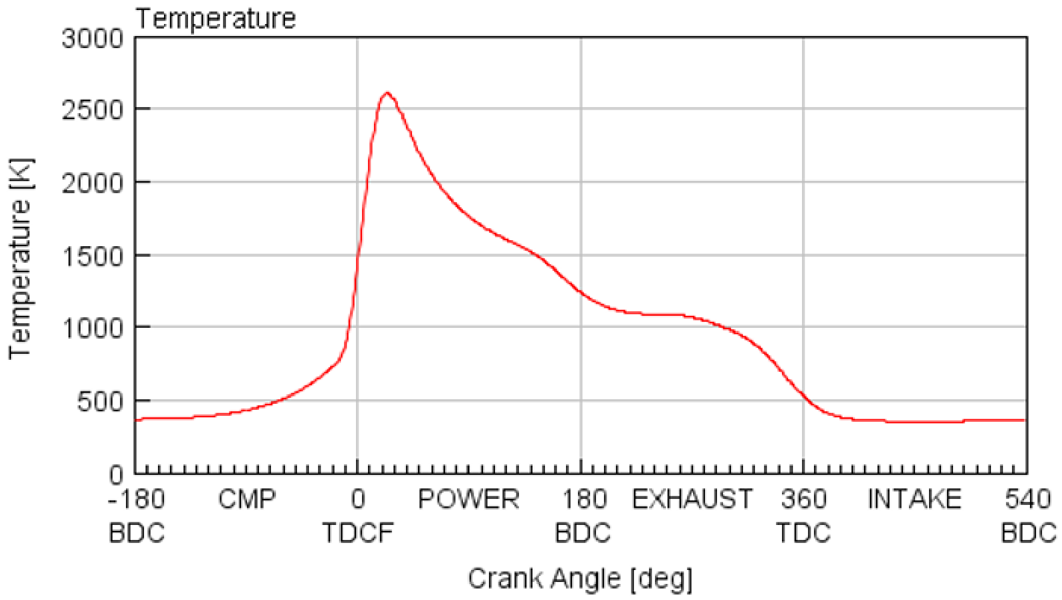


FIG. 48 CYLINDER TEMPERATURE VERSUS CRANK ANGLE

3.3 CONSTRUCTION DESIGN

The entire construction design was made in PTC Creo Parametric 3.0 on the computer in Ricardo Prague s.r.o.

3.3.1 ENGINE CONCEPT

When designing the engine concept, I decided to follow the engine layout from the KTM company. The axis of the cylinder is thus tilted forwards, specifically by 12.5 degrees. The intake system is led from the rear of the engine and the exhaust from the front. The cylinder offset is equal to 0mm, which means that the engine has a conventional cylinder. The timing chain is located on the right side of the engine and is driven via the crankshaft and the balancer shaft. The engine has a DOHC (double overhead camshaft) valvetrain with two intake and two exhaust valves, which are controlled by finger followers. The advantages and disadvantages of this concept are evaluated in Chapter 3.5.

3.3.2 CRANKSHAFT MECHANISM

3.3.2.1 PISTON

The piston must be as rigid as possible to withstand mechanical and thermal stresses, and at the same time as light as possible due to inertial forces. For this reason, a durable and lightweight material needs to be used. Pistons are generally made by forging from aluminum alloy. The aluminum has good heat conduction and can therefore be well cooled with oil. In my case, aluminum alloy AlSi12.5MgCuNi (Al4032) is chosen as the construction material. To prevent collisions of the piston with the valves, the top of the piston is milled for the heads of the valves. The milling is at the same angle as the inclination of the valves. There are two grooves for the piston rings on the wall of the pistons (upper for the compression ring and lower for the oil ring). There are several holes in the groove for the oil ring, which serve to drain the oil into the piston (red arrow in the Fig. 49). The bottom of the piston is provided with a bridge-box design. This design is shown in the green box in the Fig. 50, and its main advantage is to improve durability and reduce weight. At the bottom of the piston there is also a hole for the piston pin, in which there is a milling to lubricate the pin (blue arrow in the Fig. 50).

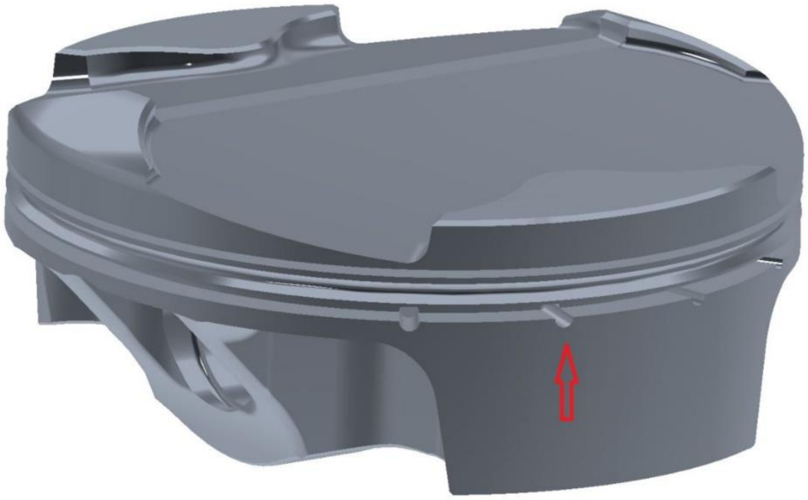


FIG. 49 PISTON

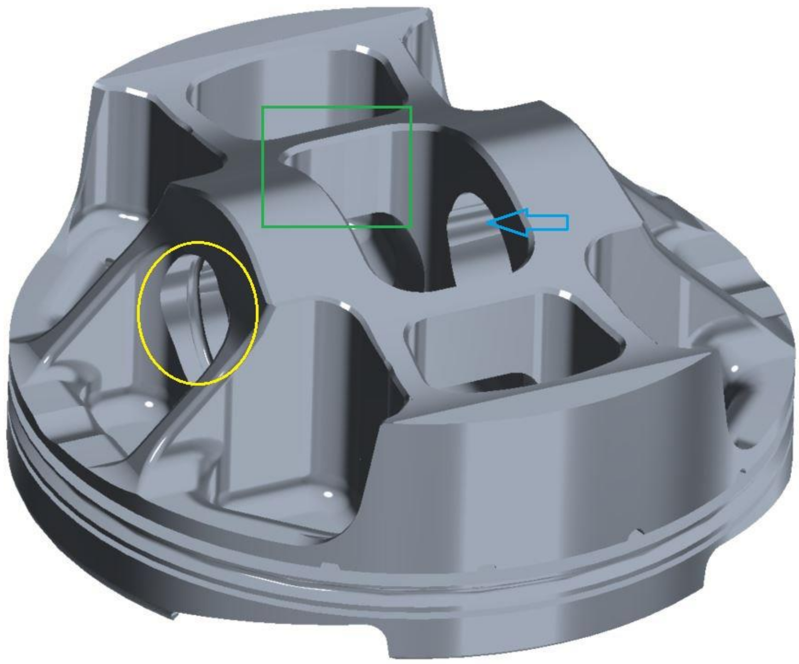


FIG. 50 BOTTOM OF THE PISTON

3.3.2.2 PISTON PIN

A piston pin provides the connection between the piston and the connecting rod. I designed a hollow pin with an outer diameter of 16mm and an inner diameter of 10mm. The piston pin is shown in the Fig. 51 where we can see that it is chamfered on both ends. The steel grade DIN 42CrMo4 was chosen as the material, which is an alloy steel for quenching and tempering. The piston surface is hardened and polished. An axial displacement of the piston pin is prevented by two wire lock rings. The wire lock rings have a diameter of 1mm and are inserted in a groove in the piston, which can be seen in the Fig. 50 in a yellow circle.

FIG. 51 PISTON PIN

3.3.2.3 PISTON RINGS

As mentioned above, in my case, the engine has only one compression ring and one oil ring to minimize the amount of friction between the rings and the engine cylinder.

The compression ring is mounted with very little axial and radial lash. The width of the ring is 0.7mm, the cross-section is 2.8mm and the groove in the piston is 0.78mm wide. It is a rectangular ring according to the ISO 6621-1 standard with a standard lock shape. The compression ring is made of cast iron plated with chromium and the resulting ring design is shown in the Fig. 52.

The oil ring regulates the thickness of the oil layer on the cylinder wall so that the contact surface of the cylinder and the piston are lubricated as to the fullest extent, and the oil consumption is as small as possible. In my case, there is an oil ring with cut-outs with a screw profile. The width of the oil ring is 1.5mm, and it is made of the same material as the compression ring. Around the perimeter of the ring there are holes with a diameter of 0.3mm through which oil flows. A model of the oil ring is shown below in the Fig. 53.

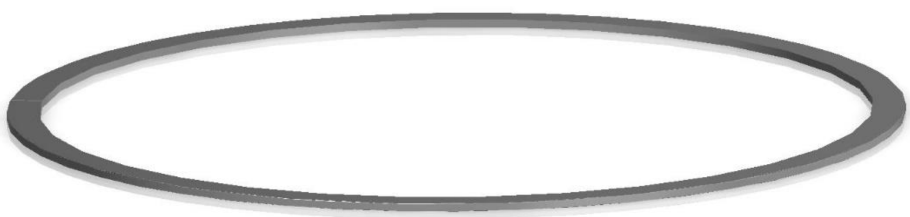


FIG. 52 COMPRESSION RING

FIG. 53 OIL RING

3.3.2.4 CONNECTING ROD

The connecting rod is undivided, as is customary with most motocross engines. The connecting rod should be light so as not to generate large inertial forces, and at the same time must be sufficiently rigid to withstand the pressure and inertial force from the piston. A steel DIN 34CrNiMo6 which is commonly used for connecting rods, was chosen as the material.

FIG. 54 CONNECTING ROD

The length of the connecting rod is 92mm, which is the distance between the centers of both ends. The diameter of the hole for the crank pin (big-end) is 34mm and the diameter of the hole for the piston pin (small-end) is 17.6mm. The width of the big-end is 18mm and the width of the small-end is 15mm. There are two small holes with a diameter of 2mm in the small-end to lubricate the piston pin. Through these holes the oil enters from the spray from the bottom of the piston into the bearing of the piston pin. The model of the connecting rod is shown in the Fig. 54, where the blue arrow points to the holes for lubricating the piston pin.

3.3.2.5 CRANK PIN

The crank pin has a diameter of 30mm, and its length is 56mm. The crank pin is hollow for two reasons. The first reason is to save weight and the second reason is to lubricate the connecting rod bearing. Because of lubrication, there are two holes in the crank pin, which can be seen in the Fig. 55 in a red and green circle. The hole in the green circle with a diameter of 2.25mm supplies oil from the right crankshaft web and the hole in the red circle with a diameter of 4mm supplies oil to the connecting rod bearing. There are aluminum covers at the end of the crank pin. The inner diameter of the pin is 8.5mm. The selected material structural steel is DIN 42CrMo4 (ČSN 41 5142), which is used to produce machine parts with high toughness, such as shafts, gears, etc. The designed crank pin is shown in the Fig. 55

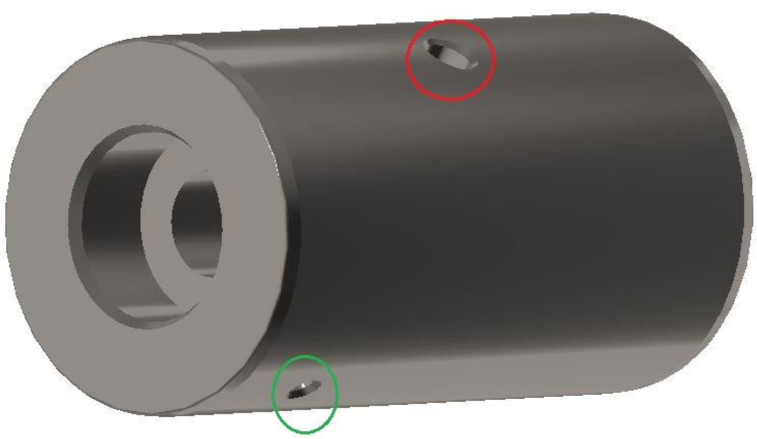


FIG. 55 CRANK PIN

3.3.2.6 CRANKSHAFT

The crankshaft is split. This means that it consists of two parts (a left crankshaft web and right crankshaft web), which are connected by a pressed crank pin. The crankshaft is made of DIN 42CrMo4 steel and is surface hardened. Both crankshafts are equipped with weights for balancing, which is described in chapter 3.3.3.1.

LEFT CRANKSHAFT WEB

The design of the left crankshaft web is shown in the Fig. 56. The inner side of the left crankshaft web is provided with a center hole ISO 6411-A 4/8.5, which is shown in a green circle. The blue arrow points to a diameter of 30mm where the cylindrical roller bearing is located. Furthermore, a thread M27x1.5 is machined on the crankshaft due to a nut which prevents axial movement of the bearing (a black arrow in the Fig. 56). The red arrow in the Fig. 56 indicates the diameter 25mm where the freewheel gear will be located, which will lead through the next gear to the electric starter. At the end of the shaft, which is chamfered, there is an alternator which is connected to the shaft by means of a Woodruff key (a purple arrow in the Fig. 56).

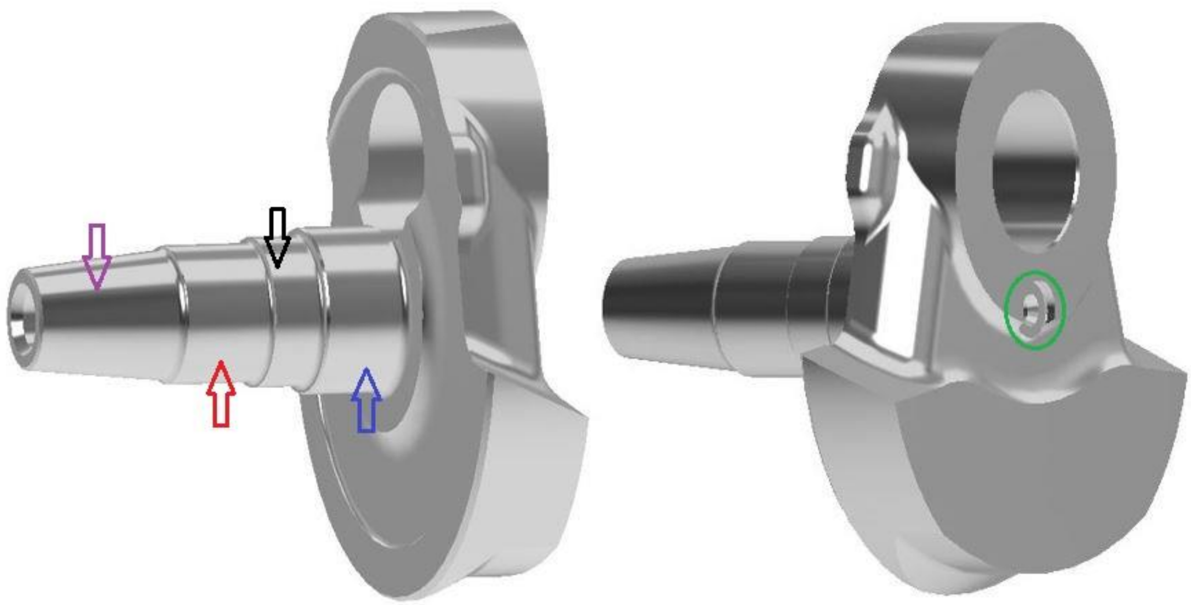


FIG. 56 LEFT CRANKSHAFT WEB

RIGHT CRANKSHAFT WEB

The inner side of the right crankshaft web is equipped with a center hole ISO 6411-A 4/8.5, as well as the left crankshaft web. The model of the right crankshaft web is shown in the Fig. 57, where the center hole is shown in a green circle. On the shaft of the diameter 30mm, indicated by the blue arrow in the Fig. 57, the same cylindrical roller bearing is located as on the left crankshaft web. The designed cylindrical roller bearings are described in the chapter 3.3.2.7. An axial displacement of the bearing is prevented by a steel spacer ring located at a diameter of 25mm (a purple arrow in the Fig. 57). Further on the right crankshaft web there is an involute spline 25x1.2x9g ČSN 014952 on which the primary gear is mounted (a red arrow in the Fig. 57). A static control of the involute spline is described below. The black arrow in the Fig. 57 points to a thread M18x1 on which a nut is fitted which prevents axial displacement of the bearing, spacer ring, and the primary gear.

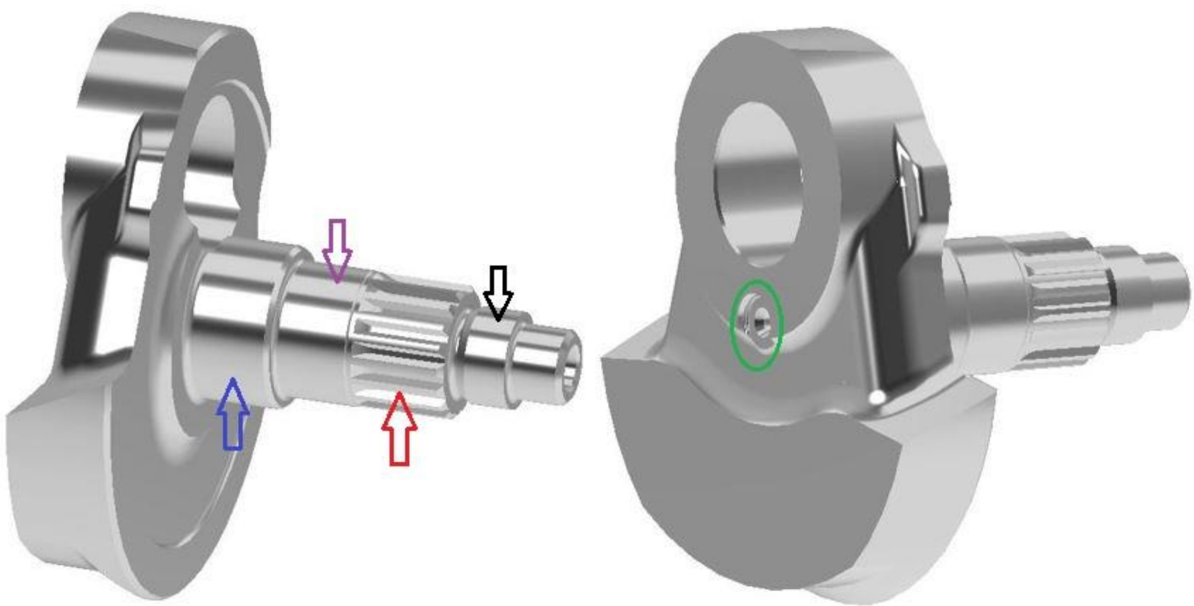


FIG. 57 RIGHT CRANKSHAFT WEB

STATIC CONTROL OF INVOLUTE SPLINE

The basic parameters of the involute spline 25x1.2x9g ČSN 014952 are shown in the Tab. 11.

m [mm]	1.25
z [-]	18
l [mm]	11
h [mm]	2.8
ψ [-]	0.85
M_k [N.m]	110

TAB. 11 PARAMETERS OF THE INVOLUTE SPLINE

Where: m – module

z – number of teeth

l – length of spline

h - height of tooth

M_k – torque

The static control is performed for the torque of 110 Nm, which was provided to me by the Ricardo from *Engdyne* analysis. The involute spline is loaded by bending stress, by shear stress and by pressure. The bending stress is calculated from equation (15), where the resulting value must be less than the allowable bending stress, which in this case is 650 MPa. The shear stress is calculated from equation (16), where the resulting value must be less than the allowable shear stress, which in this case is 390 MPa. And the pressure acting on the tooth is calculated from equation (17), where the resulting value must be less than the allowable pressure, which in this case is 200 MPa.

$$\sigma = \frac{12 \cdot M_k \cdot h}{0.5 \cdot m \cdot l \cdot z^2 \cdot (2.17 \cdot m)^2} = 225.5\ MPa \tag{15}$$

$$\tau = \frac{3 \cdot M_k}{0.5 \cdot m \cdot l \cdot z^2 \cdot 2.17 \cdot m} = 54.6\ MPa \tag{16}$$

$$p = \frac{2 \cdot M_k}{0.5 \cdot m^2 \cdot \psi \cdot z^2 \cdot l} = 92.9\ MPa \tag{17}$$

The results show that none of the calculated values exceeded the allowable limits of the material, and therefore the involute spline is suitable.

3.3.2.7 CYLINDRICAL ROLLER BEARINGS

The crankshaft is mounted in two single row, cylindrical roller bearings. Specifically, it is the NJ 206 ECP bearing from SKF Company. I was inspired by the KTM 250 SX-F motorcycle, which uses the same bearings. The technical specification of this bearing can be found on the SKF website. The model of the NJ 206 ECP bearing is shown in the Fig. 58.

FIG. 58 NJ 206 ECP CYLINDRICAL ROLLER BEARING [42]

3.3.2.8 PRIMARY GEAR

The primary gear is mounted on the right web by involute spline 25x1.2x9H ČSN 014952. The primary gear drives the balancer shaft and the clutch by means of gearing. It is made of low alloyed case-hardened manganese – chromium steel 20MnCr5 (DIN 1.7147). The basic designed parameters of the gear are shown in the Table 12. The model of the primary gear is shown in the Fig. 59.

z_1 [-]	24
m_{n1} [mm]	2
α_{n1} [°]	20
b_{w1} [mm]	9.5
β_1 [°]	0

TAB. 12 PARAMETERS OF PRIMARY GEAR

Where: z_1 – number of teeth of primary gear

m_{n1} – module of primary gear

α_{n1} – normal pressure angle of primary gear

b_{w1} – face width of primary gear

β_1 – helix angle of primary gear

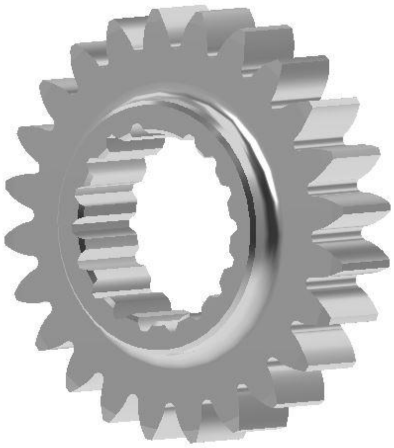


FIG. 59 PRIMARY GEAR

3.3.2.9 CRANKSHAFT MECHANISM ASSEMBLY

The resulting model of the crank mechanism assembly is shown in the Fig. 60. A nut M27 can be seen on the left crankshaft web, which prevents axial displacement of the cylindrical roller bearing. On the right crankshaft web there is a steel spacer ring with an inner diameter of 25mm and a nut M8, which prevents axial displacement of the cylindrical roller bearing, spacer ring, and primary gear.

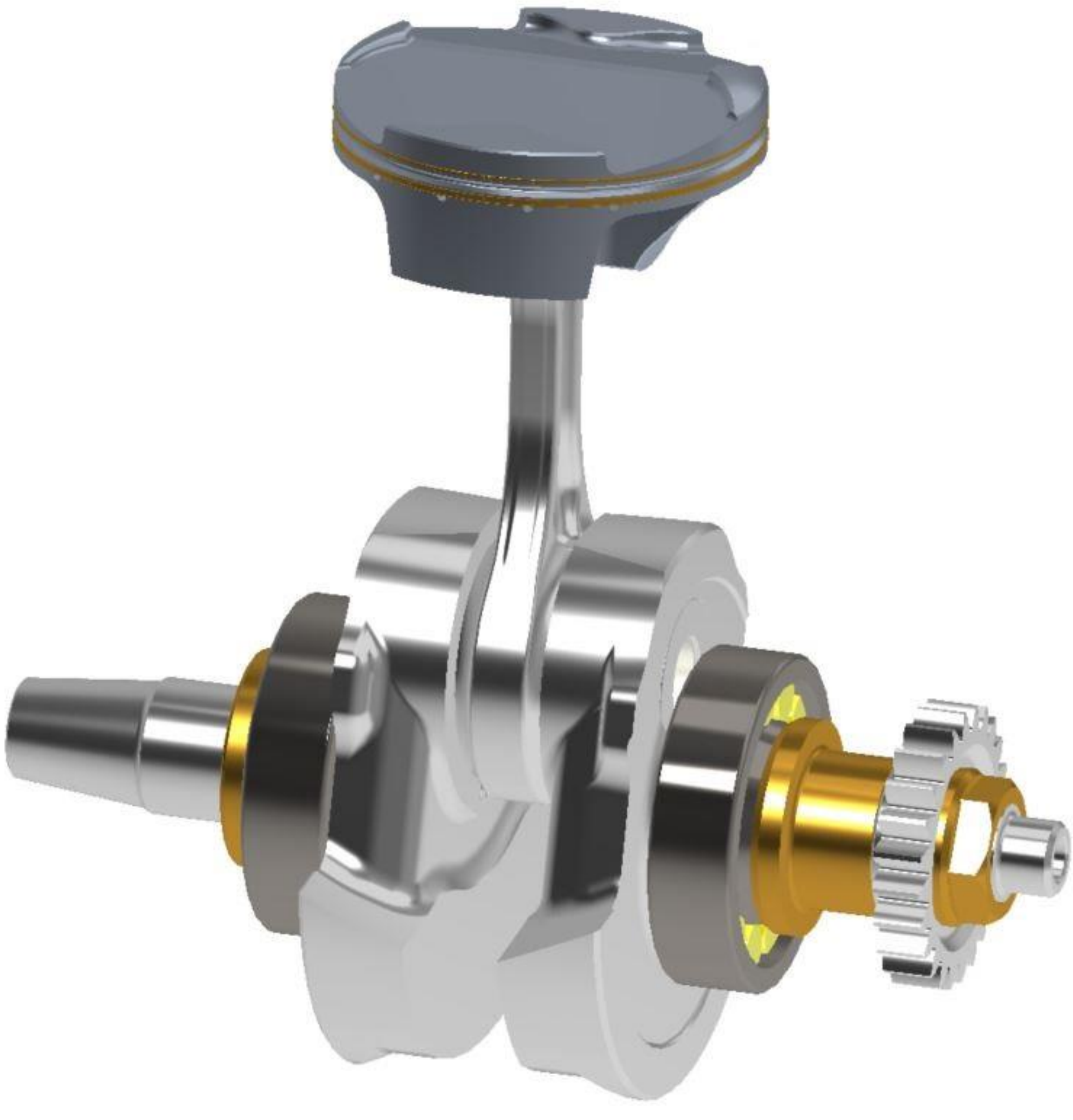


FIG. 60 CRANK MECHANISM ASSEMBLY

The Fig. 61 is a sectional view of the entire crank mechanism. Two wire lock rings are visible at the top, which prevent axial displacement of the piston pin. In the right crankshaft web, you can see a hole through which oil is fed into the crankpin, from where it leads further to the connecting rod bearing. The connecting rod is mounted at the top and bottom in bronze plain bearings.

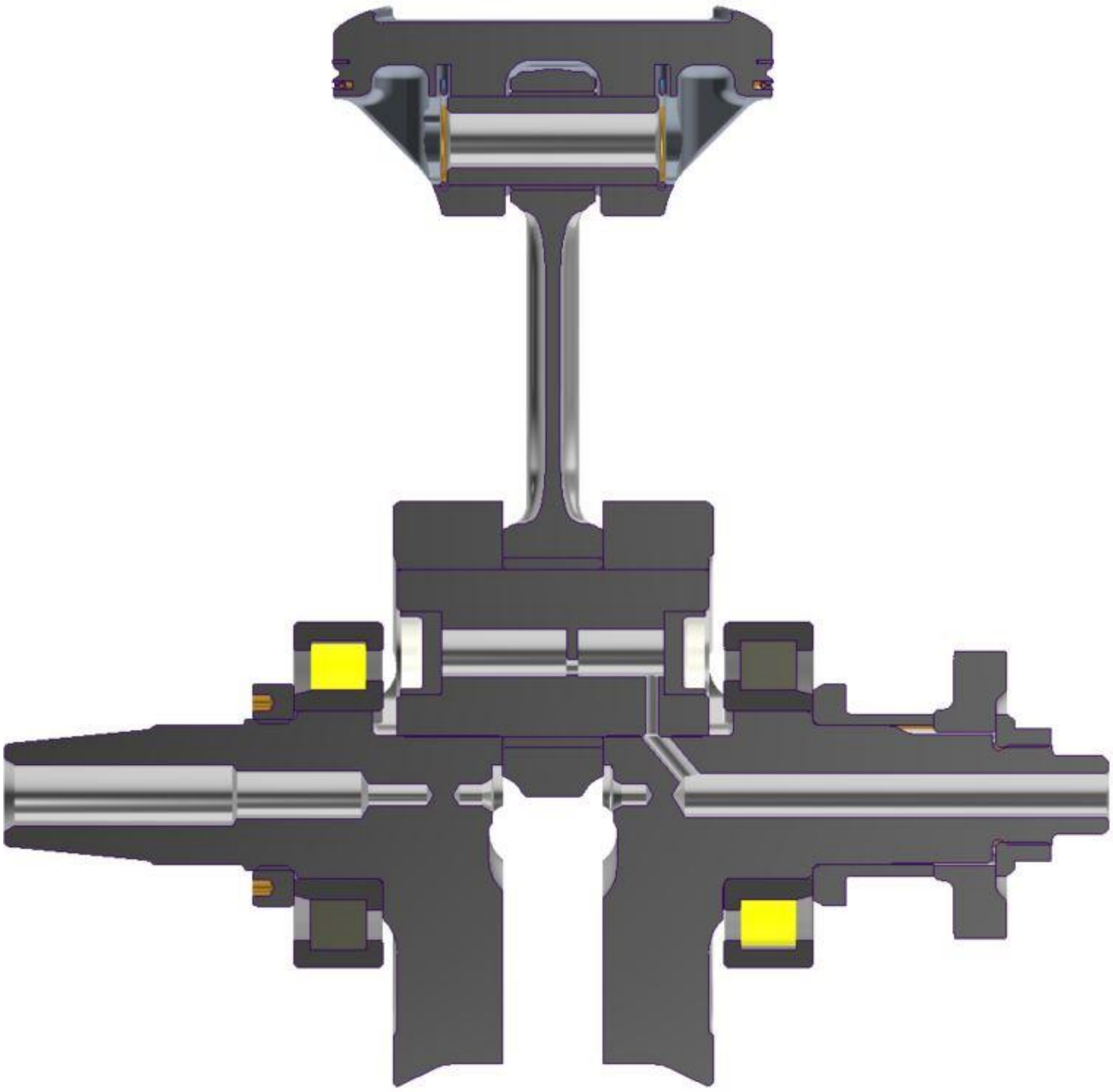


FIG. 61 SECTION VIEW OF CRANK MECHANISM

3.3.3 BALANCER SHAFT

3.3.3.1 BALANCING OF CRANK MECHANISM

In the design and manufacture of a reciprocating internal combustion engine, it is important that the crank mechanism is balanced. The inertial forces and moments of all moving parts of the motor are transmitted to the entire frame and create unwanted vibrations. It is necessary to balance two types of inertial forces. The first type is the balancing of the inertial forces of the rotating parts. This balancing is achieved by placing two balancing weights on both crankshafts (the right one and the left one), which generate the same inertia forces. The second type is balancing of the inertial force of the reciprocating parts. The calculation balancing is described below. It must be said in advance that this is a motocross engine with a capacity of 250ccm, where low weight and compactness of the engine is preferred. In practice, this means that either one or no balancing shaft is used in these engines. It is important that the balancer shaft is as close as possible to the crank axis so that it does not generate too much tilting torque. In my case, I was inspired by the engine of KTM 250 SX-F, where there is only one small balancer shaft on the right side of the engine, which drives the water pump and the cam chain is driven through it. It is therefore a compromise between balancing, weight saving, good compactness, and further use of the balancer shaft (water pump, chain).

CALCULATION OF BALANCER FACTOR

The balancer factor was calculated in an Excel file provided by Ricardo Prague s.r.o. To calculate the balancer factor, it is necessary to calculate 3 forces that arise at maximum speed, i.e. at 13,000 revolutions. The first force is the force of the reciprocating masses of the piston assembly, the second force is the force of the rotating masses of the crankshaft assembly, and the third force is the force of the rotating masses of the balancer shaft assembly.

Firstly, it is necessary to do a reduction of the connecting rod. It means that the connecting rod is divided into two mass points so that we can assign its mass to the rotating masses of the crankshaft assembly, and the remaining to the reciprocating masses of the piston assembly. The schema of the connecting rod is shown in the Fig. 62, where the mass point m_1 is calculated according to equation (18) and the mass point m_2 is calculated according to equation (19). The total mass of the connecting rod and the center of gravity of the connecting rod were measured in the Creo Parametric software.

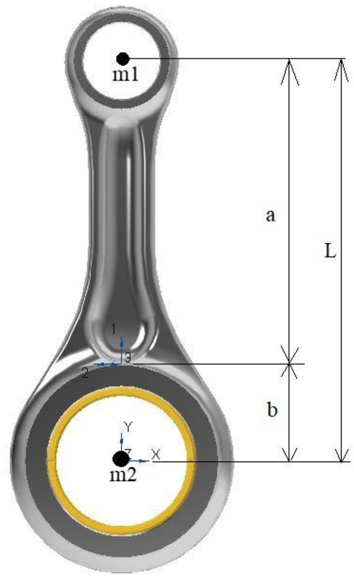


FIG. 62 REDUCTION OF CONNECTING ROD

$$m_1 = m_0 \cdot \frac{b}{L} = 261{,}6 \cdot \frac{22{,}27}{92} = 63.3g \tag{18}$$

$$m_2 = m_0 \cdot \frac{a}{L} = 261{,}6 \cdot \frac{69{,}73}{92} = 198.3g \tag{19}$$

Where: m_1 – mass point 1 (the reciprocating mass of the piston assembly)

m_2 – mass point 2 (the rotating mass of the crankshaft assembly)

m_0 – mass of the connecting rod

L – length of the connecting rod

a, b - coordinates of the center of gravity

A) THE FORCE OF THE RECIPROCATING MASSES OF THE PISTON ASSEMBLY

This force is calculated according to the equation (20). The masses of the individual components were measured in the Creo parameter program and are shown in the Tab. 13 below. The angular velocity of 13,000 rotations per minute had to be converted to radians per second.

Piston	172.2g
Piston pin	36.1g

Wire lock rings	0.6g
Compression ring	3.2g
Oil ring	4.3g
Mass point 1	63.3g
Total mass	279.7g

TAB. 13 MASSES OF THE PISTON ASSEMBLY

$$F_1 = m_{tot1} \cdot r \cdot \omega^2 = (279.7 \cdot 10^{-3}) \cdot (25.5 \cdot 10^{-3}) \cdot 1361.36^2 = 13\,218.4\,N \quad (20)$$

where: m_{tot1} – total mass the piston assembly

r – length of the crank radius

ω – engine speed

B) THE FORCE OF THE ROTATING MASSES OF THE CRANKSHAFT ASSEMBLY

This force is calculated according to the equation (21). The masses of the individual components were measured in the Creo parameter program and are shown in the Tab. 14 below. The angular velocity of 13,000 rotations per minute had to be converted to radians per second as in the previous case. The magnitude of the eccentricity of the crank assembly relative to the axis of rotation was also measured in the Creo Parametric program. Although the alternator is not part of the master´s thesis, but from practice we can estimate the weight of the alternator.

Left crankshaft web	1149.5g
Right crankshaft web	1135.2g
Inner bearing races	106.4g
Crank pin	269.7g
Nut M27 (left web)	40.6g
Primary gear	99.0g
Spacer ring	53.7g
Nut M8 (right web)	23.9g
Mass point 2	198.3g
Alternator	700.0g
Total mass	3776.3g

TAB. 14 MASSES OF THE CRANKSHAFT ASSEMBLY

$$F_2 = m_{tot2} \cdot r_2 \cdot \omega^2 = (3{,}776.3 \cdot 10^{-3}) \cdot (1.43 \cdot 10^{-3}) \cdot 1{,}361.36^2 = 10{,}008.0\ N \quad (21)$$

where: m_{tot2} – total mass the crankshaft assembly

r_2 – eccentricity of the crank assembly

ω – engine speed

C) THE FORCE OF THE ROTATING MASSES OF THE BALANCER ASSEMBLY

This force is calculated according to the equation (22). The mass of the balancer assembly was measured in the Creo Parametric program. The angular velocity of 13,000 rotations per minute had to be converted to radians per second as in the previous case. The magnitude of the eccentricity of the balancer assembly relative to the axis of rotation of the balancer assembly was also measured in the Creo Parametric program.

$$F_3 = m_{tot3} \cdot r_3 \cdot \omega^2 = (247.6 \cdot 10^{-3}) \cdot (2.81 \cdot 10^{-3}) \cdot 1{,}361.36^2 = 1{,}289.5\ N \quad (22)$$

To calculate the resulting balancer factor, it is necessary to substitute the results from equation (20), (21) and (22) into the equation (23). The resulting value tells us that the designed crank mechanism is balanced to 85.5%, which is not uncommon value for small motocross engines.

$$b_f = \frac{F_2+F_3}{F_1} = \frac{10{,}008.0+1{,}289.5}{13{,}218.4} = 0.855 \quad -> \quad 85.5\% \quad (23)$$

where: b_f – balancer factor

F_1 – force of the reciprocating masses of the piston assembly

F_2 – force of the rotating masses of the crankshaft assembly

F_3 – force of the rotating masses of the balancer assembly

3.3.3.2 BALANCER SHAFT CONSTRUCTION

As already mentioned, the balancer shaft is driven by a primary gear on the right crankshaft web with a gear ratio of 1. The parameters of the balancer gear are given in the

Tab. 15. The balancer gear is mounted on the balancer shaft using a parallel key 5e7x5x15 ČSN 022562. The inspection of the parallel key is shown below. The left side of the balancer shaft is mounted in the needle roller bearing with a machined ring, and without an inner ring. Specifically, it is a NK 16/16 bearing from the SKF company, which is also used in the KTM 250 SX-F motorcycle. On the left side of the balancer shaft there is also a balance weight and a sprocket with 17 teeth that drives the cam chain. The design of the cam chain is described in the chapter 3.3.4.4, where the diameter of the pitch circles of this sprocket is also calculated. On the right side of the balancer shaft there is a space for a water pump and a seal. The designed model of the balancer shaft is shown in the Fig. 63. The balancer gear and the balancer shaft are made of 20MnCr5 (DIN 1.7147) steel.

z_2 [-]	24
m_{n2} [mm]	2
α_{n2} [°]	20
b_{w2} [mm]	8
$\beta 2$ [°]	0

TAB. 15 PARAMETERS OF GEAR OF BALANCER SHAFT

Where: z_2 – number of teeth of balancer gear

m_{n2} – module of balancer gear

α_{n2} – normal pressure angle of balancer gear

b_{w2} – face width of balancer gear

β_2 – helix angle of balancer gear

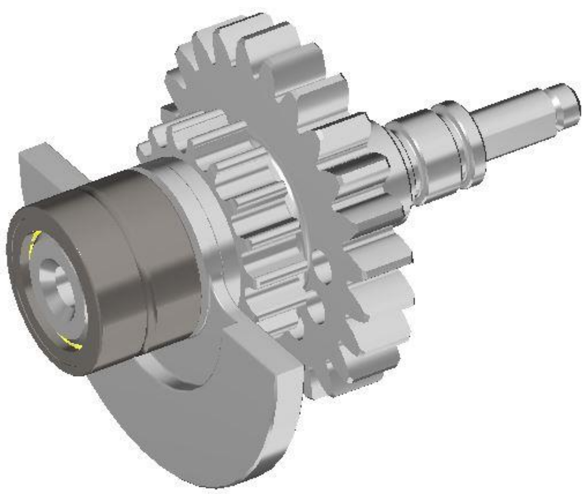


FIG. 63 ASSEMBLY OF BALANCER SHAFT

INSPECTION OF THE PARALLEL KEY

The parallel key inspection is calculated according to equation (24). The resulting value must be less than 120 MPa, which is the maximum allowable pressure. The calculation is performed for the torque of 17 Nm, which was provided to me by Ricardo form *Valdyne* analysis.

$$p = \frac{4 \cdot \mathrm{M_k}}{n_k \cdot \mathrm{d_s} \cdot l_k \cdot h_k} = \frac{4 \cdot 17{,}000}{1 \cdot 15.5 \cdot 15 \cdot 5} = 58.5 \frac{N}{mm^2} = 58.5 MPa \tag{24}$$

where: M_k – maximum torque

n_k – number of parallel keys

d_s – diameter of balancer shaft

l_k – length of parallel key

h_k – height of parallel key

3.3.4 VALVETRAIN

3.3.4.1 VALVES

As mentioned earlier, the engine has two intake and two exhaust valves. Both valves are mechanically and thermally stressed. A higher thermal stress acts on the exhaust valves because hot exhaust gases pass around them. However, the intake valves are more mechanically stressed due to the inertial force because they are larger and heavier. Therefore, it was necessary to choose a suitable material, which in this case is a titanium alloy Ti6242, which is commonly used for high-performance racing engine parts. The diameter of the intake valves head should be as large as possible to allow as much mixture as possible into the cylinder. In this case, the diameter of intake valve head is 32mm and the diameter of exhaust valve head is 26.5mm. The angle of inclination of the intake valves from the axis of the cylinder is 10.5 degrees and the angle of inclination of the exhaust valves is 13 degrees. The diameter of the stems for all four valves is 5mm. The designed models of the intake valve (blue) and exhaust valve (red) are shown in the Fig. 64.

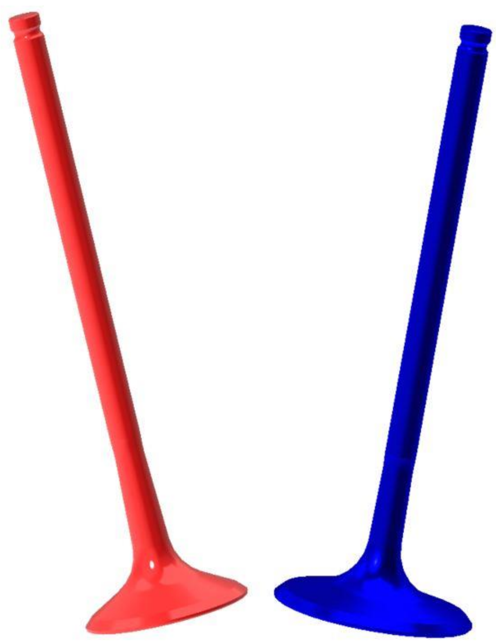


FIG. 64 INTAKE AND EXHAUST VALVES

The cross section of the valves with other parts is shown in the Fig. 65. At the top of the valve stem is a groove for valve collets. Two valve collets are inserted into the groove, which settle by means of a conical surface into a valve spring retainer. The valve retainers sit on top of the springs. As shown in the Fig. 65, the intake valvetrain includes two springs (inner and outer). This is because the intake valves are larger and heavier and thanks to the two springs, higher spring force is possible. When designing the springs, I was inspired by the KTM 250 SX-F engine and their basic dimensions are shown in the Tab. 16. There are valve spring seats on the bottom of the springs to reduce the contact stress between the spring and the cylinder head. The valves are guided in valve guides, which are described in the chapter 3.3.5. Valve stem seals (gold parts in the Fig. 65) are fitted on the valve guides to prevent oil from coming through the valves into the ports. Valve shims which have the exact thickness are placed between the valve and the finger follower. They are used to set the valve clearance, because they can be replaced with shims of a different thickness.

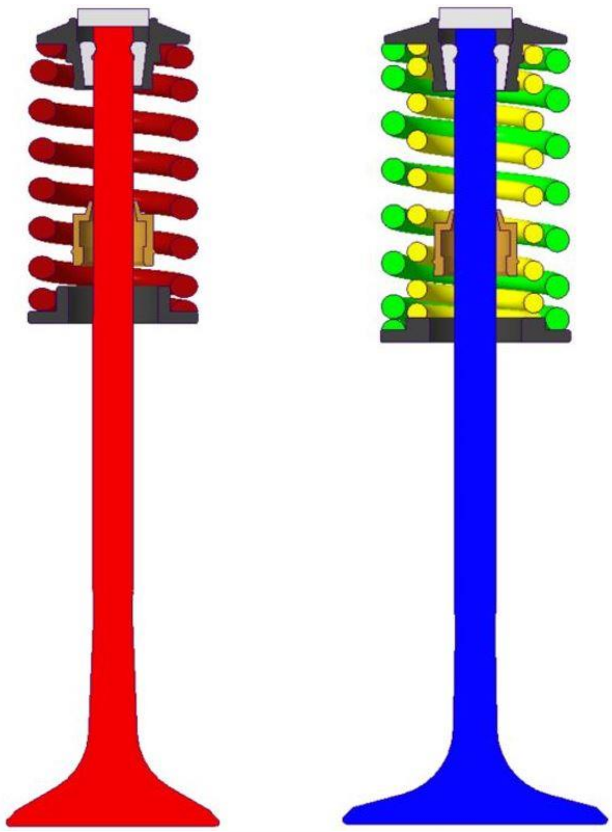


FIG. 65 SECTION VIEW OF INTAKE AND EXHAUST VALVE

	Exhaust	Intake inner	Intake outer
Wire width	3.4mm	2.25mm	2.85mm
Wire height	2.6mm	2.25mm	2.85mm
Outside diameter	21mm	16.45mm	22.65mm
Total number of turns	7.75	8.5	7
Number of dead turns	1.5	1.5	1.5

Free length	41.5mm	39.7mm	42.5mm
Fitted length	33mm	33mm	34.5mm

TAB. 16 DIMENSIONS OF VALVE SPRINGS

3.3.4.2 FINGER FOLLOWERS

A finger follower is a rotary lever that activates the opening and closing of valves by moving the camshafts. One finger follower belongs to each valve. They are mounted on a shaft with a diameter of 7.5mm, which is mounted in the head of the engine. The surface of the finger followers is DLC (diamond like carbon) coated to reduce friction. The company Ricardo Prague s.r.o. provided me with a profile of the finger follower. The finger followers are made of tool steel AISI H11. The resulting model of finger follower is shown below in the Fig. 66.

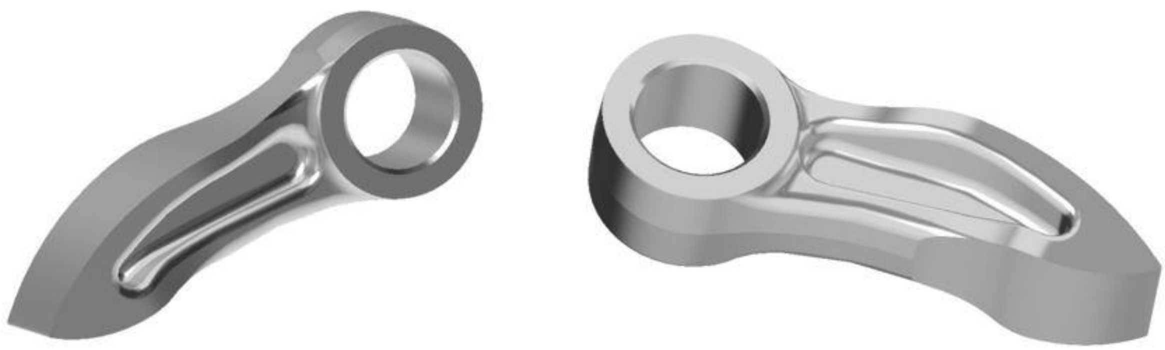


FIG. 66 FINGER FOLLOWER

3.3.4.3 CAMSHAFTS

The opening and closing of the valves are caused by a shaft with cams, which ensures the conversion of rotary motion into sliding. The designed model of camshafts is shown in the Fig. 67, where the left camshaft is for exhaust valves and the right camshaft is for intake valves. At the end of the shafts are pressed sprockets with 34 teeth, which are driven by a cam chain which is driven by a balancing shaft. The design of the cam chain is described in the chapter 3.3.4.4, where the diameter of the pitch circles of the sprockets on the camshafts are also calculated. To save weight, the shafts are hollow and there are several holes in the sprockets. The profile of the cam was provided to me by Ricardo Prague s.r.o. The maximum lift of intake valves is 10.8mm and the maximum lift of exhaust valves is 9.8mm. Both camshafts are made of 20MnCr5 steel.

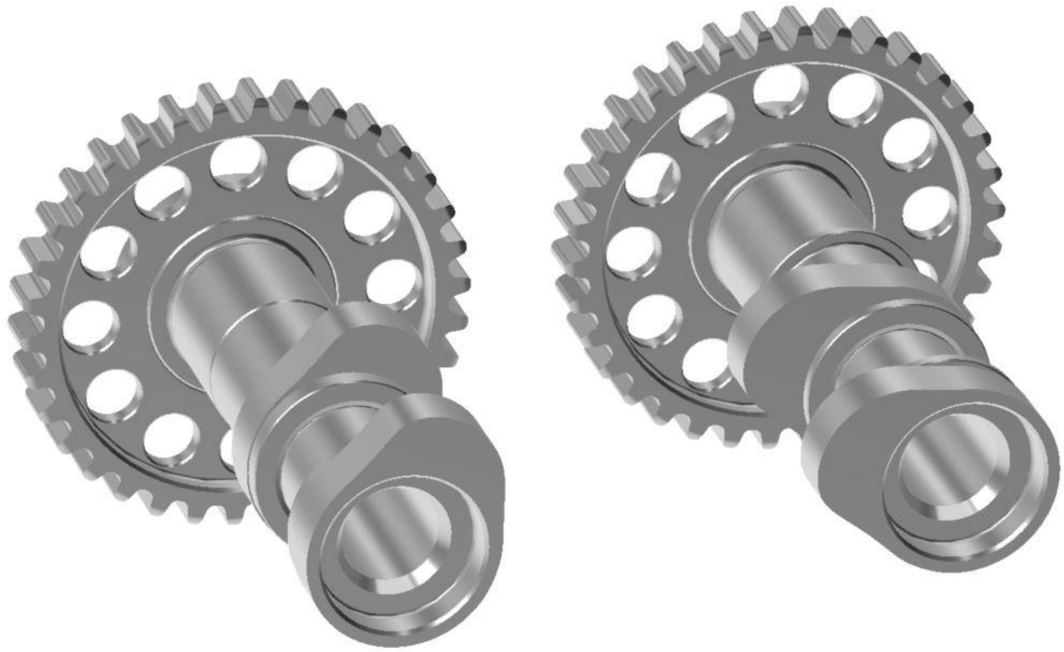


FIG. 67 CAMSHAFTS

3.3.4.4 CAM CHAIN

The transmission between the crank and cam mechanism is provided by a silent chain, which is commonly used in motocross engines. In this case it is a chain DID SCZT-0412L SV, which has a pitch of 6.35mm. Both sprockets on the camshaft are identical and the ratio between them and the sprocket on the balancer shaft is 2:1, so they have twice the number of teeth than the sprocket on the balancer shaft. The cam chain has 110 links and its length is calculated according to equation (25). The diameter of the pitch circles of the sprockets on the camshafts and the balancer shaft are calculated below according to equations (26) and (27).

$$L_{CH} = n_{CH} \cdot P_{CH} = 110 \cdot 6{,}35 = 698{,}5\ mm \tag{25}$$

where: L_{CH} – length of cam chain

n_{CH} – number of links

P_{CH} – pitch of the cam chain

$$D_1 = \frac{p_{CH}}{\sin\frac{180}{z_B}} = \frac{6.35}{\sin\frac{180}{17}} = 34.56\ mm \tag{26}$$

where: D_1 – diameter of the pitch circle of sprocket on the balancer shaft

z_B – number of teeth of sprocket on the balancer shaft

$$D_2 = \frac{p_{CH}}{\sin\frac{180}{z_C}} = \frac{6.35}{\sin\frac{180}{34}} = 68.82\ mm \tag{27}$$

where: D_2 – diameter of the pitch circle of sprocket on the camshafts

z_C – number of teeth of sprocket on the camshafts

The valvetrain also includes chain guides, which have the task of avoiding contact of the chain with other parts of the engine, preventing the chain from oscillating and keeping the chain sufficiently tensioned. The chain guides are made of polyamide PA66. The designed model of the cam chain with chain guides is shown below in the Fig. 68.

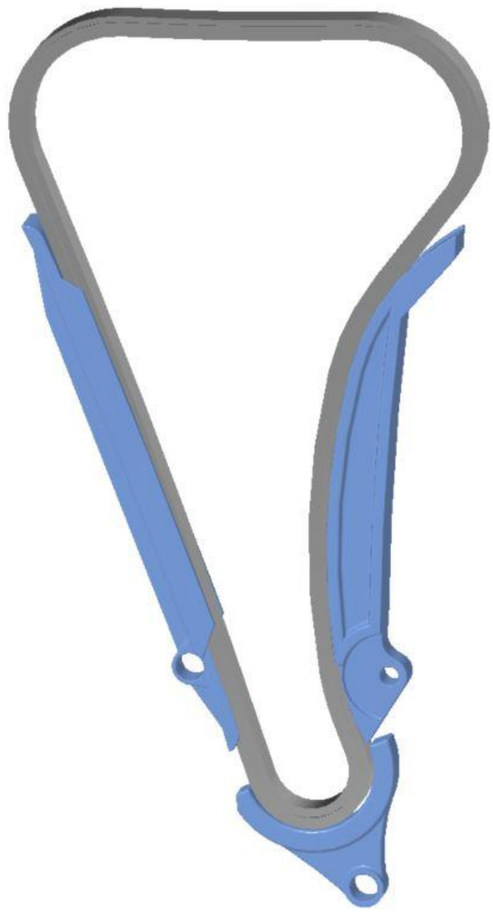


FIG. 68 CAM CHAIN WITH CHAIN GUIDES

3.3.4.5 CAMSHAFT COVER

A camshaft cover holds the camshafts in the correct position and absorbs the forces generated in the valvetrain mechanism. The camshaft cover model is shown in the Fig. 69, with a top view on the left and a bottom view on the right side. The cover is attached to the head with eight bolts DIN6921 M6x35. Thus, there are only 8 through holes with a diameter of 7mm in the cover (blue circle in the Fig. 69) and the threads are only in the engine head. So that the cover fits exactly in the correct position, the holes for the bolts on the edge are provided with dowels (yellow circle in the Fig. 69). On the top of the holder there are 3 holes with a thread (red circle in the Fig. 69), which are used to attach the magnesium cover of the engine head. From the engine head, the oil is led to the camshafts through channels that are visible on the bottom view of the cover. The camshaft cover is made of cast aluminum alloy ADC12.

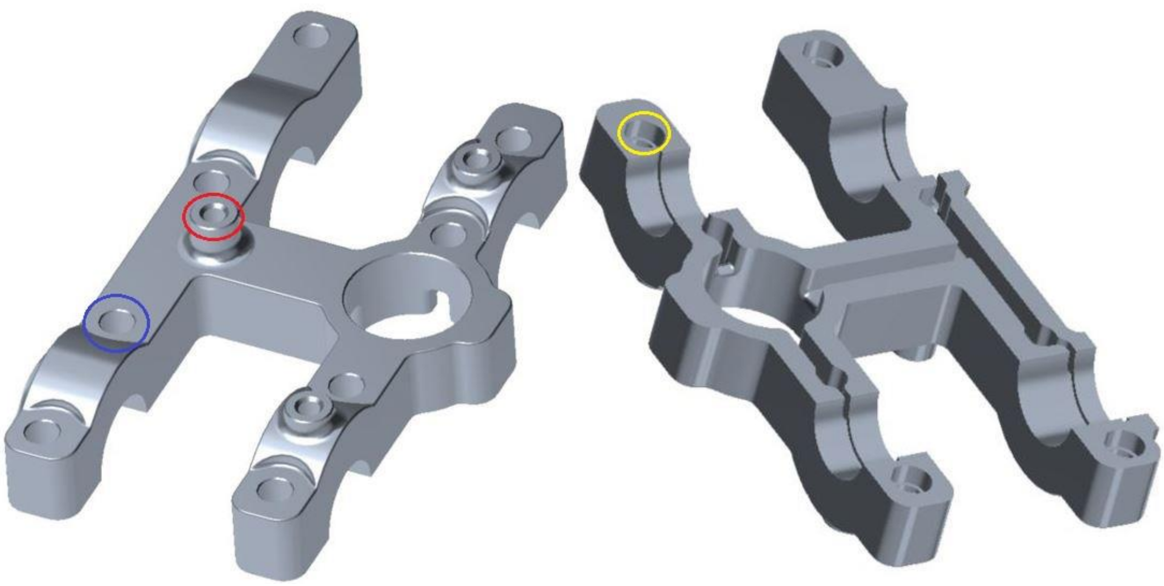


FIG. 69 TOP VIEW AND BOTTOM VIEW OF THE CAMSHAFT COVER

3.3.4.6 VALVETRAIN ASSEMBLY

An overall assembly of the valvetrain is shown in the Fig. 70. In addition, a top chain guide is shown, which is mounted to the camshaft cover and acts on the chain from above.

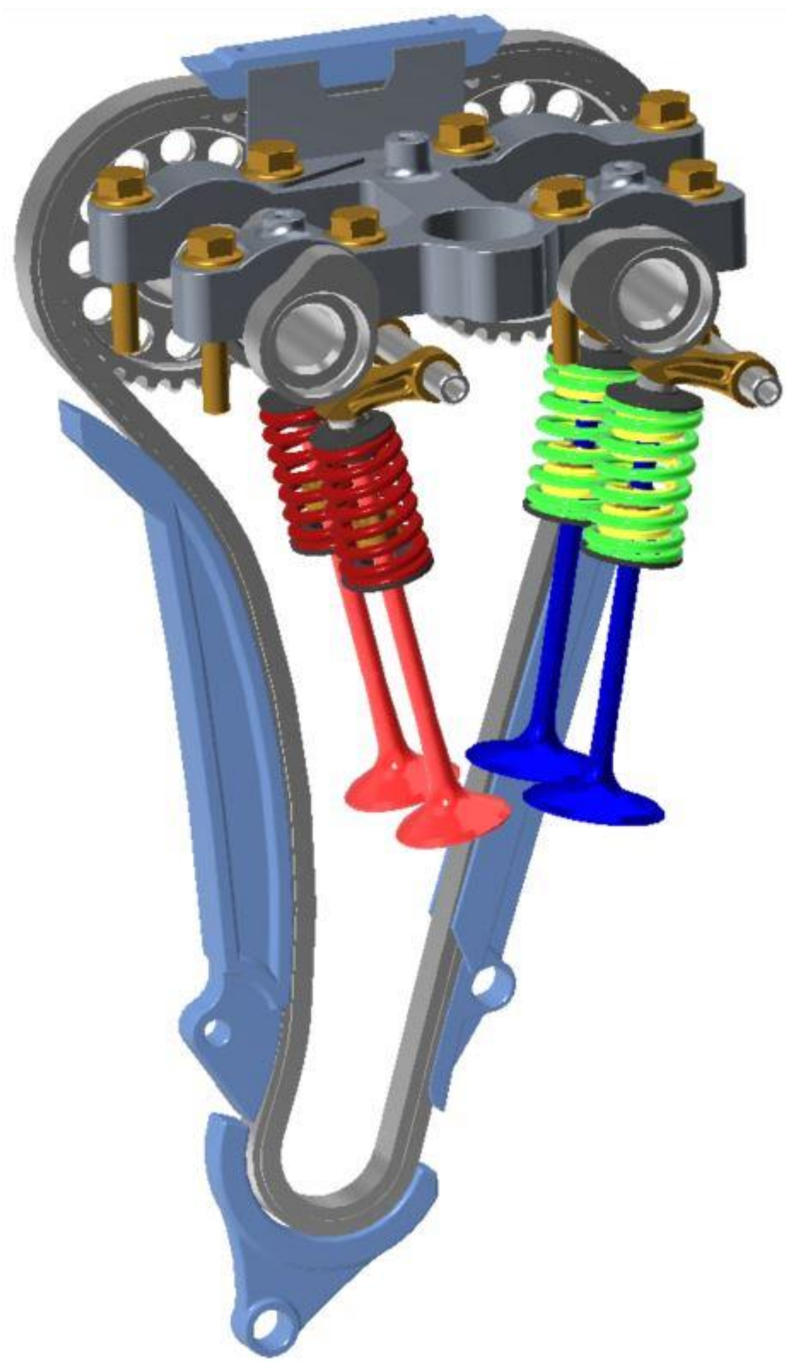


FIG. 70 VALVETRAIN ASSEMBLY

3.3.5 CYLINDER HEAD

The cylinder head is the most complex part of the engine and its overall detailed design would go beyond the task of this master´s thesis. Therefore, the head has been designed and constructed only in a simplified way to show that there is no collision of engine parts. The cylinder head is cast from the aluminum alloy A356. The Fig. 71 shows the surfaces representing the intake and exhaust channels, the space for a spark plug, the holes in the head for the valve guides, and the cooling core of the head. The intake channels coming out of the combustion chamber have a diameter of 28.5mm, their length is 98.5mm, and after mutual intersection, they have an ellipsoidal cross-section at the entrance to the head. The exhaust channels coming out of the combustion chamber have a diameter of 23mm, their length is 80mm, and after mutual intersection, they also have an ellipsoidal cross-section at the exit of the head. The holes in the cylinder head for the valve guides have the same inclination as the valves themselves, i.e. 10.5 degrees for the intake and 13 degrees for the exhaust. The cooling liquid is supplied to the cylinder head via cooling channels from the cylinder, i.e. from below the head. In practice, a simulation of flow and heat transfer is performed for the complete design of the cooling core system of the engine head, which is, however, beyond the task of my thesis. Therefore, the cooling system was modeled only in a simplified way, where I tried to concentrate the volume of the cooling core in places around the exhaust channels, where the head is most thermally stressed. This is not needed for intake channels, because an intake air with a temperature of around 20 degrees Celsius passes through them, which cools them.

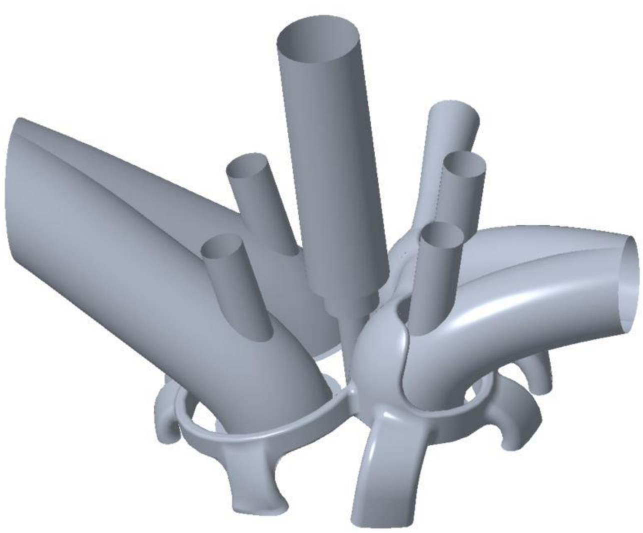


FIG. 71 INT. AND EXH. CHANNELS, SPARKPLUG, VALVE GUIDELINES AND COOLING SYSTEM

The model of the cylinder head is shown in the Fig. 72. The cylinder head is turned so that the exhaust channels with a flange for connection to the exhaust are turned towards us. The connection of the cylinder head, cylinder, and engine cases is ensured by four main and one secondary bolts, which are described in the Chapter 3.3.9. In the head there are two holes for these bolts with a diameter of 10.7mm from the outside (red circle in the Fig. 72), two holes with the same diameter from the inside and one hole with a diameter of 6.5mm for the secondary bolt on the side of the cam chain. The green circle in the Fig. 72 shows a hole for the shaft on which the finger followers of the intake valves are mounted. On the same side there is also a hole for the shaft for the exhaust finger follower. These shafts are secured against sliding out with screws. The orange arrow in the Fig. 72 shows a hole for the chain tensioner that presses on the chain guide that tensions the chain. The coolant is led from the cylinder head to the engine radiator through a hole shown in the blue circle in the Fig. 72. The yellow circle shows a hole through which oil is supplied to the camshafts cover, from where the surfaces between the head and the camshafts are lubricated. At the top of the head there are eight holes with a M6 thread, thanks to which a cover is mounted to the head.

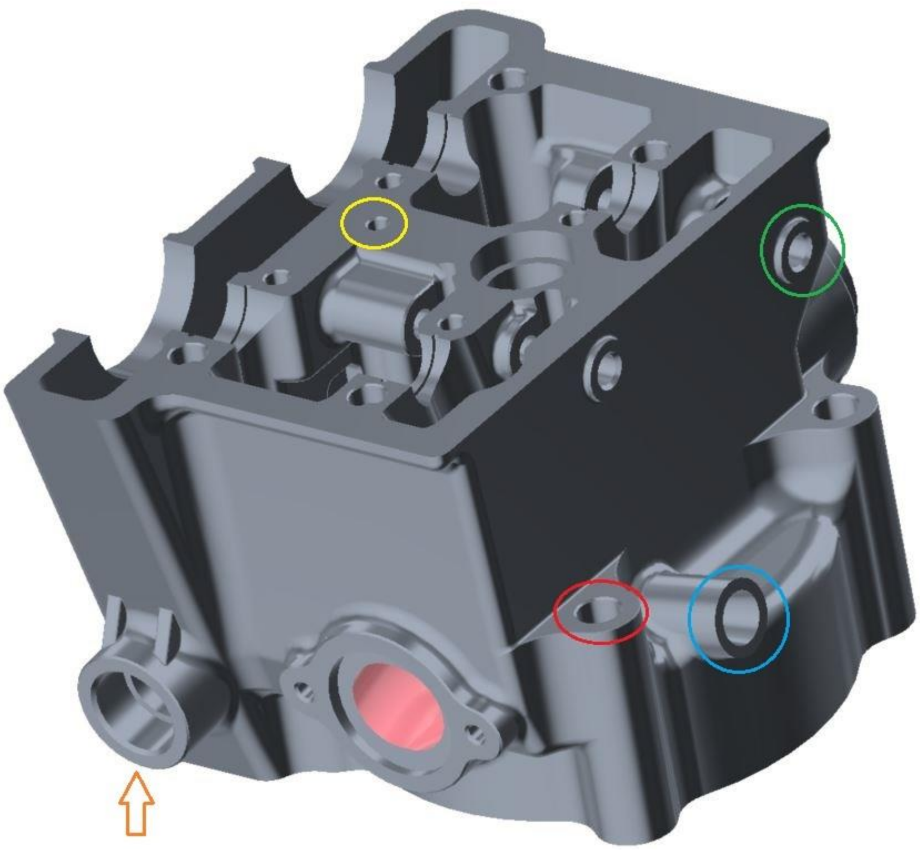


FIG. 72 CYLINDER HEAD

The Fig. 73 shows the bottom of the head. It can be seen from the Fig. 73 that it is a pent-roof combustion chamber. The combustion chamber volume should be as small as possible due to the compression ratio, valves as large as possible, and burning paths as short as possible. Therefore, the spark plug hole is in the center of the cylinder. The eight cooling channels leading to the head are visible in the Fig. 73 as well. The head includes four steel valve guides, which are pressed into the head. Another part of the head are four steel valve seats, which are also pressed into the head. The valve guides and valve seats are shown in the Fig. 74.

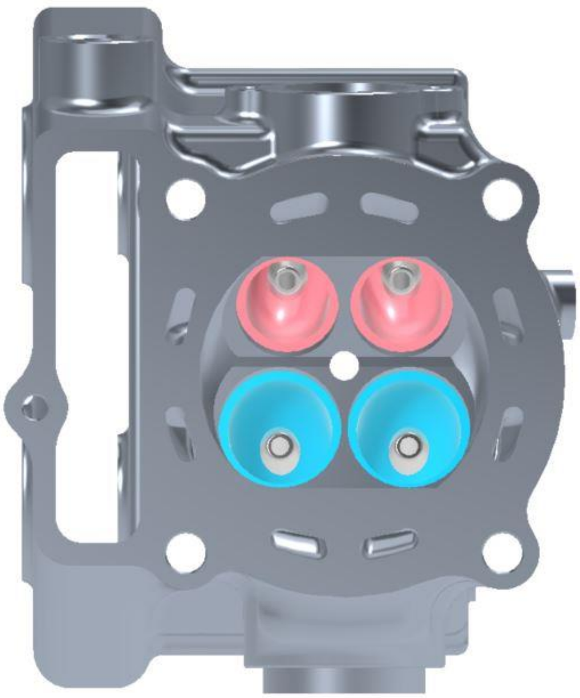


FIG. 73 BOTTOM OF THE CYLINDER HEAD

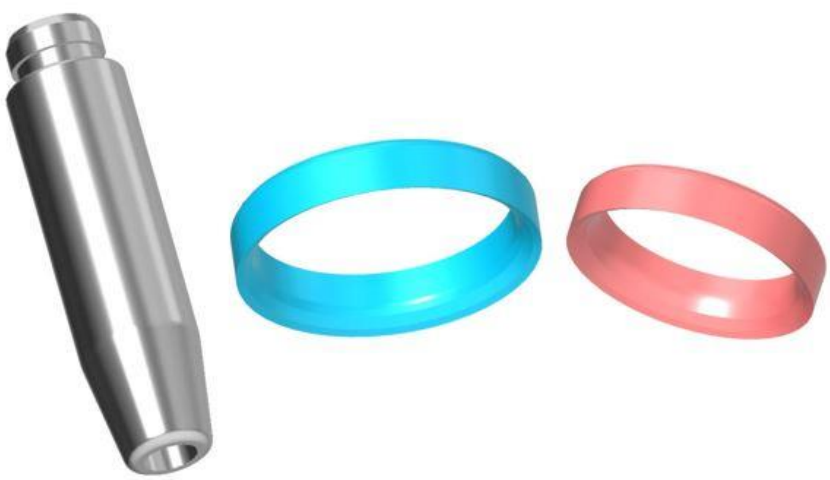


FIG. 74 VALVE GUIDE AND VALVE SEATS

3.3.6 CYLINDER

The task of the cylinder is to guide the piston along its trajectory and its inner diameter has the same value as the designed bore, i.e. 79mm. Its other function is heat dissipation, where a cooling liquid is supplied from the engine case (blue arrow in the Fig. 75), which cools the walls of the cylinder and then passes through the eight holes on the top of the cylinder into the head. There are five holes for bolts (four main and one secondary) that hold the head, cylinder, and engine cases together. A lubrication is further fed to the cylinder head through the holes shown in the red circle in the Fig. 76. The material had to be removed at the bottom of the cylinder (green arrow in the Fig. 76) because there was a collision of the cylinder with the engine case where the bearing of the balancer shaft is mounted. The cylinder is made of the aluminum alloy AlSi12CuMg.

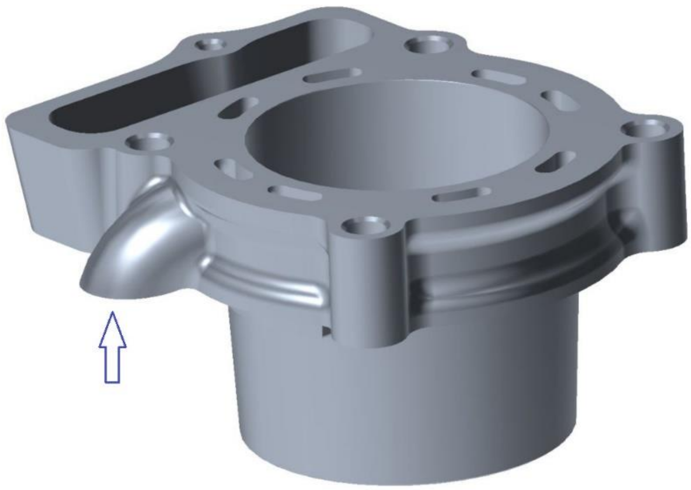


FIG. 75 CYLINDER

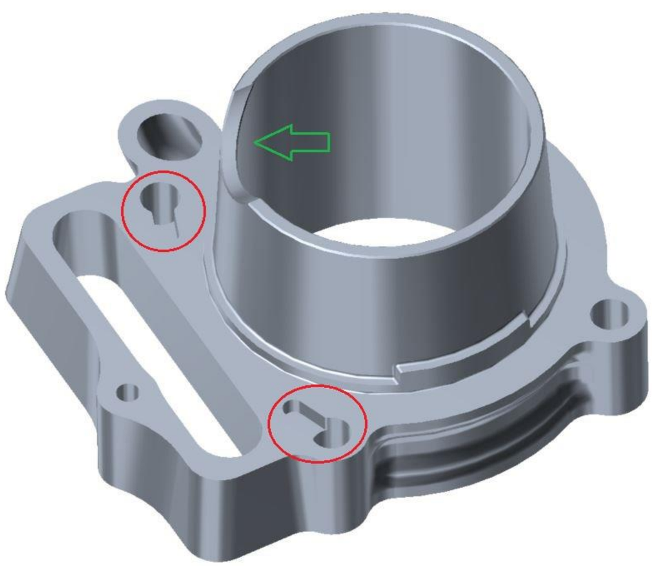


FIG. 76 BOTTOM OF THE CYLINDER

3.3.7 CLUTCH AND TRANSMISSION

The clutch and 5-speed transmission are designed only schematically, because it is not part of this thesis. The basic dimensions of the clutch and transmission need to be known due to the design of the engine cases, so they were taken over from the engine of KTM 250 SX-F. A simplified model of the clutch and transmission is shown in the Fig. 77.

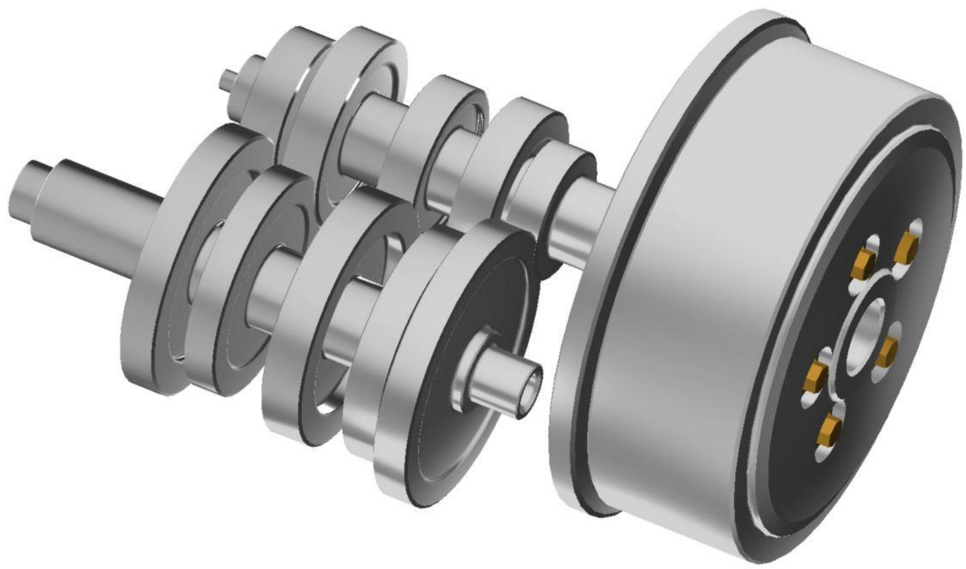


FIG. 77 CLUTCH AND TRANSMISSION

3.3.8 ENGINE CASES

The complete design of the engine case is very complex and would go beyond the scope of this thesis. Therefore, the engine case is designed only in a simplified way, so that all engine parts fit into it and there is no collision among them. The engine case consists of two parts (right and left case), which are connected by eleven torx hex flange head bolts M6. Both parts are made of the aluminum alloy Al4046.

3.3.8.1 LEFT ENGINE CASE

A view of the inside of the left engine case is shown in the Fig. 78. There are two holes with a thread M10 at the top of the case, in which the bolts that hold the cases, cylinder, and head together are located. The green arrows in the Fig. 79 point to the holes where the gearbox shafts are mounted. The red circles show holes with M5 thread, which together with the sheet metal and screws prevent axial movement of the cylindrical roller bearing on the crankshaft. The purple arrow points to the hole where the oil filter is located.

The orange arrow points to the channel through which the oil is drawn from the crankshaft space.

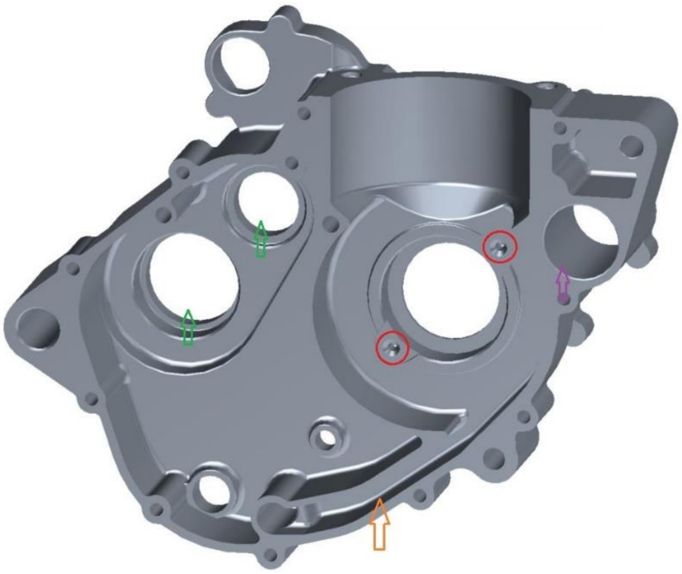


FIG. 78 LEFT ENGINE CASE INSIDE

A view of the outside of the left case is shown in the Fig. 79. The blue circle indicates where the alternator, which is on the crankshaft, is located. Then there is an upward space for the gears that lead to the red arrow where the electric starter is located. The orange arrow in the Fig. 79 indicates where the suction oil pump is located.

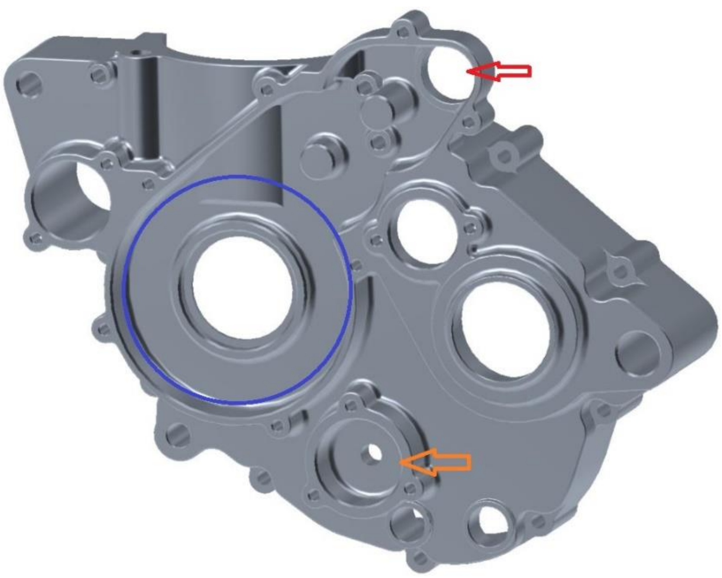


FIG. 79 LEFT ENGINE CASE OUTSIDE

3.3.8.2 RIGHT ENGINE CASE

A view of the inside of the right engine case is shown in the Fig. 80. At the top of the case, in addition to the two holes with a thread M10, there are 2 smaller holes (yellow circles in the Fig. 80) through which oil flows from the case along the bolts to the cylinder head. Furthermore, there is a hole shown in a blue circle through which the coolant flows into the cylinder. The green arrows in the Fig. 80 point to the holes where the gearbox shafts are mounted. The orange arrow points to the hole where the oil filter is located. The red circles show holes with M5 thread, which has the same purpose as the left case.

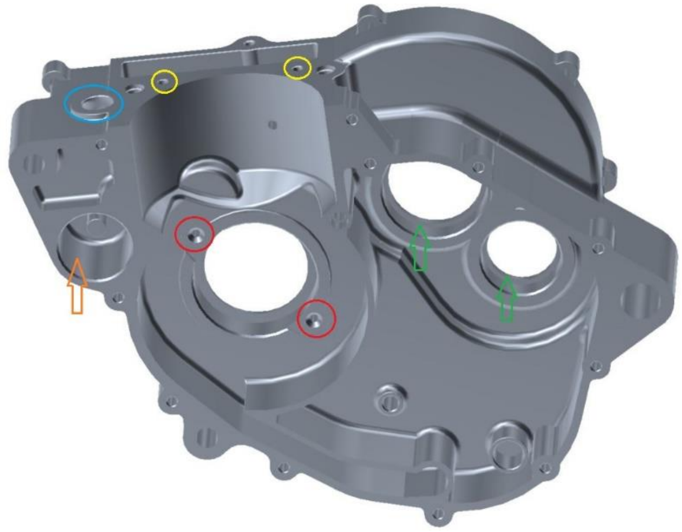


FIG. 80 RIGHT ENGINE CASE INSIDE

A view of the outside of the right engine case is shown in the Fig. 81. Holes with M6 thread are shown in red circles, thanks to which the chain guides are attached to the case. The green circle indicates where the needle roller bearing of the balancer shaft is located. The blue arrow in the Fig. 81 indicates the space where the water pump pushes the coolant to the cylinder. The orange arrow in the Fig. 81 points where the force oil pump is located.

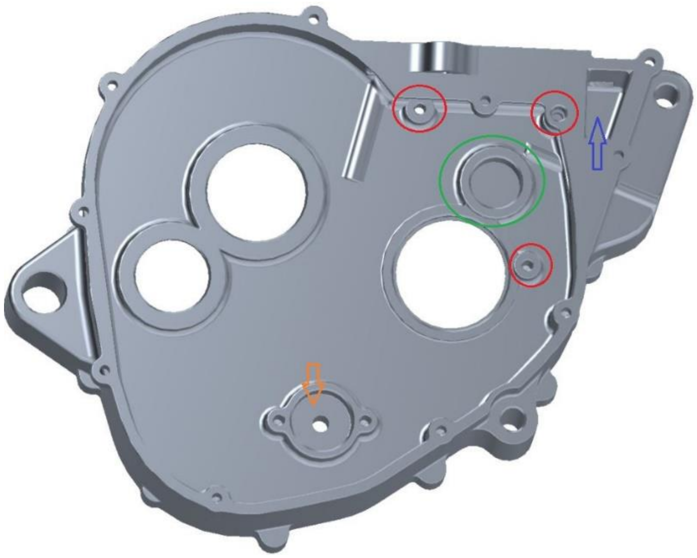


FIG. 81 RIGHT ENGINE CASE OUTSIDE

3.3.9 ENGINE ASSEMBLY

A left view of the entire engine assembly is shown in the Fig. 82 and a right view of the entire engine assembly is shown in the Fig. 83. The cylinder head, cylinder, and engine cases are connected by stud bolts M10x1.25, which are inspected in this section below. Between the head and the cylinder and between the cylinder and the engine cases there are sheet metal seals which copies the seating surfaces of the components.

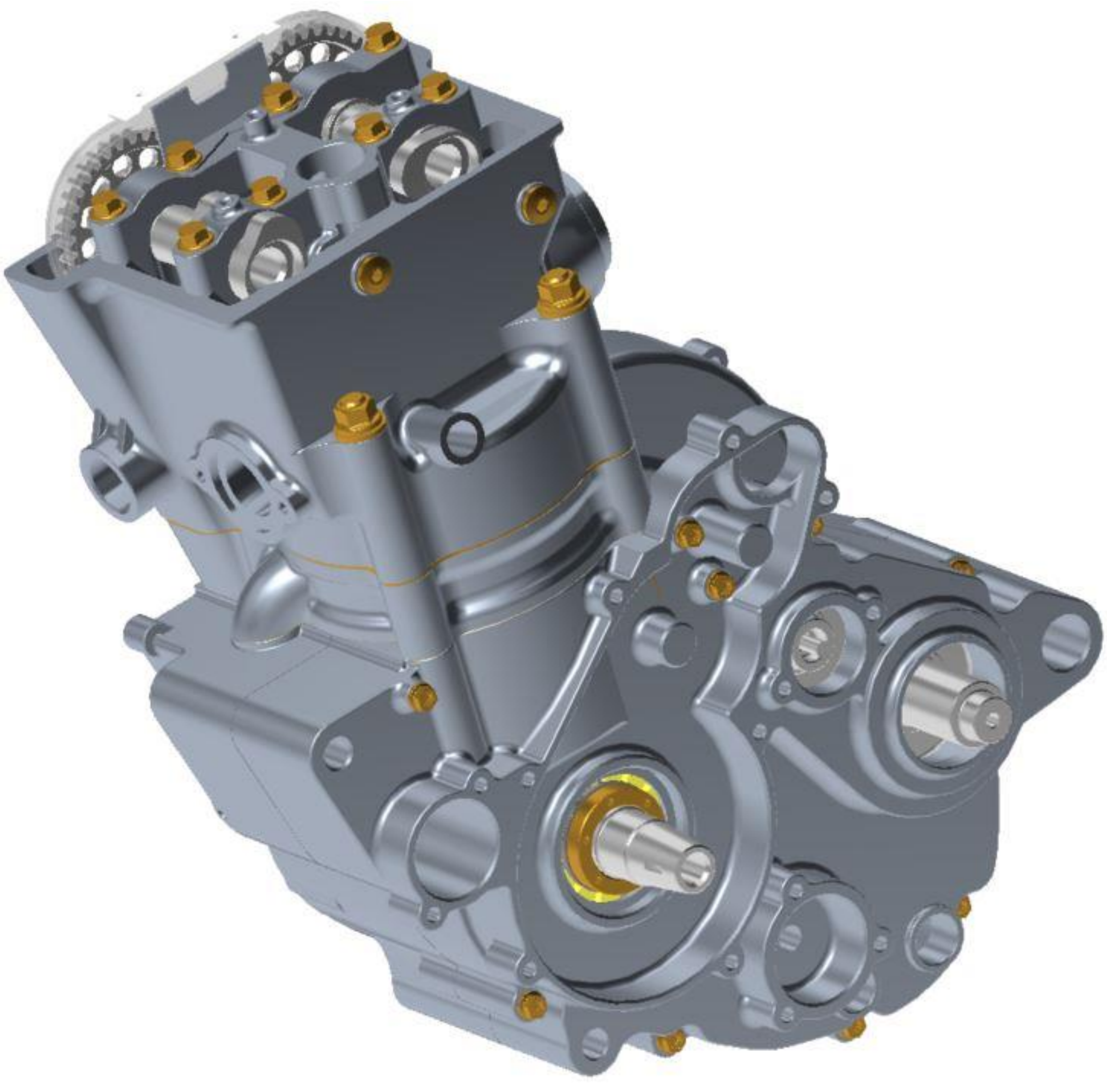


FIG. 82 LEFT VIEW OF THE ENGINE ASSEMBLY

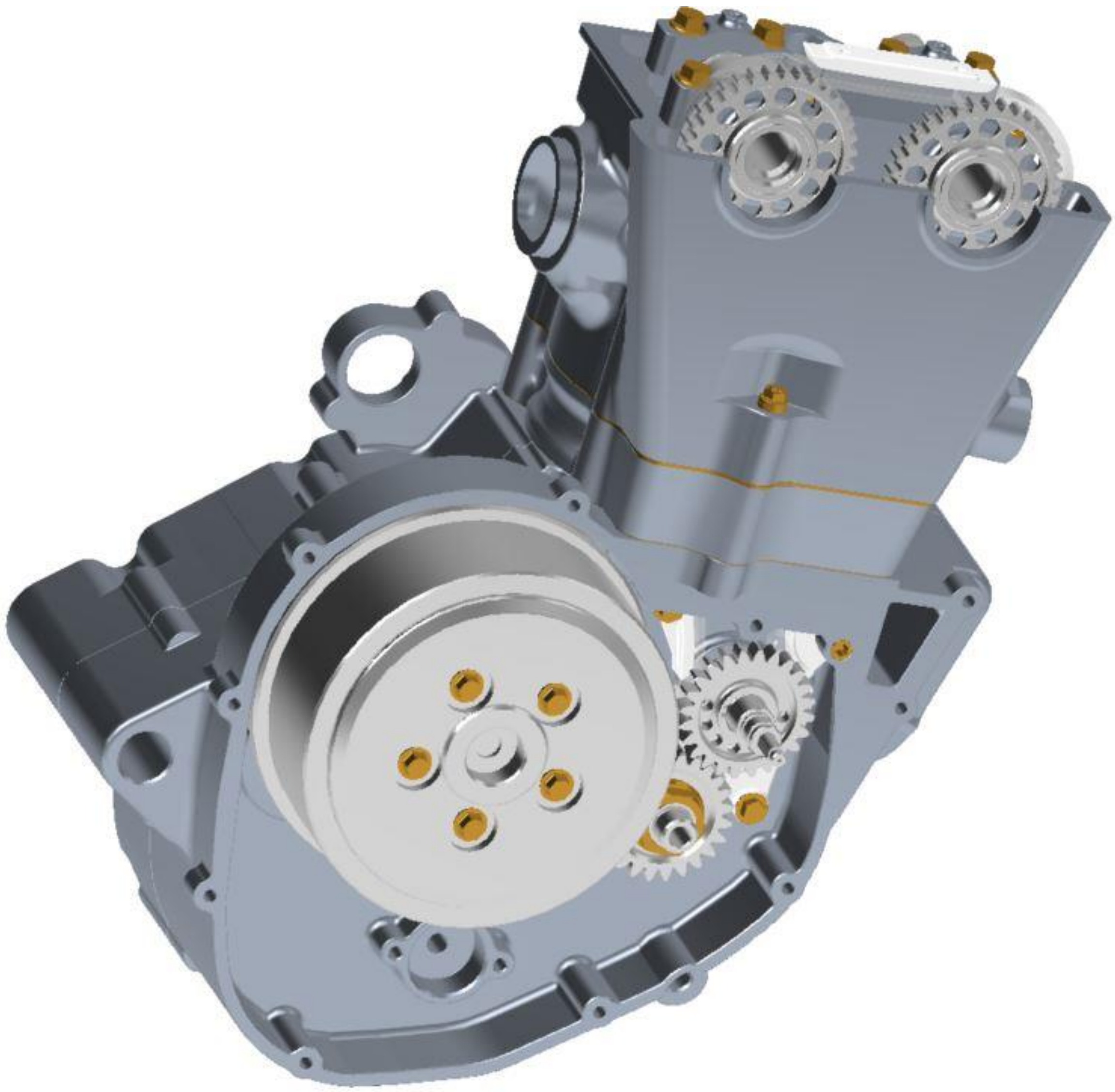


FIG. 83 RIGHT VIEW OF THE ENGINE ASSEMBLY

INSPECTION OF THE STUD BOLTS

The inspection is performed for the highest pressure. The yield strength for bolts with a strength grade of 8.8 is 640 MPa. The force acting on all bolts is calculated according to equation (28). The force acting on one bolt is calculated according to equation (29). The allowable stress is calculated according to equation (30), where the safety factor *k* is equal to 3. The minimum diameter of the bolt is calculated according to equation (31).

$$F_{B4} = p \cdot S = p \cdot \frac{\pi \cdot D^2}{4} = 9.52 \cdot \frac{\pi \cdot 79^2}{4} = 46{,}663.9\ N \tag{28}$$

where: F_{B4} – force acting on all bolts

p – maximum pressure

D – bore

$$F_{B1} = \frac{F_{B4}}{4} = \frac{46{,}663.9}{4} = 11{,}665.98\ N \tag{29}$$

where: F_{B1} – force acting on one bolt

$$\sigma_D \geq \frac{\sigma_s}{k} = \frac{640}{3} = 213.3\ MPa \tag{30}$$

where: σ_D – allowable stress

σ_s – yield strength for bolts with a strength grade of 8.8

k – safety factor

$$d_b \geq \sqrt{\frac{4 \cdot F_{B1}}{\pi \cdot \sigma_D}} = \sqrt{\frac{4 \cdot 11{,}665.98}{\pi \cdot 213.3}} = 8.34\ mm \tag{31}$$

where: d_b – minimum diameter of the bolt

According to the Engineering Tables [43], the small diameter of the M10x1.25 thread is equal to 8.466mm. The calculated minimum diameter of the bolt is smaller than this value and therefore the designed bolts comply.

3.4 STRENGTH ANALYSIS OF THE CRANK MECHANISM

The strength analysis is performed in the software Creo Parametric. The analysis is performed at three different engine speeds. The first case is at maximum speed, which is 13,000rpm. The second case is at the speed when the highest brake power occurs, which is 12,000rpm. And the third case is at the speed when the highest brake torque occurs, which is 9,000rpm.

3.4.1 MODEL PREPARATION

First, it is necessary to correctly engage the crank mechanism in the Creo software. Then in the software, we go to an application called a *Mechanism*. Here it is necessary to load the crank mechanism. First, the top of the piston is loaded with a force from the gases, which changes over time. This table of values is converted from the GT-Suite software. The engine speed is then assigned to the crank mechanism. Then the dynamic analysis is started, where the duration of analysis is set so that the mechanism makes two rotations. Subsequently, loads in individual connections are exported into an Excel file and after that maximal forces in the connections are applied to individual components.

3.4.2 CONNECTING ROD

The connecting rod is most stressed by tension and compression. It acquires the highest values at the maximum pressure in the cylinder (a few degrees after TDC during a power stroke) and at the beginning of the intake stroke, when the piston is pulled up. Therefore, the control is performed in four different cases. The first case is the control of a small end for tension, when the connecting rod is attached to a big end. The second case is the control of the small end for compression, when the connecting rod is again attached to the big end. The third case is the control of the big end for tension, when the connecting rod is attached to the small end. And the last case is to check the big end for compression, when the connecting rod is again attached to the small end. Both ends are constrained in a cylindrical connection and the individual loads are performed by means of *Bearing loads*. The basic material properties of the individual components are defined. Then we need to create a mesh. It means that the geometry of the model is decomposed into individual elements based on the finite element method. The resulting number of elements affects the accuracy of the calculation as well as the calculation time. Therefore, it is good to choose a compromise between these two parameters. The maximum element size was set to 3mm. Subsequently, a simulation is run, where the stress is calculated according to the von Mises hypothesis. The maximum stress values for the individual cases are shown in the Tab. 17. The Fig. 84 shows the stress distribution when the small end is loaded by tension. The Fig. 85 shows the stress distribution when the small end is loaded by compression. The Fig. 87 shows the stress distribution when the big end is loaded by tension. And the Fig. 86 shows the stress distribution when the big end is loaded by compression.

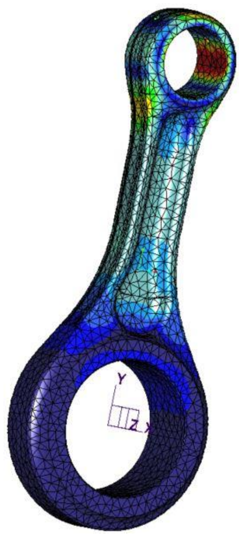


FIG. 84 STRESS DISTRIBITION – TENSION ACTS ON SMALL END

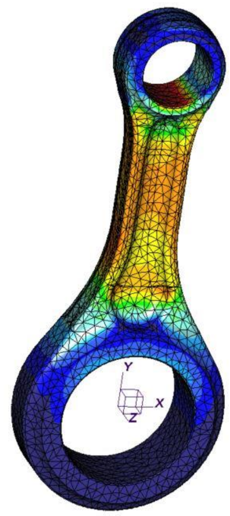


FIG. 85 STRESS DISTRIBITION – COMPRESSION ACTS ON SMALL END

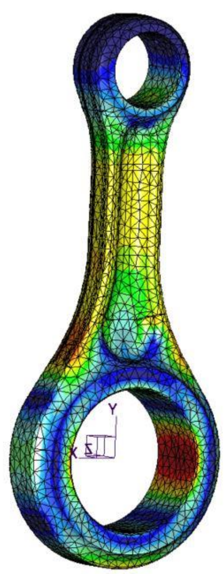


FIG. 87 STRESS DISTRIBUTION – TENSION ACTS ON BIG END

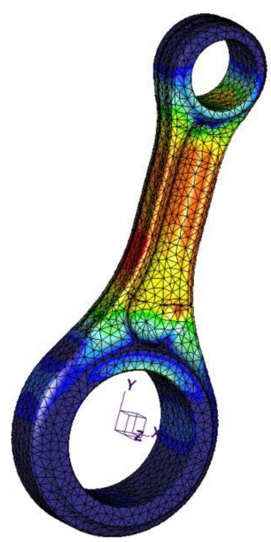


FIG. 86 STRESS DISTRIBUTION – COMPRESSION ACTS ON BIG END

	13,000 RPM	12,000 RPM	9,000 RPM
Small End			
• Tension	461 MPa	392 MPa	214 MPa
• Compression	379 MPa	453 MPa	613 MPa
Big End			
• Tension	442 MPa	377 MPa	208 MPa
• Compression	248 MPa	235 MPa	423 MPa

TAB. 17 MAXIMUM STRESS OF CONNECTING ROD

The result shows that the small and big ends are most stressed by tension at 13,000 revolutions and that the small and big ends are most stressed by compression at 9,000 revolutions. All maximum stress values are permissible, as the mechanical properties of 34CrNiMo6 steel are as follows:

Ultimate tensile strength = 1,210 MPa
Yield strength = 1,084 MPa
Cyclic yield strength = 660 MPa.

3.4.3 CRANKSHAFT

The crankshaft is also stressed by tension and compression. The highest values occur in the same way as for the connecting rod, i.e. at a few degrees behind the TDC during the power stroke and at the beginning of the intake stroke. The control was therefore performed for both types of stress (tension and compression) at three different engine speeds. Because the Creo software sometimes has problems simulating an assembly, the assembly has been simplified to a separate part, where the right crankshaft web has been mirrored and the crank pin has been created between the two cranks using the *Extrude* function. In order to make the simulation as accurate as possible, the *Weighted links* are created in the place of the roller bearings, which are attached to the ground by means of the springs. Furthermore, the front surface of the involute spline for a primary gear is constrained so that it cannot rotate. The force acting on the crank pin is defined using the *Bearing load* function. Then the material is defined, and a mesh is created that has a maximum element size of 3mm. Then a simulation is run, which calculates the stress according to the von Mises hypothesis. The Tab. 18 shows the maximum calculated stress values at different speeds for different types of loads. The Fig. 88 shows the stress distribution in the crankshaft when the crankshaft is loaded by tension. And the Fig. 89 shows the stress distribution in the crankshaft when the crankshaft is loaded by compression.

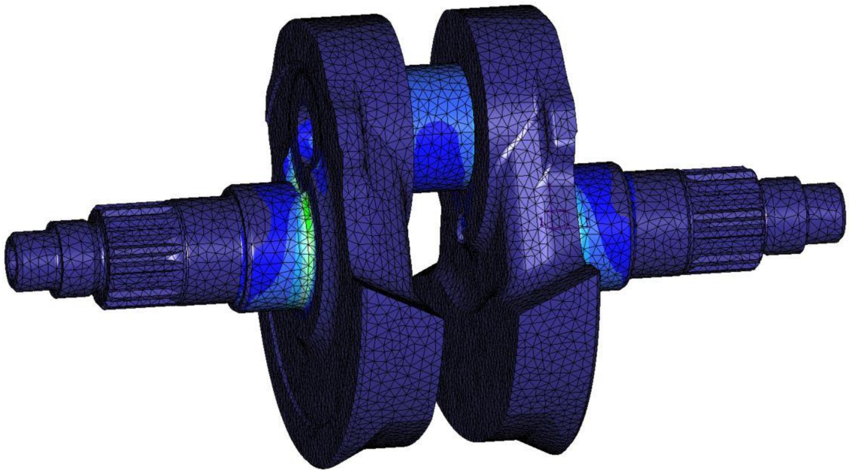


FIG. 88 STRESS DISTRIBUTION – TENSION ACTS ON CRANKSHAFT

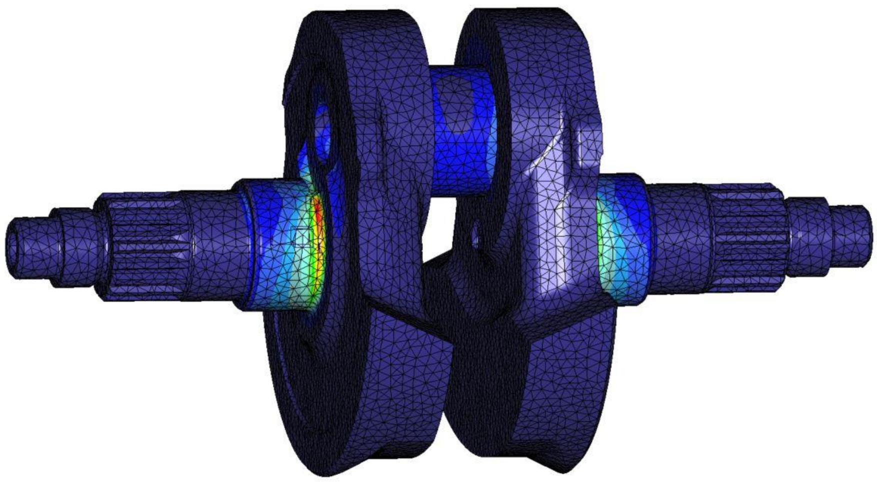


FIG. 89 STRESS DISTRIBUTION – COMPRESSION ACTS ON CRANKSHAFT

	13,000 RPM	12,000 RPM	9,000 RPM
Tension	238 MPa	203 MPa	112 MPa
Compression	130 MPa	181 MPa	301 MPa

TAB. 18 MAXIMUM STRESS OF CRANKSHAFT

The results show that the crank is most stressed by compression at 9,000 rpm and by tension at 13,000 rpm. All maximum stress values are permissible, as the mechanical properties of 42CrMo4 steel are as follows:

Ultimate tensile strength = 1,100 MPa

Yield strength = 980 MPa

Cyclic yield strength = 640 MPa.

3.4.4 PISTON

Because the piston is symmetrical, the piston can be halved and only one half of it can be inspected. Furthermore, the grooves for the wire lock rings are removed because Creo Parametric software had a problem creating the mesh. A cylindrical rotary constraint is created in the pin hole, which prevents the piston from rotating. Furthermore, a constraint is created on the surface in the section plane, which prevents the piston from moving along the axis of the piston pin. Then the piston material is defined, and a mesh is created so that the maximum size of the element is equal to 3mm. In the place of the piston pin, the maximum size of the element is set to 2mm. The piston is loaded by gas pressures and temperature. Therefore, the inspection is performed at three types of loads, namely: load only by the pressure, load only by the temperature, and finally a combined load of pressure and temperature. The gas pressure acts on the entire upper surface of the piston and its value is calculated in a thermodynamic model in the GT-Power software. Then we switch to the *Thermal Mode*, where it is necessary to set the heat transfer using the *Convection condition* function. The following heat transfers are set here: the heat transfer from the burnt gases; the heat transfer from the oil that cools the bottom of the piston; the heat transfer from the piston rings and the heat transfer from the piston pin. The individual heat transfer coefficients and their integral temperatures are either taken from the thermodynamic model from the GT-Suite software or are estimated. Then a thermal simulation is started, where the result is the temperature distribution in the piston. An example of the temperature distribution in the piston is shown in the Fig. 90. This temperature distribution is then transferred to the *Structural mode*, where a thermal stress is created by which the piston is loaded.

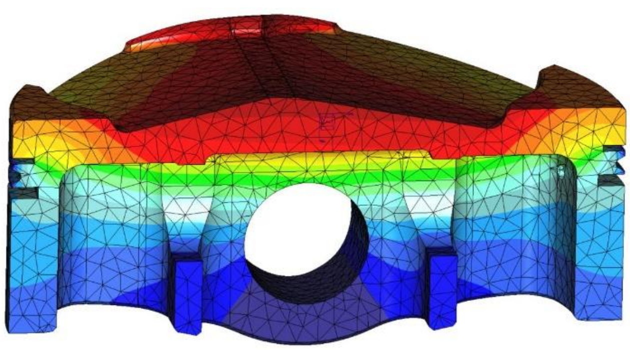


FIG. 90 TEMPERATURE DISTRIBUTION IN THE PISTON

The stress calculation is calculated according to the von Mises hypothesis. The Fig. 91 shows the stress distribution when the piston is loaded by the pressure. The Fig. 92 shows the stress distribution when the piston is loaded by the temperature. The Fig. 93 shows the stress distribution when the piston is loaded by the pressure and temperature. At the edge of the pin hole, where the stress distribution is shown by red colour, the stress value is around 600MPa for load by pressure, around 1,000MPa for load by temperature and around 1,500 MPa for the combination of these two loads. These values are of course unacceptable, and in the future, it is necessary to refocus on this location and try to recalculate it in another software. It is possible that the calculation is wrong at this location because it is on the edge of the boundary condition (constraint). If I leave this place out on purpose, the values of the maximum stresses in the rest of the piston are already acceptable and are shown in the Tab. 19.

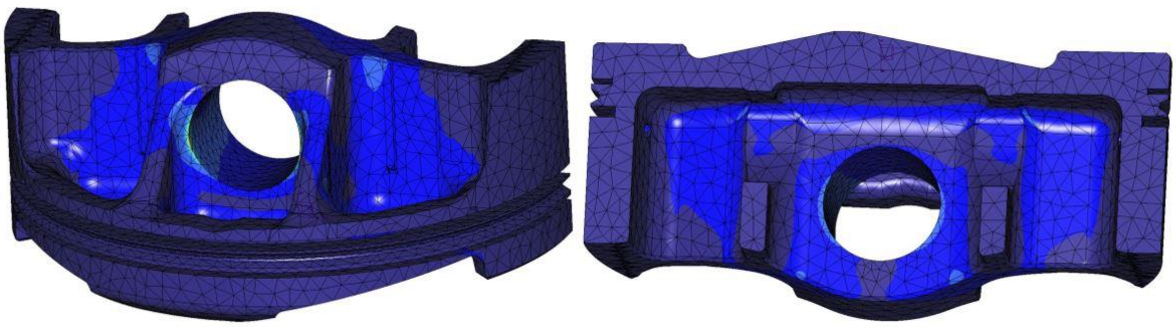


FIG. 91 STRESS DISTRIBUTION IN THE PISTON – PRESSURE LOAD

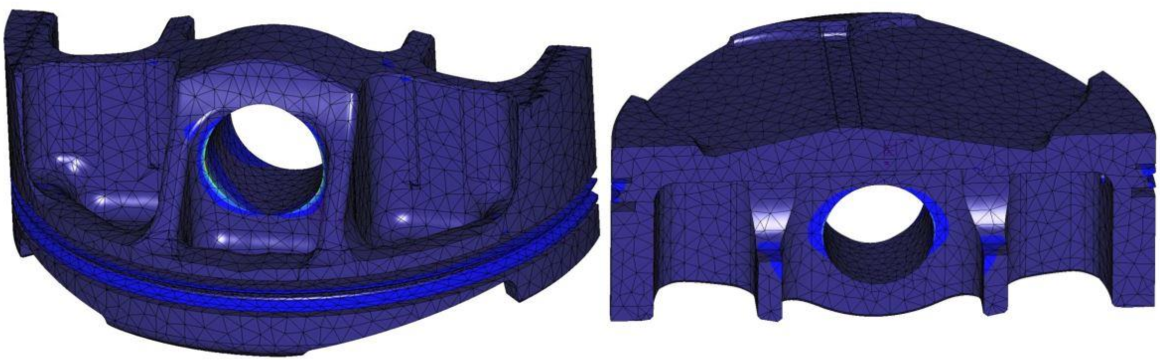


FIG. 92 STRESS DISTRIBUTION IN THE PISTON – TEMPERATURE LOAD

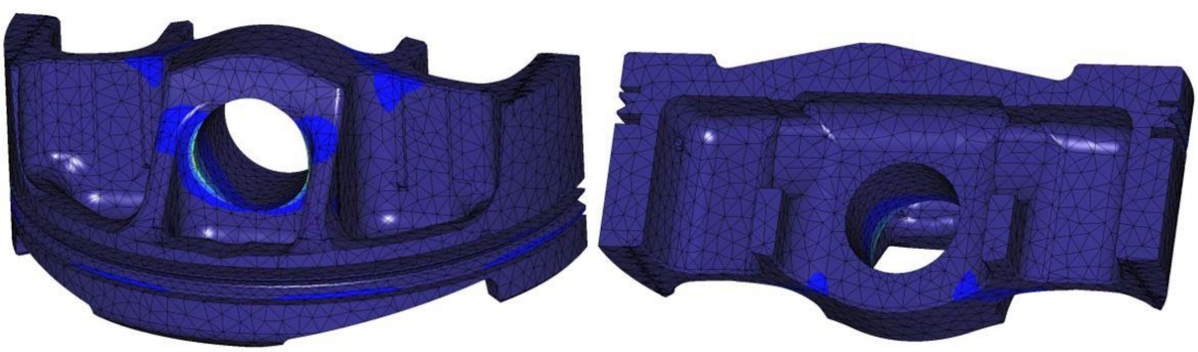


FIG. 93 STRESS DISTRIBUTION IN THE PISTON – COMBINED LOAD

	13,000 RPM	12,000 RPM	9,000 RPM
Pressure	95 MPa	101 MPa	119 MPa
Temperature	163 MPa	166 MPa	154 MPa
Pressure + Temperature	256 MPa	264 MPa	275 MPa

TAB. 19 MAXIMUM STRESS OF PISTON

The results show that the piston is most stressed by pressure at 9,000 rpm, by temperature at 12,000 rpm and by a combination of these two loads at 9,000 rpm. All these maximum stress values are permissible, as the mechanical properties of Al4032 alloy are as follows:

Ultimate tensile strength = 380 MPa

Yield strength = 315 MPa.

3.4.5 PISTON PIN

The pin is most stressed for bending in two cases. The first case occurs during power stroke a few degrees behind the TDC, and the second case occurs at the beginning of the intake stroke. The control is therefore performed for these two different cases at three different engine speeds. In the first case, the pin is mounted by means of a cylindrical rotary constraint in the place where it touches the connecting rod, i.e. in the middle. In the second case, the pin is mounted by means of a cylindrical rotary constraint in the places where it touches the piston, i.e. at the edges. The load values are obtained from the dynamic control in the *Mechanism* mode and are set using the *Bearing load* function. In the first case, the load is applied to the edges of the piston and in the second case, the load is applied to the center of the piston. Then the piston material is defined, and a mesh is created so that the maximum size of the element is equal to 3mm. In the middle of the pin, the maximum size of the element is set to 2mm. Then the simulation is started, and the result is the calculated stress, according to the von Mises hypothesis. The Fig. 94 shows the stress distribution in the piston pin for the first case and the Fig. 95 shows the stress distribution in the piston pin for the second case.

At the edge of the constraint the maximum stress in the first case is about 2,000 MPa and for the second case about 650 MPa. These values are, of course, unacceptable. It is possible that the calculation is wrong at this location because it is on the edge of the boundary condition (constraint). Furthermore, the mistake may be caused by the fact that the edges of the connecting rod and the piston are rounded (or chamfered), but in this case the simulation evaluates this as a sharp edge. In the Fig. 94 and in the Fig. 95, where a section view of the pin is shown, it can be seen that this large stress occurs only on the surface. In any case, it will be necessary to focus on this place in the future and try to recalculate it in another software. If I leave this place out on purpose, the values of the maximum stresses in the rest of the piston are already acceptable and are shown in the Tab. 20.

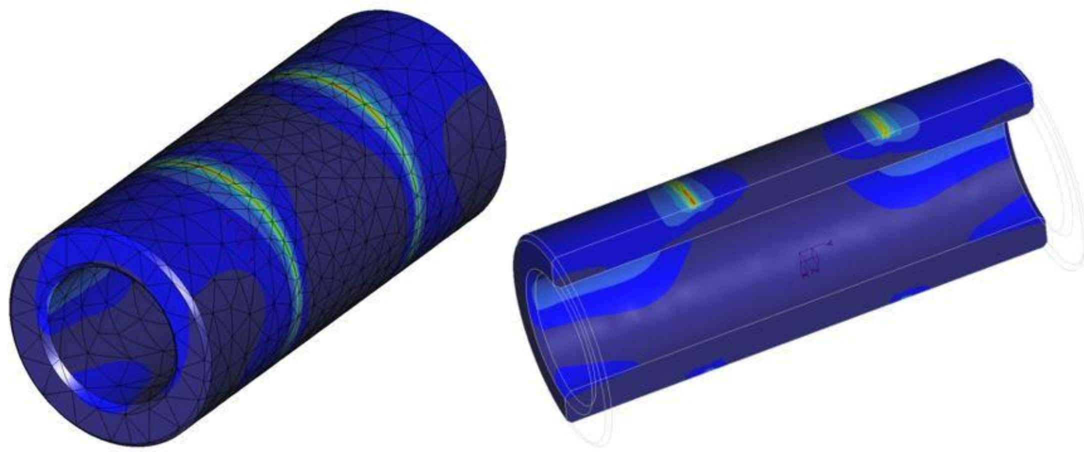


FIG. 94 STRESS DISTRIBUTION IN THE PISTON PIN – 1ST CASE

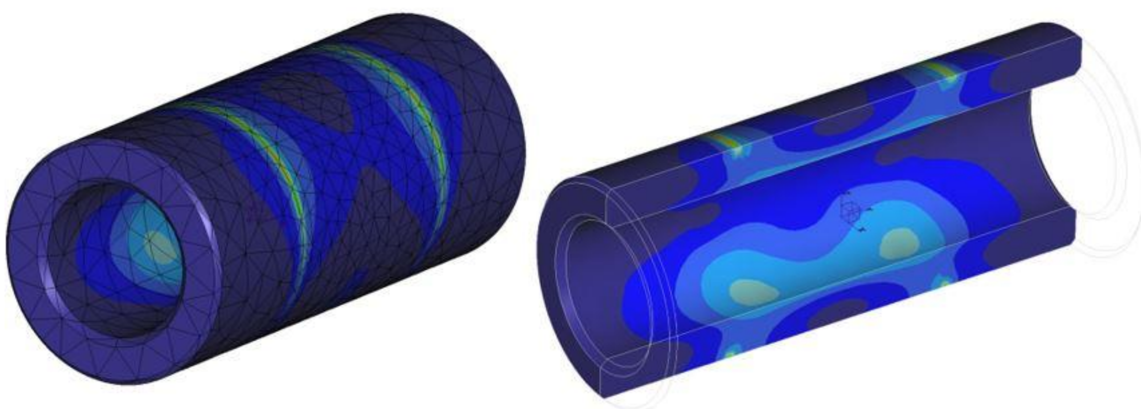


FIG. 95 STRESS DISTRIBUTION IN THE PISTON PIN – 2ND CASE

	13,000 RPM	12,000 RPM	9,000 RPM
1st case	397 MPa	472 MPa	622 MPa
2nd case	211 MPa	179 MPa	96 MPa

TAB. 20 MAXIMUM STRESS IN THE PISTON PIN

It can be seen from the table that in the first case the pin is most stressed at 9,000 revolutions and for the second case it is most stressed at 13,000 revolutions. All these maximum stress values are permissible, as the mechanical properties of 42CrMo4 steel are as follows:

Ultimate tensile strength = 1,100 MPa
Yield strength = 980 MPa
Cyclic yield strength = 640 MPa.

3.5 ADVANTAGES AND DISADVANTAGES OF THIS CONCEPTUAL DESIGN

- The designed engine uses finger followers to open and close the valves. The finger followers have several advantages over valve buckets. They are lighter and therefore create less inertial forces. Another advantage is that they allow the valve to open and close faster because they are not as restricted by contact point movement as the buckets.
- The designed engine is equipped with a DOHC valvetrain, which is more common in small motocross engines than SOHC valvetrain. Its advantage is that, thanks to the easier design of the valve position, a flat combustion chamber can be created, which guarantees better combustion. Another advantage is that with DOHC valvetrain we can achieve higher speeds, as the individual components of the valvetrain have lower weights. Furthermore, the DOHC also allows better setting of valve timing. Its disadvantage is that engines with SOHC valvetrain have a more compact head and consist of fewer components.

- The designed engine has a conventional cylinder, which means that the cylinder offset of the cylinder centerline from the center of the crankshaft is equal to zero. This can be considered a disadvantage, as the greater lateral force acts on the piston than with a cylinder offset, and thus the piston can wear out quickly. This also increases the friction between the piston and the cylinder.
- The designed engine is under-square, which means that it has larger bore than stroke. The advantage of large bore is the use of larger intake valves, so more air is sucked in and thus greater power can be achieved. We can also achieve higher engine speed and therefore more power. At the same time, however, the disadvantage is that the smaller the stroke, the more difficult it is for the air-fuel mixture to mix, and combustion problems can occur.
- Due to the compactness and weight of the engine, the engine includes only one balancer shaft on the right side of the engine. The advantage of this solution is that the balancer shaft has other functions in addition to balancing (cam chain drive, water pump drive). Since the balancing shaft is only on one side, the balancing is not symmetrical, which can be taken as a disadvantage. However, it is a motocross engine where vibrations do not play such a big role and that is why this compromise was chosen.
- As mentioned above, the cam chain that drives the camshafts is driven by a balancer shaft. This means that another gear is inserted between the crankshaft and the camshafts, which can cause various clearances and deviations, which is taken as a disadvantage. However, the advantage is that the engine does not have to be as wide as in the case where the chain is driven directly by the crankshaft, because the sprocket would have to be located on the crankshaft. Another advantage of this design is the fact that the chain can be shorter.
- The engine head is designed so that the intake is located at the rear and the exhaust channels are located at the front. The advantage of this is that the overall design of the motorcycle allows a classic arrangement (concept), which means that the fuel tank is located in front of the rider above the engine and the airbox is located behind the engine. The disadvantage is that if the intake was guided from the front, the engine should theoretically have a more direct intake channels and hydrodynamic losses would be reduced. Another disadvantage may be that when driving, the intake air is affected by the heat radiated by the engine.

- The inclination of the cylinder is related to the location of the intake and exhaust and is also chosen with regard to the centralization of the weights of the entire motorcycle. Therefore, the advantages and disadvantages of the inclination of the cylinder cannot be clearly determined.
- The piston is equipped with only one compression ring, which means that the piston does not have to be so high and is therefore lighter. Following this, the cylinder can also be lower and thus the overall height of the engine will be lower. Another advantage is the reduction of friction between the piston and the cylinder. However, one compression ring is common only on motocross bikes, which may not meet as strict emission standards as other bikes.
- The designed piston has a bridged-box design. The advantage of this design is to increase the rigidity of the piston and thus reduce its stress. As a result, the piston does not have to be so big and robust, which saves weight. However, the disadvantage of this design is the high production cost.
- The design of the engine case provides space for an electric starter. Its advantage is that it allows faster and more reliable starting during the race, it is more comfortable for the rider and it is more suitable for starting in cold conditions. However, its disadvantage is that the addition of an electric starter increases the total weight of the engine.
- This conceptual design assumes that the lubrication system will be common to the entire engine, meaning that one lubrication oil will lubricate and cool all components (crank mechanism, transmission, DOHC valvetrain, etc.). The advantage of this design is to save weight and improve the compactness of the entire engine. The disadvantage of this, however, is that the oil must be changed more often.

4 CONCLUSION

The goal of this diploma thesis was the conceptual design of a four-stroke single-cylinder motocross engine with an engine displacement of up to 250ccm. The thesis was made in the company Ricardo Prague s.r.o.

Before starting to design a motocross engine, it was necessary to get acquainted with motocross engines, which are available on the current market. In the theoretical part, I first gave some information about the motocross itself and the basic differences between the four-stroke and two-stroke engine. Then I analyzed in more detail three current engines of motocross bikes, namely: KTM 250 SX-F, Yamaha YZ250F and Honda CRF250R. At the end of the theoretical part, I compared the basic parameters of these three engines.

Based on the analysis of these three engines, I started with the practical part of the thesis, where I first determined the basic parameters of the designed engine, which include bore, stroke, and compression ratio. The bore is equal to 79mm, the stroke is equal to 51 mm and the compression ratio was equal to 13.95 after all adjustments. The bore and stroke had to be chosen so that the resulting engine displacement did not exceed 250ccm. Then I compiled a thermodynamic model in the GT-Suite software, which was changed several times during the overall design. Then I started creating a conceptual design, where I decided to follow the layout of the engine from KTM 250 SX-F motorcycle. Subsequently, I started modeling individual components in the Creo Parametric software. I started with the crank mechanism, which is considered the heart of the engine. Then it was necessary to create a model of the cylinder, valvetrain, and cylinder head to determine the final compression ratio. Then the balancer shaft was designed. At the end of the construction part, the engine cases, clutch, and transmission were schematically designed. The resulting model of the whole engine is shown in the Fig. 82 and Fig. 83. Ricardo Prague s.r.o. provided me with some useful materials during the construction part and therefore does not wish to publish more detailed information, manufacturing drawings or the 3D model. Then in the Creo Parametric in the *Simulation section* I performed a strength analysis of the crank mechanism using the finite element method, where I focused on the connecting rod, crankshaft, piston, and piston pin. For the piston and the piston pin, unacceptable stress values have been calculated at the edge of the boundary condition and therefore it will be necessary to refocus on these two calculations in the future. The advantages and disadvantages of this conceptual design were described in the final chapter of the practical part.

In conclusion, I would like to say that engine design is a very difficult and time-consuming matter. In practice, several engineers participate in the development and construction of the engine, who use various simulations to tune the individual components to the last detail. Therefore, this design is only conceptual and can be used for further processing. During the processing of this diploma thesis, I gained a lot of new knowledge and experience. Above all, I obtained valuable information in the field of construction of the internal combustion engine.

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