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ÁREA DE INGENIERÍA MECÁNICA

TREN DE POTENCIA PARA UN VEHÍCULO “HIGH EFFICIENCY”

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1. INTRODUCCIÓN

La Shell Eco-Marathon es una competición anual en la cual los participantes diseñan un vehículo con el fin de obtener la máxima eficiencia posible. Los coches son diseñados por diferentes universidades. Cada una tiene su propio equipo, el cual está compuesto únicamente por estudiantes.

La primera competición tuvo lugar en 1939, cuando algunos científicos de Shell hicieron una competición para ver que coche podría hacer más distancia con un galeón como combustible.

En la categoría de gasolina, el record está establecido por el equipo de la universidad "Lycée La Joliverie". Esta universidad francesa ha hecho 3.410km con tan sólo un litro de gasolina.

Figura 1-1: Vehículo usado por el equipo francés "Lycée La Joliverie"

La competición está dividida en dos diferentes categorías. La clase Prototipos, destinada a obtener máxima eficiencia, y diseño urbano. El objetivo de esta última está destinada a diseños más prácticos.

El principal objetivo de este proyecto es el de desarrollar una nueva transmisión de potencia para el vehículo, la cual debe transmitir la potencia del motor a la rueda de atrás. El par es transmitido mediante cadenas, correas, engranajes u otros conceptos. Un embrague es necesario para desconectar la transmisión de la rueda de atrás en la etapa en la cual la transmisión está desconectada.

El coche utiliza un motor de gasolina. Con sólo un litro de combustible el objetivo es hacer 600 km o más. La principal diferencia de este modelo con respecto al anterior, es el uso de un motor más potente (el par máximo de este nuevo es 11 Nm, del modelo antiguo 1,6 Nm) para simplificar la transmisión, reduciendo el número de etapas.

2. ESTADO DEL ARTE.

2.1. Correas de transmisión.

Una cadena de transmisión es un material flexible usado para unir dos o más ruedas. Transmiten potencia de un eje a otro.

Al principio, cuero curtido al cromo era utilizado como material para las correas. Este material presentaba una gran desventaja: Degradación debido a la fricción y al ruido. Hoy en día, son totalmente sintéticas (tienen una capa de poliéster o poliamida).

Este tipo de transmisión presenta varias ventajas: Absorben vibraciones y ruidos, el mantenimiento es escaso y no es necesario lubricarlas.

Por otro lado, presentan una importante desventaja: Debido al deslizamiento, la relación de transmisión no es constante a menos que se utilicen correas dentadas.

2.2. Cadenas de transmisión.

Las cadenas de transmisión son un mecanismo utilizado para transmitir potencia entre ruedas dentadas. La potencia es transmitida mediante la cadena

Los materiales más comunes son aceros (templado o bonificado).

La relación de transmisión en este tipo de mecanismo es constante, debido a que no hay deslizamiento. Este no debe ser mayor de 8.

2.3. Correas vs. Cadenas.

Las cadenas son más fuertes que las correas, pero su mayor inercia aumenta la inercia. También son más estrechas. También son más flexibles

Los materiales utilizados para cadenas son principalmente aceros, mientras que para correas se utilizan plásticos.

En cuanto a las aplicaciones, las cadenas de transmisión suelen utilizarse para mover cosas verticalmente, como por ejemplo un elevador de cangilones. En cambio, las correas se utilizan para movimientos horizontales, como las cintas transportadoras.

La relación de transmisión de las cadenas es siempre constante. A menos que se utilicen correas dentadas, ésta no es constante.

La eficiencia máxima de las cadenas es 98%. En cambio, la eficiencia máxima de las correas es de aproximadamente un 95%.

Por todas estas razones, las cadenas de transmisión son mejores para este tipo de aplicaciones que las correas.

2.4. Engranajes.

Un engranaje es un mecanismo utilizado para transmitir potencia en una máquina de un componente a otro.
La eficiencia de este mecanismo es de aproximadamente un 98%.

Los materiales más utilizados para engranajes son: Acero, bronce…o incluso hoy en día es posible utilizar impresoras 3D para fabricar engranajes.

2.5. Cadenas vs. Engranajes.

La principal razón del uso de engranajes es la geometría. Considerando una distancia entre centros de $a = 422{,}3mm$, una relación de transmisión de $i = 6{,}6$, y el diámetro de la rueda del coche (valor fijo) $d = 500mm$ and y usando estas dos ecuaciones, el resultado final no tiene sentido:

$i = 6{,}6 = \frac{r_2}{r_1}$ and $r_1 + r_2 = 422{,}3mm$

Resolviendo este sistema de ecuaciones, los resultados son: $\boldsymbol{r_2} \approx \mathbf{366{,}76}\boldsymbol{mm}$ and $r_1 \approx 55{,}57\ mm$.

El radio de la rueda conducida ($\boldsymbol{r_2}$) es mayor que el radio de la rueda del coche ($\boldsymbol{r_c} = \mathbf{250}\boldsymbol{mm}$.).

Usando cadenas, es también posible cambiar la distancia entre centros y mantener las ruedas dentadas. Esta situación no es posible con engranajes.

3. DESARROLLO DEL SISTEMA DE TRANSMISIÓN.

3.1. Características del motor.

El motor usado para este proyecto es Honda GX200. A este motor se le han hecho varias modificaciones. Por ejemplo, el cilindro utilizado corresponde al del motor GX120, o la capacidad del mismo. ésta ha sido reducida de 196 ccm a 153 ccm.

En la tabla 3-1 se pueden ver los valores de algunos parámetros importantes del motor. Como sigue todavía en fase de pruebas, estos valores no son totalmente definitivos.

Parámetros	**Valor**
Potencia máxima [kW]	3,5
Máxima velocidad de rotación [rpm]	3500
Mínima velocidad de rotación [rpm]	1800
Máximo par a 2500 1/min [Nm]	11

Tabla 3-1: Características principales del motor.

3.2. Relación entre la velocidad del motor y la del coche.

En la tabla 3-2 es posible ver las máximas y mínimas velocidades del coche y del motor:

Parámetros	**Valor**	**Valor**
Máxima velocidad del coche	50 [km/h]	13,89 [m/s]
Máxima velocidad de rotación del motor	3500 [rpm]	58,33 [rev/s]
Máxima velocidad de rotación de la rueda del coche	530,4 [rpm]	8,84 [rev/s]
Relación de transmisión (i)	6,6	
Mínima velocidad del coche	25,71 [km/h]	7,14 [m/s]
Mínima velocidad de rotación del motor	1800 [rpm]	30 [rev/s]
Mínima velocidad de rotación de la rueda del coche	272,98 [rpm]	4,55 [rev/s]

Tabla 3-2: Máximas y mínimas velocidades del coche y del motor.

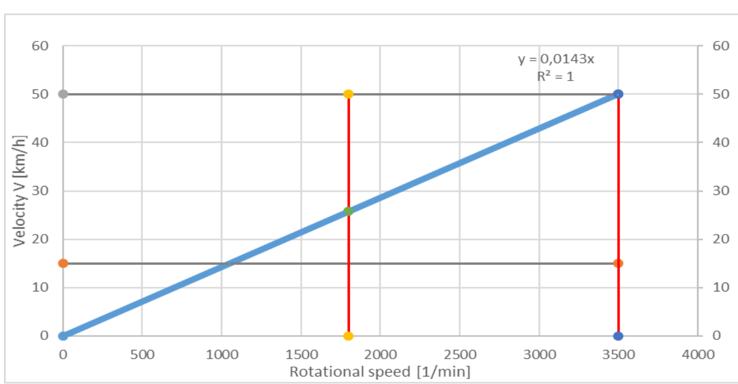

Figura 3-1: Relación entre la velocidad del coche y del motor.

En la figura 3-1 se puede ver la relación que existe entre las velocidades del coche y del motor. El motor, en reposo, comienza a rotar hasta alcanzar su velocidad máxima: 3500 rpm. A esas revoluciones, la velocidad del coche es máxima: 50 km/h. Una vez que el coche alcanza esa velocidad, el motor se apaga y comienza a descender su velocidad de rotación.

El punto verde (25,71 km/h) representa la velocidad del vehículo cuando la velocidad del motor es mínima. Pero el coche no ha descendido hasta su mínima velocidad, así que en esa fase el embrague no está sincronizado (simplemente desliza). Durante esta etapa se producen pérdidas. Si éste fenómeno sólo se produjera una vez, las pérdidas no serían muy altas. El problema es que se produce 30 veces en un ciclo, así que las pérdidas totales son altas.

Una posible solución es aumentar la mínima velocidad del coche, o disminuir la mínima velocidad rotacional del motor.

3.3. Espacio destinado al sistema de transmisión.

En las Figuras X-Y e X-Y se muestra el espacio destinado en el vehículo (con las principales dimensiones) al tren de potencia.

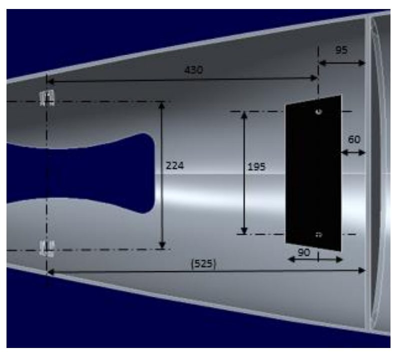

Figura 3-2: Vista en planta del espacio destinado al tren de potencia

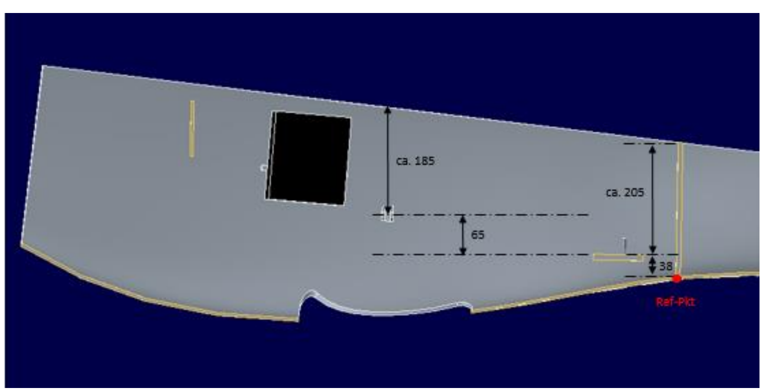

Figura 3-3: Vista lateral del espacio destinada al tren de potencia.

3.4. Sistema de frenado.

Una de las normales de la competición sobre el sistema de frenado es la siguiente: "El vehículo se colocará en un plano inclinado con una pendiente del 20% con el conductor dentro. Cada uno de los sistemas de frenado se activarán de forma independiente. Cada sistema debe mantener el vehículo inmóvil".

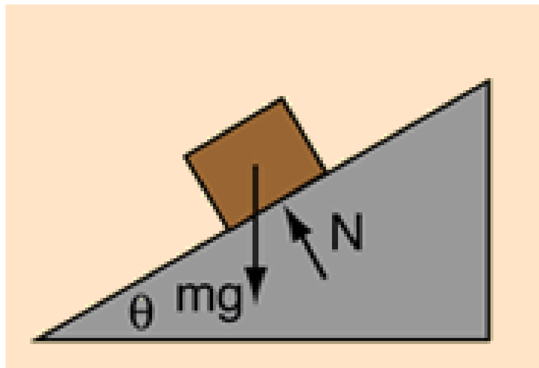

Figura 3-4: Diagrama de fuerzas en un plano inclinado.

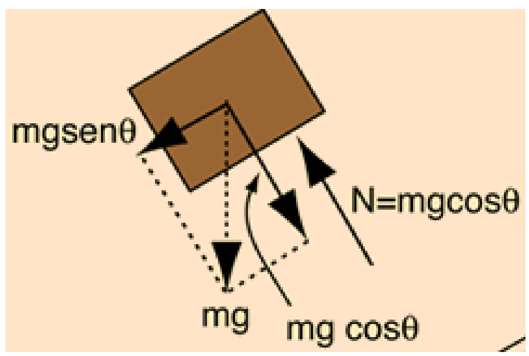

Figura 3-5: Descomposición de las fuerzas en un plano inclinado.

$\theta = \tan^{-1}\left(\frac{20}{100}\right) = 11{,}3$
$F = m \times g = 100 \times 9{,}8 = 980N$
$F_X = F \times \sin(\theta) = 980 \times \sin(11{,}3) = 192N$
$F_Y = F \times \sin(\theta) = 980 \times \cos(11{,}3) = 961N$ = N

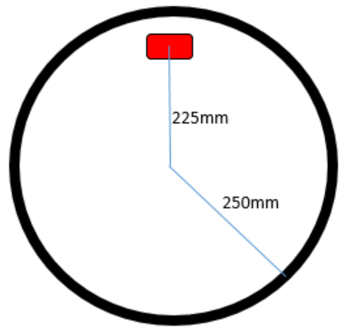

Figura 3-6: Shoe location on the wheel.

$$F_{BR} = 192N \times \frac{250}{225} = 213{,}33\ N$$

The friction static coefficient μ is 0,5:

$$\frac{213.33}{2} = \mu \times N_{BR}$$

$$\boldsymbol{N_{BR} = 213{,}33\ N}$$

3.5. Descripción de las posibles soluciones.

En la tabla 3-2 las diferentes alternativas son descritas. Es posible ver el diámetro de las ruedas dentadas, el número de dientes, el peso, etc…

Alt.	**D1 [mm]**	**Z1**	**D2 [mm]**	**Z2**	**Tipo de cadena**	**Peso [kg]**	**FX [N]**	**FY [N]**
1	60,89	20	400,25	132	06-B	1,03	334,86	135,7
2	66,93	22	439,66	145	06-B	1,43	300,22	133,85
3	49,07	12	323,49	80	08-B	0,5	426,38	138,6
4	61,08	15	400,28	99	08-B	1,4	333,55	135,92
5	61,34	12	404,36	80	10-B	1,84	332,67	134,0
6	**63,91**	**21**	**418,44**	**138**	**06-B**	**1,21**	**317,4**	**130,66**

Tabla 3-3: Características principales de las diferentes alternativas.

3.6. Comparación de las diferentes alternativas.

Según diferentes criterios, como por ejemplo la resistencia, peso, espacio utilizado, o dificultad de montaje, las 6 alternativas son comparadas en la tabla X-Y:

	Alt. 1	**Alt. 2**	**Alt.3**	**Alt.4**	**Alt.5**	**Alt.6**
Peso	+	-	++	-	--	**0**
Espacio utilizado	0	-	+	0	0	**+**
Resistencia	-	++	--	-	-	**++**
Facilidad de ensamblaje	+	-	+	+	-	**+**

Tabla 3-4: Comparación de las posibles soluciones.

La escala de puntuación es la siguiente: ++ (muy bien), + (bien), 0 (neutro), - (mal), -- (muy mal).

Puede observarse que las mejores alternativas son la 3 y la 6. La alternativa 3 tiene un problema, la resistencia. Ésta no es suficiente para las fuerzas de la cadena sobre las ruedas dentadas. Por lo tanto, la alternativa escogida es el número 6.

4. DESCRIPCIÓN DETALLADA DE LA SOLUCIÓN FINAL.

El material de rueda dentada conductora es Acero AISI 1010. El material de la conducida es aluminio Al6061. La razón de usar aluminio en vez de acero es fundamentalmente el peso. Debido a que la densidad del acero es mucho mayor que la del aluminio, el peso de la rueda se incrementaría demasiado. Como se trata de un prototipo "High Efficiency", reducir el peso al máximo resulta una buena práctica.

Ambas ruedas dentadas presentan 4 agujeros Ø6 mm. En dichos agujeros irán tornillos cuya calidad es 3.6.

En las Tablas 4-1 y 4-2 se pueden ver los principales parámetros de las ruedas dentadas.

Parámetros	**Valor**
Pitch [mm]	9,525
Teeth	21
Pitch diameter [mm]	63,91

Tabla 4-1: Principales parámetros de la rueda dentada pequeña.

Parámetros	**Valor**
Pitch [mm]	9,525
Teeth	138
Pitch diameter [mm]	418,44

Tabla 4-2: Principales parámetros de la rueda dentada grande.

La cadena está estandarizada de acuerdo con DIN 8187. El fabricante es IWIS. En las tablas es posible ver algunos parámetros importantes de la misma.

Parámetros	**Valor**
d_1 [mm]	6,35
b_1 [mm]	5,72
X	176
Length [mm]	1676,4
Total weight [kg]	0,687

Tabla 4-3: Principales parámetros de la cadena.

4.1. Bastidor

El material del bastidor es Al 7075-T6. La razón de usar aluminio en vez de acero es la misma que en el caso de la rueda dentada grande: El peso. Utilizando un bastidor de aluminio el peso actual (sobre 2,5 kg) se incrementaría casi hasta 7 kg. Además, con esta aleación (Resistencia última aproximada: 600 MPa), el bastidor no va a romper o sufrir importantes daños.

El bastidor va fijo al chasis del coche en 4 puntos mediante tornillos. Tiene 3 partes principales, las cuales están fijas entre sí también mediante tornillos.

En la Figura 4-1 las 3 partes del bastidor están representadas.

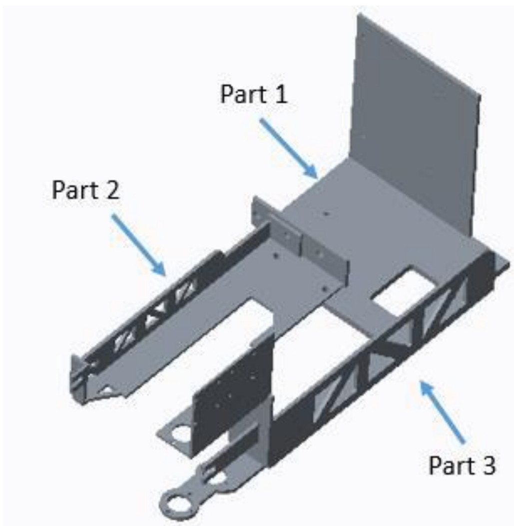

Figura 4-1: Partes del bastidor.

En la Figura 4-2 se muestra un ensamblaje del tren de potencia, con todos sus elementos: Entre ellos están el embrague, cadena de transmisión, motor….

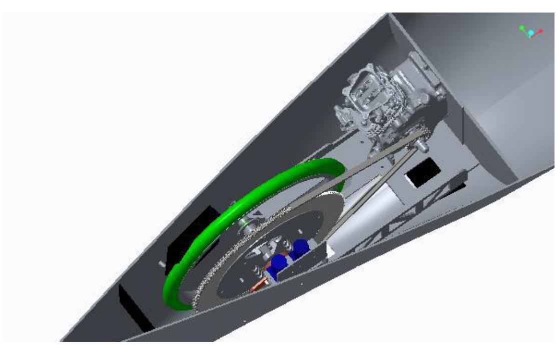

Figura 4-2: Ensamblaje de todo el tren de potencia.

5. CONCLUSIONES Y MEJORAS FUTURAS.

Debido a las ventajas que ofrecen las cadenas de transmisión con respecto a las correas y engranajes, éstas han sido consideradas para llevar a cabo el proyecto.

Todas las alternativas han sido comparadas de acuerdo a diferentes criterios, como por ejemplo el peso y la resistencia.

El coeficiente de seguridad en todos los cálculos es siempre mayor de 1,5.

La solución final es más simple y ligera que la anterior. La nueva tiene sólo una nueva etapa, teniendo la vieja 3. Considerando que cada etapa tiene un 98% de eficiencia, esta nueva solución es al menos un 4% más eficiente. Además, es más fácil de reparar durante la carrera.

Para reducir las pérdidas comentadas en el punto 3.2, varias soluciones son posibles: Aumentar la velocidad mínima del vehículo, o disminuir la velocidad de rotación del motor.

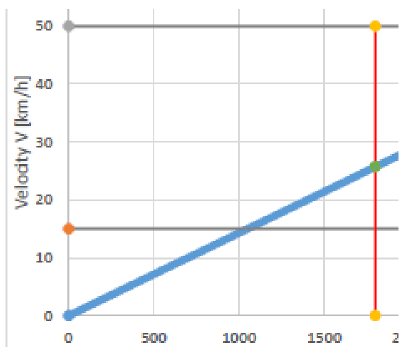

Figura 5-1: Zona en la que se producen pérdidas debido a la desincronización del embrague.

Hochschule Karlsruhe
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Fakultät für Maschinenbau und Mechatronik

Power Train for Ultra Efficient Prototype Vehicle

Bachelor Thesis (B.Eng.)

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Karlsruhe, 11.07.2016

Statutory Declaration

I declare that I have authored this thesis independently, that I have not used other than the declared sources / resources, and that I have explicitly marked all material which has been quoted either literally or by content from the used sources.

Karlsruhe, 11.07.2016

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Jose Luis Areces Fernandez

Appreciation

I would first like to thank the Hochschule Karlsruhe and especially my coordinator Prof. Volker Hirsch, the opportunity they gave me to develop this interesting project which opens me a lot of doors in the automotive industry. I do not also forget the High-Efficiency-Karlsruhe team, because they always helped me when I needed. And of course the Polytechnic School of Engineering of Gijón for the training they have given me over the years.

I also want to dedicate this project to my family and particularly to my parents, Pepe and July, for the education they have given me and their good advice in all aspects, and my sister Lucia. She is the person who motivated me to study this engineering degree. She always supports me (both good and bad times) and knows she can always count on me.

Last but not least, my friends, who are always there, both for better and worse.

Abstract

Power Train for Ultra Efficient Prototype Vehicle

This bachelor thesis was developed in collaboration with the High-Efficiency-Team of the Hochschule Karlsruhe. The goal of this thesis is to develop a new power train for the prototype vehicle WIM II (Weniger ist Mehr -Less is More-) in High-Efficiency-Team. The report includes various solutions and results for the power train and comparing them.

The vehicle WIM II is used in Shell-Eco-Marathon in Europe by the High-Efficiency-Team. The aim of this competition is to design a vehicle that drives the maximum distance with one litre of fuel, in order to reduce pollution. Various vehicles compete against each other in a variety of classes. For instance, gasoline, diesel or alternative energy are used. The vehicle developed by the High-Efficiency-Karlsruhe team participates in the gasoline competition.

In order to reach the goals, it is important to reduce losses and improve the efficiency of the whole car, including the drive train. For this purpose, weight should be reduced as much as possible. This means that a lightweight construction is necessary in order to reduce the weight, and therefore consumption.

Some different proposals have been analysed and the most effective one is detailed and described in order to improve the next vehicle for the Shell Eco-Marathon.

Kurzfassung

Antriebsstranges für ein ultraeffizientes Prototypenfahrzeug.

Die vorliegende Bachelor-Thesis wurde im High-Efficiency-Team der Hochschule Karlsruhe erstellt. Das Ziel der Arbeit ist es, einen neuen Antriebsstrang für das Prototypfahrzeug WIM II, welches das High-Efficiency-Team für den Eco-Marathon der Fa. Shell aufbaut und optimiert, zu entwickeln. Alle notwendigen Komponenten dieses Antriebsstrangs sind auf einem gemeinsamen Rahmen befestigt, so dass ein selbsttragendes Modul entsteht.

Das Fahrzeug WIM II wird vom High-Efficiency-Team im Shell-Eco-Marathon eingesetzt. Das Ziel dieses Wettbewerbs ist es, ein Fahrzeug zu entwickeln, das eine maximale Distanz mit nur einem Liter Kraftstoff zurücklegt. Verschiedene Fahrzeuge konkurrieren miteinander in unterschiedlichen Klassen, wie Benzinantrieb, Dieselantrieb oder Antriebe mit alternativen Energien. Das Fahrzeug des Teams HEK wird durch einen Benzinmotor angetrieben.

Es ist wichtig, die Verluste des Antriebsstrangs zu minimieren und diesen so hinsichtlich der Effizienz zu verbessern. Leichtbau steht dabei im Vordergrund. Das Gewicht soll so weit wie möglich reduziert werden.

Es werden verschiedene Lösungsansätze analysiert und die effizientesten Konzepte detailliert beschrieben. Das aussichtsreichste Konzept soll für den nächsten Shell Eco-Marathon aufgebaut werden.

CONTENTS

LIST OF ABREVIATIONS

i	Gear ratio
f1	Service Factor
f2	Teeth Factor
P	Pitch
F_B	breaking load
q	load per meter
ISO	International Organization for Standardization
DIN	Deutsches Institut für Normung (German Institute for Standardization)
b1	Roller width
d_R	Roller diameter
e	Distance
g	Max. height of the plate
A	Pressed surface
X	Number of links
a	distance between centres
Z_1	Number of teeth of the small sprocket
Z_2	Number of teeth of the big sprocket
P_D	Corrected Power
n	Rotational Speed
V	Velocity
b	Width
S_B	Static breaking strength
S_D	Dynamic fracture resistance
F	Force
F_f	Centrifugal force
F_d	Dynamic force
P	Power
F_r	Radial force
F_a	Axial force
M	Torque
Dp	Pitch diameter
De	External diameter
Di	Internal diameter
m	Module
α	Pressure angle
β	Helix angle
L	Nominal Lifetime
Lf	Nominal lifetime in hours
P'	Dynamic load
P0	Static load
C	Basic dynamic load rating
C_0	Basic static load rating
M_x	Bending moment in X-plane
M_y	Bending moment in Y-plane
M_t	Torsional moment
d_{min}	Minimum diameter
POM	Polyoxymethylen
ρ	Density
F_x	Force in X-direction
F_Y	Force in Y direction

F_{BR}	Braking Force
N_{BR}	Normal
μ	Friction coefficient
B_1	Tooth width
t	Pitch angle
d_f	Root diameter
d_a	Tip diameter
d_s	Diameter of free rotation
r_1	Roller bed radius
r_2	Tooth flank radius
c	Chamfer
χ	Roller bed angle
k	Tooth height above pitch polygon
B_Y	Final width
K_V	Dynamic factor
K_A	Application factor
K_1	Teeth quality factor
K_2	Straight teeth factor
Y_{SA}	Stress correction factor
Y_{FA}	Teeth form factor
SF	Safety factor
σ_y	Yield Strength
σ_{max}	Maximum Tensile Strength
n_f	Safety screw factor
σ	Tensile Strength
τ_{max}	Maximum Shear Stress
τ	Shear Stress
σ_F	Local Tooth root stress
σ_{F0}	Tooth root stress
ε	Contact ratio
p_{zul}	Allowable surface pressure
p_m	Surface pressure
K_λ	Load distribution factor
l'	Supporting key forms
φ	Contributing factor considering wear during the use of several matching feathers

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1. INTRODUCTION

1.1. Shell Eco Marathon

The **Shell Eco-Marathon** is an annual competition sponsored by Shell, in which participants design special vehicles in order to achieve the highest possible fuel efficiency.
The cars are designed by different Universities. Each University has its own team, composed only for students and a faculty advisor.

In 1939, some scientists of the Shell company in USA made a competition in order to see who could drive their own car with only one gallon of fuel. The efficiency of the winner was 0,0473l/km. Every year the challenge was repeated, and the results were highly improved (1).

The European record using a combustion engine was set in 2004 by the team from Lycée La Joliverie (France) at 3.410 km on the equivalent of a single litre of fuel (0,0003l/km = 0,03l/100km) (2).

Hydrogen can be used as fuel too. In 2005 a team from Zurich (Switzerland) achieved 3.836km by using one litre of fuel (0,026l/100km) (3).

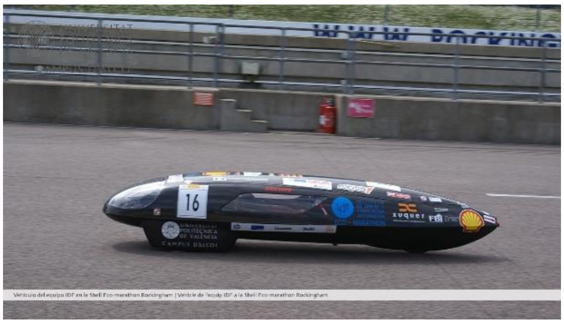


Figure 1-1: Vehicle used by the Spanish team from Valencia (4).

Figure 1-2: Vehicle used by the French team from “Lycée La Joliverie” (3).

In 2013 a new record of 3.100 km was set by a team from Toulouse (France) by using only a single litre of ethanol (0,032l/100km) (4).

In 2010, a Spanish University (Valencia) set up the record by using diesel as a fuel. The distance achieved was 1396km. (0,0716l/100km) (5).

This year, 2016 Shell-Eco Marathon takes place in London, UK, from 30th June to 3rd July.

The competition is split into two categories. The Prototype class focuses on maximum efficiency, while passenger comfort takes a back seat. The Urban Concept class encourages more practical designs.
According to the energy source used (solar cells, fuel cells, diesel…) the competition is also splitted in several categories.

The main **objective** is to travel the furthest on the equivalent of one litre of fuel, and the main concept is not speed, but efficiency. The efficiency of the cars has been improved year by year. The best one of the first competition was 4,17l/100km in 1939. Nowadays, teams (using, for instance, gasoline as a fuel) try to obtain a consumption around 0,03l/100km.

The competition inspires the energies that will be used in the future to turn the vision of sustainable mobility into reality.

Figure 1-3: Some eco-vehicles ready for start the race (6).

The main **rules** of the Shell Eco Marathon are:

Accessing the competition:

- A Team Manager, a Driver and a Faculty Advisor have to be designated.
- The Team Manager must be a student member of the team and enrolled in the institution.
- All the information will be sent to the Team Manager, who can be only responsible for one vehicle.

The **dimensions** of the vehicle:

- Maximum height must be less than 100 cm
- Maximum width must not exceed 130 cm
- Maximum length must be less than 350 cm
- Maximum weight (without the driver) must be less than 140 kg
- The wheelbase must be at least 100 cm
- The ratio of maximum height divided by track width muss be less than 1,25
- The track width must be at least 50 cm, measured where the tyres of the outermost wheels touch the ground.

The main purpose of this project is to design the drive train. The other objectives of the project will be described in more detailed in Chapter 1.2.
Regarding each design, following criteria must be fulfilled:

- All parts of the drive train, including fuel tank, hydrogen system components, etc. must be within the confines of the body cover.
- All objects in the vehicle must be securely mounted. Bungee cords or other elastic material are not permitted for securing heavy objects like batteries.
- All vehicles (including Prototypes) must be fully covered. Open top vehicles are not allowed. Vehicles that look like bicycles, tricycles or wheelchairs are not acceptable.
- A permanent and rigid Bulkhead must completely separate the vehicle's propulsion and energy storage systems from the driver's compartment.
- The bulkhead must effectively seal the driver's compartment from the propulsion and fuel system.
- The bulkhead must prevent manual access to the engine/energy compartment by the driver.
- All vehicle propulsion must be achieved only through the friction between the wheels and the road.
- All vehicles with internal combustion engines must be equipped with a clutch system.
- For centrifugal/automatic clutches the starter motor speed must always be below the engagement speed of the clutch.
- For manual clutches the starter motor must not be operable with the clutch engaged. An interlock is required to facilitate this functionality.
- The installation of effective transmission chain or belt guard(s) is mandatory.

Another important rule to take into account is that the average speed of the cars should be at least 23 km/h over a distance of 16 km (7).

1.2. Objectives of the project

The main objective of this project is the development of a new drive train module for a vehicle. This drive train has to transmit the rotational power of the motor to the rear wheel. The torque should be transmitted by using chains, gear wheels, belts or other concepts. A jaw clutch at the rear wheel shaft is needed to disconnect the drive train from the rear wheel during rolling phase. It must be integrated into the design. The drive train module has to be designed considering a very low weight and minimum friction. In addition, a friction clutch, that should replace the jaw clutch in future, must be integrated into the design. The car will compete in the Shell-Eco Marathon in 2017.

The propulsion of the car will be made by using a combustion engine and gasoline as fuel. With only one liter, the main goal is to drive the car along a distance of 600 km or more (0,167 l/100km). In other words, the objective is the design of a drive train in order to achieve the highest possible fuel efficiency.

The main difference with the previous model is the use a powerful engine. For instance, the maximum torque of the old motor is 1,6 Nm. The maximum torque of the new engine is around 11 Nm. This new engine is used in order to simplify the drive train and reduce the number of stages (with deals with reducing energetic losses).

2. STATE OF THE ART

2.1. Power Train

There are a lot of different mechanism which can be used in order to achieve the power transmission: Belt drives, chain drives, gear wheels…. They have different characteristics, like the materials used or the geometry, but their purpose is the same. Depending of the development application, it will be used a certain mechanism that better fulfils the requirements of it.

2.1.1. Belt drives

A belt drive is a flexible material used to join two or more wheels that carry out a rotational movement. Belt drives transmit power from one shaft to other shaft, most of them in parallel, or track a relative movement. Belts are used also in pulleys. The efficiency of the belt drives is very high, around 95% (8).

During the industrialization (it began with the invention of the steam engine, in the 18th Century), chrome tanned leather belt drives were very important. When the electric motor was introduced, leather belt drives lost its high importance, because they had a lot of disadvantages, such as degradation due to friction or noise. The invention of the polyamide in the Second World War, made possible to create a flat belt with stable characteristics (9).

Nowadays, chains are fully synthetic. They have a layer of polyester or polyamide (very strong), and some other wear, oil and blubber resistance layers, made by NBR (elastomer), which provide a high degree of uniformity and friction between the belt and the pulley (9).

This kind of transmission is called also flexible transmission, because it absorbs vibrations and shocks. They are the cheapest way for power transmission between shafts that are not axially aligned. Their maintenance is low, and lubrication is no needed. Due to slip and stretch, the gear ratio **(i)** is not constant unless toothed belts are used (8).

Figure 2-1: Transmission belt drives (10).

2.1.2. Chain drives

A chain drive is a mechanism used in order to transmit power between toothed wheels from one place to another. The power is transmitted using a roller chain, also known as a sprocket gear, with the teeth of the gear meshing with the holes in the links of the chain. The efficiency of the chain drives is very high, around 98% (11).

The first sketches about chains dated from the 3rd Century BC, described by a Greek engineer. (12) Around 16th Century, Leonardo Da Vinci was the first man who made the first designs of a flat-link chain (the first steel chain) (13).

One of the most important parameters is the gear ratio **(i)**. This gear ratio should not be greater than 8. As there is not slipping, **i** is constant.

The common materials used for chain drives are hardened or bonus steel. The wear of the joints produces a permanent elongation in the chain, which may be up to about 3% (13).

Figure 2-2: Single, double, triple roller chain (14).

2.1.3. Chain vs Belts

Chain drives may be stronger than belts, but their greater mass increases the drive train inertia.

The material of chain and belt drives is also different. The first ones are made of metal (steel), while materials used for belts are often rubber or plastic (15).

In order to vary the gear ratio, chain drives are better than belt drives. Chains are usually narrower than belts, and this can make easier to shift them to smaller or larger gears.

Both types of mechanisms can be used to move objects. On the one side, chains are often used to move things vertically by holding them in frames, like a bucket elevator. On the other side, belts are good at moving things horizontally, such as conveyor belts (12).

It is common for some systems that chains and belts are used in combination; for example, the rollers that drive conveyor belts are themselves often driven by drive chains.

The efficiency of chain drives is higher than the efficiency of belt drives (98% vs. 95%).

The gear ratio of chain drives is always constant. Unless toothed belts are used, the gear ratio of belt drives is not constant (due to slip and stretch) (16).

For all of these reasons, chain drives are better than belt drives, for this application.

2.1.4. Gear drives

A gear is a mechanism used in order to transmit power in a machine from one component to another component. The circular movement of the gears, using toothed wheels, allow the change in rotational speed or torque. The big wheel is called crown, and the small one pinion. The efficiency of this mechanism is very high, around 98%. The cog can be internal or external (17).

Gears are used for more than 4000 years. The first example of them come from China. In Europe, the earliest one was circa CE 50. Its origin dates back to the Greek mechanics of the Alexandrian school, in the 3rd Century BCE. They were developed by Archimedes, who design also the worm screw (18).

The common materials are: steel (tempered steel, casting steel, nitriding steel, case hardening steel…), bronze, iron or even nowadays it is possible to use plastic (with a 3D printer) as a material for gears (19).

The gear ratio (i) is also important for gear drives. Changing the number of teeth, it is possible to change the output speed:

$$\mathrm{i} = \frac{\mathrm{n_2}}{\mathrm{n_2}} = \frac{\mathrm{Z_2}}{\mathrm{Z_1}}$$

Equation 2-1.

2.2. GEAR AND CHAIN DRIVES

2.2.1. Chain drives

Chain drives are often used in bicycles or motorcycles. Other applications of this mechanism are lifting or dragging objects. There are different kind of chain drives, for instance, single, double or triple roller chain, or inverted chains (20).

Roller chains: Their plates are riveted with bolts in one of the ends, and in the other with articulated sleeves. These chains are very suitable for all working conditions and for this reason they are the most used. They are also standardized, for example: DIN 8187.

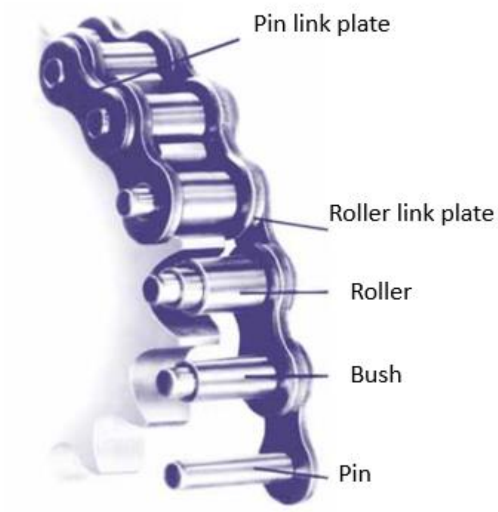


Figure 2-3: Parts of a roller chain (21).

The **link plates** act as the main tension carrier. They must resist shock loading. The link plates are the critical part of the chain when looking at fatigue strength. For this reason, the **roller link plate** is more critical than the **pin link plates.**

Roller: The roller is subject to impact load as it strikes the sprocket teeth during the chain engagement with the sprocket.

Bush: The bushing is subject to shearing and bending stresses transmitted by the plate and roller, and also gets shock loads when the chain engages the sprocket. It must have great tensile strength against shearing and be resistant to dynamic shock and wear.

Pin: The pin is subject to shearing and bending forces transmitted by the plate. At the same time, it forms a load-bearing part, together with the bush, when the chain flexes during sprocket engagement. Therefore, the pin needs high tensile and shear strength and resistance to bending (22).

In order to calculate a chain drive, it is possible to use different formulae according to the kind of chain, but the procedure is the same. As chain drives are standardized, it is obvious that some tables have to be used in order to define all the parameters, such as the number of links, the pitch, the distance between the centres of the wheels or the lineal velocity of the chain.

The method for calculating chains is going to be explained, according to the book: *Maschinen-elemente, Decker K.H.* (23).

- The gear ratio **i**: $i = \frac{n_1}{n_2} = \frac{Z_2}{Z_1}$
- Corrected Power: $P_C\,[kW] = P[kW] \times f_1 \times f_2$, being **P** the power, $\mathbf{f_1}$ an operational factor considering uniform operation and $\mathbf{f_2}$ a factor that depends of the number of teeth of the pinion.

Z_1	11	12	13	14	15	16	17	18	19	20	25	30	35	40	45
f_2	1,8	1,64	1,5	1,39	1,29	1,2	1,13	1,06	1	0,95	0,74	0,61	0,52	0,45	0,39

Table 2-1: Values of f_2 according DIN ISO 10823 (24).

- Select a normalized chain according to this graph. In the X-axis, the rotational speed [rpm], and in the Y-axis the corrected power. After selecting the chain, an additional table is needed in order to know some parameters of the chain, such as the pitch **P** [mm], the breaking load $\mathbf{F_B}$ [kN] or the load per meter **q** (weight) [kg/m]:

Typical values of power-rotational speed in order to select the chain size according ISO 10823.

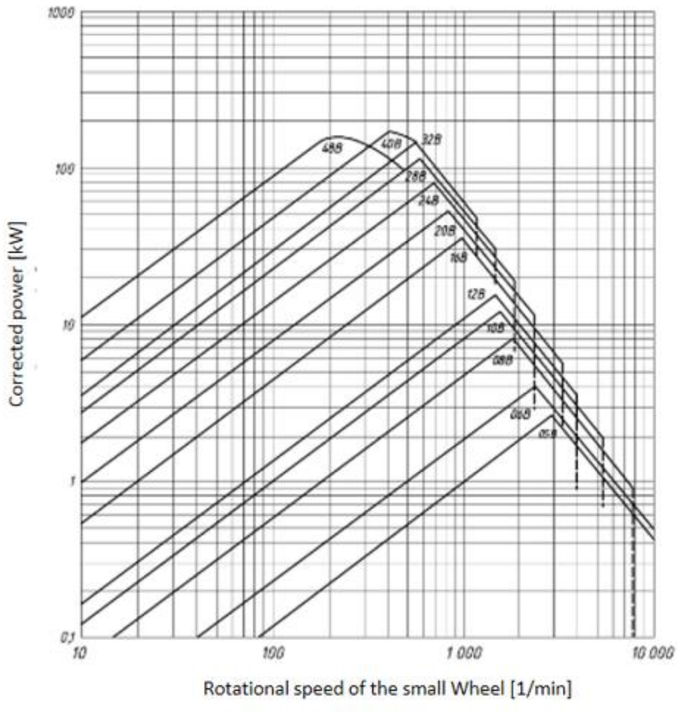


Figure 2-4: Typical values of chain size with the relation power-rotational speed according ISO 10823 (24).

Technical data for chain drives according DIN 8187 (for the values l_1, F_B, A, q, correspond to single chains).

Chain numb.	P [mm]	b1 [mm]	d_R [mm]	e [mm]	g [mm]	l_1 [mm]	F_B [kN]	A [cm2]	q [kg/m]
06 B	9,525	5,72	6,35	10,24	8,2	13,5	9	0,28	0,41
08 B	12,7	7,75	8,51	13,92	11,8	17	18	0,5	0,7
10 B	15,875	9,65	10,16	16,59	14,7	19,6	22.4	0,67	0,95
12 B	19,05	11,68	12,07	19,46	16,1	22,7	29	0,89	1,25

Table 2-2: Technical data for chain drives according DIN 8187 (24).

- Number of links: $X = 2 \times \frac{a_0[mm]}{P[mm]} + \frac{Z_1+Z_2}{2} + f_3 \times \frac{P\,[mm]}{a_0[mm]}$, being $\mathbf{a_0}$ the (theoretical) distance between centres, **P** the pitch, $\mathbf{Z_2}$ and $\mathbf{Z_1}$ the teeth of the wheels, and $\mathbf{f_3}$: $\left(\frac{Z_2-Z_1}{2\times\pi}\right)^2$, in absolute value.
- Length of the chain: $L[mm] = P[mm] \times X$, being **P** the pitch and **X** the number of links.
- It should be recalculated the (real) distance between centres:

$$a[mm] = f_4 \times P[mm] \times [2 \times X\,[mm] - Z_1 - Z_2]$$

Being **P** the pitch, $\mathbf{Z_2}$ and $\mathbf{Z_1}$ the teeth of the wheels, **X** the number of links, and $\mathbf{f_4}$ a non-dimensional parameter, called calculation factor, calculated using this table (Z_S is the number of teeth of the smallest wheel):

$f_u = \frac{X-Z_S}{Z_2-Z_1}$,

Values of f_4 according DIN ISO 10823

f_u	f_4	f_u	f_4	f_u	f_4	f_u	f_4
13	0,24991	2,7	0,24735	1,54	0,23758	1,26	0,22520
12	0,24990	2,6	0,24708	1,52	0,23705	1,25	0,22443
11	0,24988	2,5	0,24678	1,50	0,23648	1,24	0,22361
10	0,24986	2,4	0,24643	1,48	0,23588	1,23	0,22275
9	0,24983	2,3	0,24602	1.46	0,23524	1,22	0,22185
8	0,24978	2,2	0,24552	1,44	0,23455	1,21	0,22090
7	0,24970	2,1	0,24493	1,42	0,23381	1,20	0,21990
6	0,24958	2,0	0,24421	1,40	0,23301	1,19	0,21884
5	0,24937	1,95	0,24380	1,39	0,23259	1,18	0,21771
4,8	0,24931	1,9	0,24333	1,38	0,23215	1,17	0,21652
4,6	0,24925	1,85	0,24281	1,37	0,23170	1,16	0,21526
4,4	0,24917	1,80	0,24222	1,36	0,23123	1,15	0,21390
4,2	0,24907	1,75	0,24156	1,35	0,23073	1,14	0,21245
4	0,24896	1,70	0,24081	1,34	0,23022	1,13	0,21090
3,8	0,24883	1,68	0,24048	1,33	0,22968	1,12	0,20923
3,6	0,24868	1,66	0,24013	1,32	0,22912	1,11	0,20744
3,4	0,24849	1,64	0,23977	1,31	0,22854	1,10	0,20549
3,2	0,24825	1,62	0,23938	1,30	0,22793	1,09	0,20336
3,0	0,24795	1,60	0,23897	1,29	0,22729	1,08	0,20104
2,9	0,24778	1,58	0,23854	1,28	0,22662	1,07	0,19848
2,8	0,24758	1,56	0,23807	1,27	0,22593	1,06	0,19564

Table 2-3: Values of f_4 according DIN ISO 10823 (24).

- Calculation of the lineal velocity of the chain: $V\left[\frac{m}{s}\right] = Z \times P[m] \times n[1/s]$, being **Z** the number of teeth, **P** the pitch and **n** the rotational speed.

- Calculation of the pitch diameters: $d_1[mm] = \frac{P[mm]}{\sin(\tau/2)}$, being **P** the pitch, and $\boldsymbol{\tau}$ the pitch angle [in rad]. But there should be calculated more parameters, such as the tip diameter, the diameter of free rotation, the width….

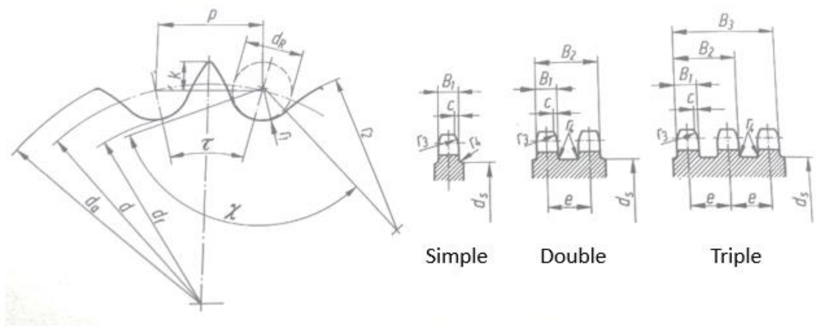


Figure 2-5: Geometrical parameters of a sprocket according DIN 8187 (24).

Root diameter [mm]	$d_f = d - d_R$
Tip diameter [mm]	$d_{a\,min} = d + 1{,}25 \times P - d_R$
	$d_{a\,max} = d + (1 + \frac{1{,}6}{Z}) \times P - d_R$
Diameter of free rotation [mm]	$d_{s\,min} = \frac{P}{\tan(\tau/2)}) - 1{,}05 \times g - 2 \times r_{4\,min} - 1$
	$d_{s\,max} = \frac{P}{\tan(\tau/2)}) - 1{,}05 \times g - 2 \times r_{4\,max} - 1$
Roller bed radius [mm]	$r_{1\,min} = 0{,}5 \times d_R$
	$r_{1\,max} = 0{,}5 \times d_R + 0{,}069 \times \sqrt[3]{d_R}$
Tooth flank radius [mm]	$r_{2\,min} = 0{,}12 \times d_R \times (Z + 2)$
	$r_{2\,max} = 0{,}008 \times d_R \times (Z^2 + 180)$
Chamfer [mm]	$C_{min} = 0{,}1 \times P$
	$C_{min} = 0{,}15 \times P$
Roller bed angle [°]	$X_{min} = \frac{120 - 90}{Z}$
	$X_{max} = \frac{140 - 90}{Z}$
Tooth height above pitch polygon [mm]	$k_{min} = 0{,}5 \times (P - d_R)$
	$k_{max} = 0{,}625 \times P - 0{,}5 \times d_R + 0{,}8 \times \frac{P}{Z}$
Final width [mm]	$B_y = (Y - 1) \times e + B_1$

Table 2-4: Main geometrical parameters of sprockets and their equations (23).

Being **P** the pitch of the chain, τ the pitch angle, $\mathbf{d_R}$ the roller diameter, $\mathbf{g_1}$ the tab height, $\mathbf{r_4}$ the wheel chamfer radius. In this application the chain drive is simple, so **Y=1**.

P ≤ 9,525	B_1	P > 9,525	B_1
Single chain (Y=1)	$0{,}93b_1$	Single chain (Y=1)	$0{,}95b_1$

Table 2-5: Calculation of parameter B_1 according ISO 8187 (24).

For obtaining the range of $\mathbf{r_4}$, this table according to DIN 8187 is used:

P in mm from	P in mm Until (inclusive)	$\mathbf{r_4}$ in mm min	$\mathbf{r_4}$ in mm max
	9,525	0,2	1
9,525	19,05	0,3	1,6

Table 2-6: Values of r_4 according DIN 8187 (24).

Forces are also important in chain drives and they need to be calculated.
In order to calculate the **tensile force** and the **centrifugal force**, these formulae are applied:

$$F[N] = \frac{P\ [W]}{V\ [m/s]}$$

*Formula to calculate the **tensile force***

$$F_f[N] = q[kg/m] \times V[m/s]^2$$

*Formula to calculate the **centrifugal force***

Dynamic force: $F_d[N] = F[N] \times f_1$

Being **P** the power that should be transmitted, **V** the velocity of the pitch line velocity, and **q** the load per meter.

The total force: $F_g[N] = F_d[N] + F_f[N]$

Factors about security of the chain:
Static breaking strength: $S_B = \frac{F_B[N]}{F[N]} \geq 7$
Dynamic fracture resistance: $S_D = \frac{F_B[N]}{F_g[N]} \geq 5$

(23).

2.2.2. Gears

Gear drives are typical elements in the power train. It is possible to use different gears, for instance spurs gears, helical gears or bevel gears.

Spurs gears (or straight-cut gears) are the simplest type and the most common type of gears. They have straight teeth, and are mounted in parallel shafts. Although they are used in many devices, such as an Electric Screwdriver or in washing machines, it is not easy to find many of them in cars, because of the loud. It also increases the stress (25).

Helical gears are cut at an angle to the face of the gear. When two teeth on a helical gear system engage, the contact starts at one end of the tooth and gradually spreads as the gears rotate, until the two teeth are in full engagement. Helical gears reduce the noise and stress (26).

This gradual engagement makes helical gears operate much more smoothly and quietly than spur gears. For this reason, helical gears are used in almost all car transmissions. Because of the angle of the teeth on helical gears, they create a thrust load on the gear when they mesh. Devices that use helical gears have bearings that can support this thrust load (26).

They can be left handed or right handed:

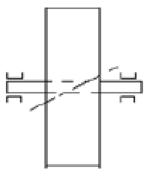


Figure 2-6: Left Handed gear (27).

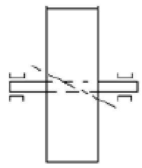


Figure 2-7: Right Handed (27).

Bevel gears can be used to change the direction of drive in a gear system by 90 degrees. A good example is seen as the main mechanism for a hand drill. As the handle of the drill is turned in a vertical direction, the bevel gears change the rotation of the chuck to a horizontal rotation. The pitch surface of bevel gears is a cone (28).

Figure 2-8: Bevel gears (29).

Gears have different parameters:

- **Pitch diameter (Dp):** Diameter of the primitive circumference.
- **External diameter (De):** Also known as total diameter, is the one corresponding to the circumference where the sprocket wheel is inscribed.
- **Internal diameter (Di):** Diameter of the circumference which limit inwardly the teeth.
- **Pitch (P):** It is the distance between two homologous points of two consecutive teeth, measured on the pitch circle. Two wheels engage if and only if they have the same pitch.
- **Number of teeth (Z):** The number of teeth that have a gear.
- **Module (m):** The module is the quotient obtained by dividing the pitch diameter between the number of teeth of the wheel: $m = \frac{D_p}{Z}$, being *m* and D_p in the same units. The common ones are millimetres *[mm]*. This value is standardized.
- **Pressure angle($\boldsymbol{\alpha}$):** It is the angle between the tooth face and the gear wheel tangent (its default value is 20º). Another possible value is 25º.
- **Helix angle($\boldsymbol{\beta}$):** It is the angle between any helix and an axial line on its right, circular cylinder or cone. This characteristic is unique for helical gears.

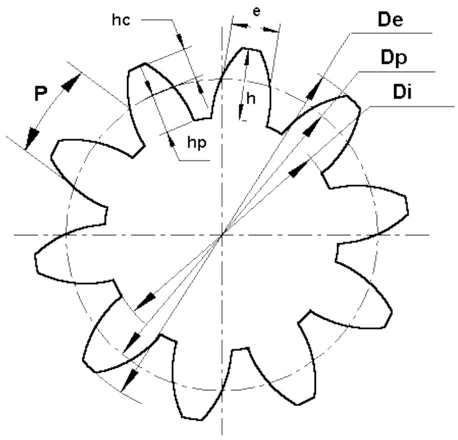


Figure 2-9: Parts of the gear (27).

The procedure to calculate the **forces** in gears is not complicated. The first thing that has to be calculated is the Torque [M], being *N* the power and *n* the angular speed of the gear wheel. The method for calculating forces in gears is taken from the book: *Maschinenelemente* (*Machine elements), Decker K.H.* (23).

$$M\ [Nm] = \frac{N\ [kW] \times 9550}{n\ [rpm]}$$

For calculating the **tangential forces** on gears, this formula is used:

$$F_t[N] = \frac{M\ [Nmm]}{R\ [mm]}$$

The formula for calculating the **radial force** in gears differs in the case of spurs or helical gears.

$$F_r[N] = F_t[N] \times \tan(\alpha)$$

For spur gears

$$F_r[N] = \frac{Ft\ [N]}{\cos(\beta)} \times \tan(\alpha)$$

For helical gear

Helical gears have also another force that should be calculated: The **axial force**:

$$F_a[N] = F_t[N] \times \tan(\beta)$$

For a bevel gear only the formula for the tangential force is the same as in the previous cases. In order to calculate the **axial** and **radial** forces, these formulae are used:

$$F_r[N] = F_t[N] \times \tan(\alpha) \times \cos(\delta_1)$$

For calculating the radial force.

$$F_a[N] = F_t[N] \times \tan(\alpha) \times \sin(\delta_1)$$

For calculating the axial force.

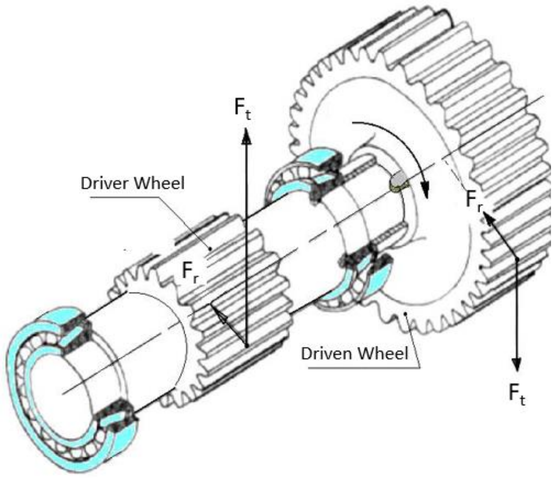


Figure 2-10: Forces and their direction (radial and tangential) in a spur gear (27).

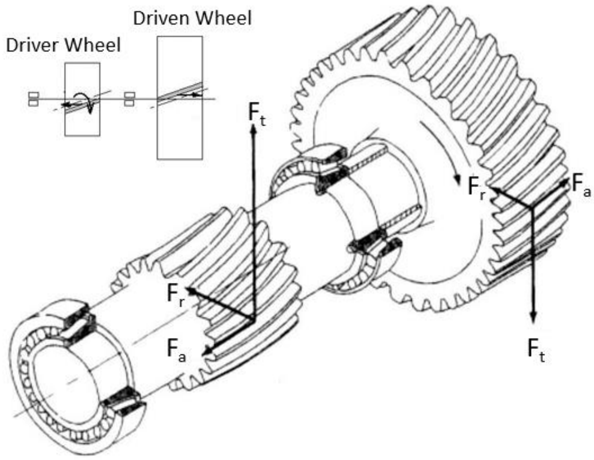


Figure 2-11: Forces and their direction (radial, tangential and axial) in a left handed helical gear (27).

As it is possible to see in the pictures, the **tangential** force in a driver wheel is opposite to the movement. In a driven one, this force facilitates it. The **radial** force acts always radially to the circumference of the gear. The **axial** force is present only in helical gears (23).

2.3. OTHER ELEMENTS/MECHANISM

2.3.1. Bearings

A bearing is a mechanical element that reduces friction between moving parts, such as shafts or gear wheels.

The invention of the roller bearing (with woods rollers) may predate the invention of the wheel. The oldest recovered example of a roller bearing is a wooden ball bearing. The wrecks were dated to 40 BC. Leonardo Da Vinci designed also ball bearings for a helicopter. This was the first design in the aerospace industry (30).

There are a lot of kind of bearings, for instance roller bearings, ball bearings, gear bearings, etc.

Roller bearings carry a load by placing rolling elements between two bearing rings called races. The relative motion of the races causes the rolling elements to roll with very little rolling resistance and with little sliding (31). Common roller bearings use cylinders of greater length than diameter (Cylindrical roller bearing). They have higher load capacity than ball bearings, but high friction coefficient. Other common types of roller bearings are spherical rollers or tapered roller (32).

Ball bearings use balls to maintain the separation between the bearing races. The objective is reducing the rotational friction and support both radial and axial loads (33). With the rolling of the balls, it is possible to reduce more the friction than in the case two surfaces were sliding against each other. They have lower load capacity than roller bearings. However, they can tolerate some misalignment of the inner and outer races (34).

Figure 2-12: Cylindrical roller bearing (35).

Figure 2-13: Ball bearing (36).

The procedure to calculate the life time of bearings is explained now, according to the book: *Maschinenelemente* (*Machine elements), Decker K.H.* (23).

- The dynamic load P: $P' \, [kN] = X' \times F_r \, [kN] + \, Y' \times \, F_a \, [kN]$, being X one non-dimensional parameter (Radial factor) taken from different tables, Fr is the radial force, Y is a non-dimensional parameter (axial factor) taken from different tables, Fa is the axial force.
- The nominal lifetime L: $L = \left(\frac{C \, [kN]}{P \, [kN]}\right)^k \times 10^6$, being **C** the dynamic load rating coefficient and P the dynamic load; **k** is a factor, *k=3* for ball bearings and *k=10/3* for roller bearings.
- Nominal lifetime in hours L_h: $L_h = \frac{L}{n \, [rph]}$, being **L** the nominal lifetime and **n** the rotational speed.
- The static load P: $P_0 \, [kN] = X_0 \times F_{r0} \, [kN] + \, Y_0 \times \, F_{a0} \, [kN]$, being X one non-dimensional parameter (Radial factor) taken from different tables, F_{r0} is the radial force during down time, Y_0 is a non-dimensional parameter (axial factor) taken from different tables, F_{a0} is the axial force down time

Depending the type of bearings, it is possible to take from different tables some parameters (with a previous selected bearing), such us diameter, the dynamic load rating coefficient, or the non-dimensional factors.

Type	d [mm]	Serie 60			
		D[mm]	B[mm]	C[kN]	C_0 [kN]
00	10	32	8	5,6	2,85
01	12	35	8	6,00	3,25
01	15	42	8	6,95	4,05
(d+D)/2		20	60	150	400
f_0		12	15,2	15,9	15,6

Table 2-7: Bearing series 60 characteristics, according DIN 625 (FAG) (24).

$f_0 \times F_a/C_0$	0,3	0,5	0,9	1,6	3,0	6,0	$F_a/F_r > e$: X=0,56 $F_a/F_r \leq e$: X=1, Y=0
e	0,22	0,24	0,28	0,32	0,36	0,43	
$F_a/F_r > e$, Y =	2	1,8	1,59	1,4	1,2	1	$F_{a0}/F_{r0} \leq 0{,}8$: $P_0 = F_{r0}$ $F_{a0}/F_{r0} > 0{,}8$: X_0=0,6, Y_0=0,5

Table 2-8: Calculation of the parameters according DIN 625 (FAG) (24).

2.3.2. Shafts

Shafts are rotating machine elements used to transmit power from a mechanism which produces power to another mechanism or machine that absorbs it. They have usually a circular cross section.

The materials used for shafts can be very different, but the common ones are steels (mild steel is used in most of the cases). When high strength is required, alloy steel such as nickel, chromium or vanadium steel can be used (37).

The manufacturing process of shaft is hot rolling and finished by cold drawing (38).

Keyways are very often used in shafts. They are machine elements used to connect a rotating element (for instance a gear wheel) to a shaft. The key prevents relative rotation between the two parts (39).

The method to calculate shafts is going to be explained, according to the book: *Machine elements, Niemann*

- The first step is discomposed the forces acting on the shafts in two different planes (vertical and horizontal ones) and calculate the bending diagram.
- $M_b = \sqrt{M_x^2 + M_y^2}$, being M_x and M_y the maximum bending moment in the horizontal and vertical plane respectively or the bending moment in in the section that will be used to calculate the minimum diameter.
- $M_v\ [Nmm] = \sqrt{(M_b[Nmm])^2 + \frac{3}{4} \times (\alpha_0 \times M_t\ [Nmm])^2}$, being **$M_b$** the resulting bending moment, **M_t** the torsion moment and **α_0** a non-dimensional parameter (0,7 in the case of bending and torsion).
- $\sigma_{b,adm}[MPa] = \frac{\sigma_{b,w}[MPa]}{SF}$, being **$\sigma_{b,w}$** a characteristic value for each material, taken from some tables, and **SF** a safety factor (the typical values are between 2 and 4).
- Calculate the minimum diameter **d_{min}**: $d_{min}[mm] = 2{,}17 \times \sqrt[3]{b' \times \frac{M_v\ [Nmm]}{\sigma_{b,adm}\ [MPa]}}$, being b' = 1 if the shaft is solid.
- Select and standardized diameter (one which's value is greater than **d_{min}**). The standardized diameters [mm] are: 10; 12; 15; 17; 20; 25; 30; 35; 40; 45; 50; 55; 60; 70; 80; 90; 100; 110; 125; 140; 160; 180; 200; (40).

Types of steels	DIN	UNE	AISI	% C	Strengh [N/mm²] σ_{bW}	[N/mm²] σ_{bSch}
Structural steels						
	St 42-2	Fe430	1020	0,25	220	360
	St 50-2	BFN	A570Gr.50	0,30	260	420
	St 60-2	Fe490-2FN	A572Gr.65		300	470
	St 70-2	Fe590-2FN	-		340	520
		Fe690-2FN		0,50		
Carbon steels						
	C 22,Ck 22	F-1120	1020	00,2 2	280	490
	C 35,Ck 35	F-1130	1035	0,35	330	550
	C 45,Ck 45	F-1140	1045	0,45	370	630

Table 2-9: Different types of steels and their $\sigma_{b,w}$ [MPa] (27).

For calculating the forces of the pinion and wheel of a chain transmission this diagram is used.

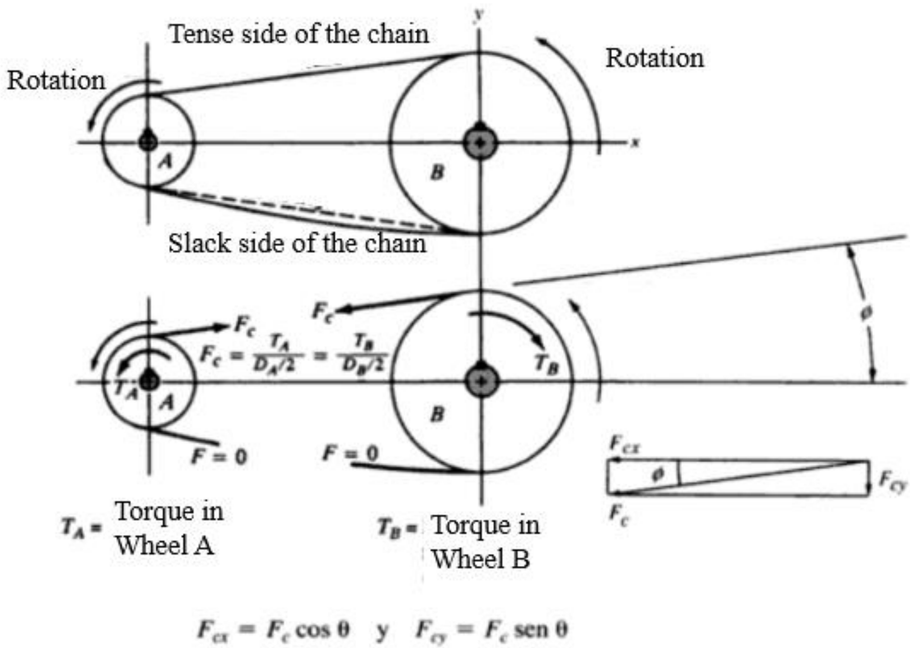


Figure 2-14: Forces of the sprockets (27).

2.4. Previous design

The previous design has three stages, as it is possible to see in Figure 2-16. The motor is the HONDA GX35 (its maximum rotational speed is n=6000 1/min). The engine is started with an electric starter and operates only in the acceleration phase. The engine runs at an average of about 10 seconds and will be run around 35 times every 30 minutes of race.

It has 3 different stages and 2 clutches (K1 is a centrifugal clutch and K2 is a claw clutch)
The main important parameters of this solution are (41):

$$r_{Hinterrad} = 0{,}25$$

$$U_{Hinterrad} = 2 \cdot \pi \cdot r_{Hinterrad} = 2 \cdot \pi \cdot 0{,}25\ m = 1{,}57\ m$$

$$v = 50\ \frac{km}{h} = 50 \cdot \frac{1000\ m}{3600\ s} = 13{,}89\ \frac{m}{s}$$

$$n_{Hinterrad} = \frac{v}{U_{Hinterrad}} = \frac{13{,}89\ \frac{m}{s}}{1{,}57\ m} = 8{,}85\ \frac{1}{s} = 531{,}0\ \frac{1}{min}$$

$$i_{ges} = \frac{n_{Motor}}{n_{Hinterrad}} = \frac{6000\ \frac{1}{min}}{531{,}0\ \frac{1}{min}} = 11{,}30$$

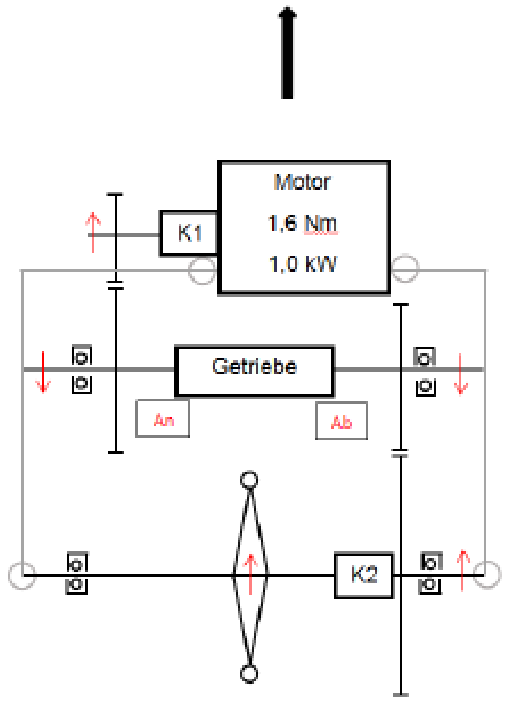


Figure 2-15: Previous Design (41)

The material of the sprockets is Polyoxymethylen (POM). POM combines low friction and high wear resistance with the stiffness and strength needed in parts designed to replace metal. It provides a wide operating temperature range (from -40°C to 120°C), good colourability and good mating with metal and other polymers, as well as dimensional stability in high precision moulding. With a density of $\rho = 1400\frac{kg}{m^3}$. Its chemical formula is: $(CH_2O)_n$ (42).

The main applications of POM are (43):

- Mechanical Gears, in order to make them lighter, like in this project.
- Medical Device Materials.
- Plastics for Sporting goods, like Ski base plates.
- Food contact materials.

Figure 2-16: Ski Base Plate made by POM (44).

3. TOOLS USED

3.1. Creo Parametric

Creo Parametric, formerly known as Pro/Engineer, is a 3D software developed by PTC (Parametric Technology Corporation). This software competes directly with CATIA, Siemens NX or Solidworks.

Creo is available for Microsoft Windows and provides applications for 3D parametric feature solid modelling, 3D direct modelling, 2D views, schematic design or Finite Element Analysis and simulation.

In 1985 the Company was founded, and in 1988 Pro/ENGINEER software was launched. It was the first to market with parametric, associative feature-based, solid modelling software. This year John Deere becomes PTC's first customer (45).

4. DEVELOPMENT OF THE POWER TRAIN

4.1. Required data of the car

4.1.1. Characteristics of the motor

The motor used for this project is a Honda GX200. It is not the original version, but a modified one. The piston and the cylinder heads correspond to the GX120 engine and the capacity of the motor has been reduced from 196 ccm to 153 ccm.

In the following table, it is possible to see the technical data of the modified motor:

Parameter	Value
Maximum Power [kW]	3,5
Maximum rotational speed [1/min]	3500
Minimum rotational speed [1/min]	1800
Maximum torque at 2500 1/min [Nm]	11

Table 4-1: Technical data of the modified Honda Motor.

Figure 4-1: Honda Motor GX200 (46).

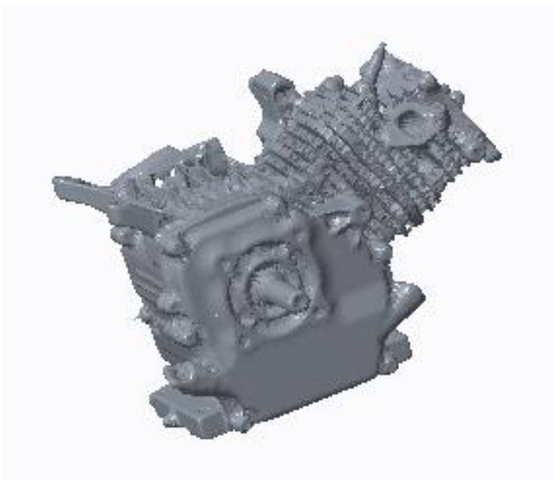


Figure 4-2: CAD model of the modified motor.

Figure 4-2 represents the CAD model of the new motor. As mentioned before, it differs in some aspects from the original one. For instance, it does not have a ventilator because the motor does not work continuously and for this reason a ventilator is not required.

The maximum velocity of the car is 50km/h, and the minimum one is 15km/h. For this purpose, the rotational speed of the motor will vary for 1800 rpm to 3500 rpm.

Applying the equation [2-1] the gear ratio is obtained

$$i = \frac{n_{motor}}{n_{wheel\ car}} = 6{,}6$$

Parameter	Value	Value
Maximum velocity of the car	50 [km/h]	13,89 [m/s]
Maximum rotational speed of the motor	3500 [1/min]	58,33 [1/s]
Maximum rotational speed of the wheel of the car	530,4 [1/min]	8,84 [1/s]
Gear Ratio (i)	6,6	
Minimum Velocity of the car	25,71 [km/h]	7,14 [m/s]
Minimum rotational speed of the motor	1800 [1/min]	30 [1/s]
Minimum rotational speed of the wheel of the car	272,98 [1/min]	4,55 [1/s]

Table 4-2: Maximum and minimum speeds of the motor and car.

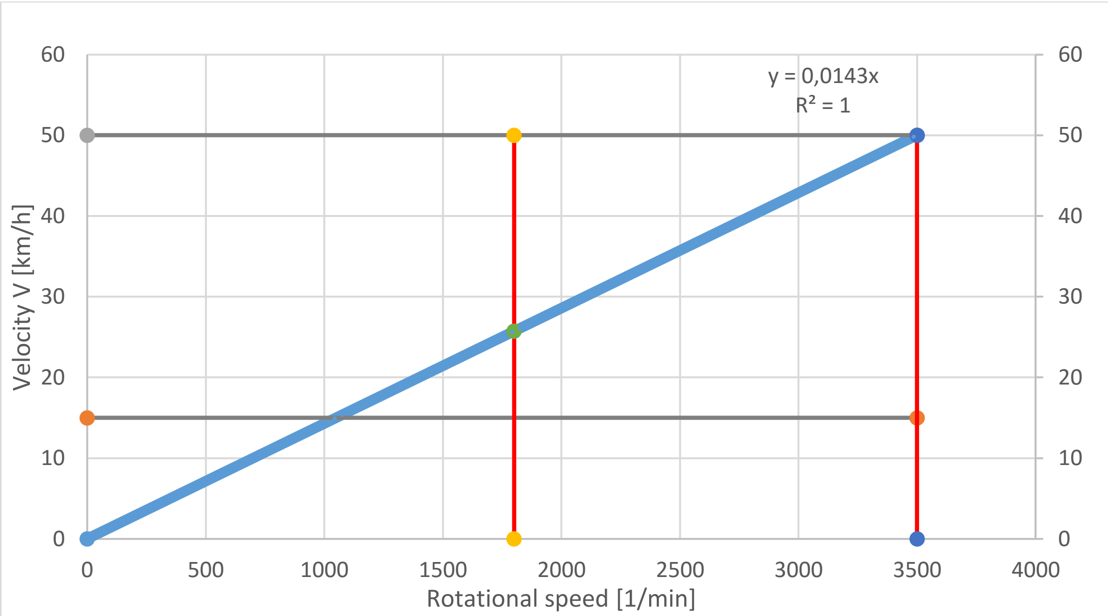


Table 4-3: Relationship between the rotational speed of the motor and the velocity of the car

With these characteristics and with gear ratio i=6.6, the minimum velocity of the car is 25.71 km/h (green point in table 4-3), instead of 15 km/h.

The engine starts at rest and accelerate to 3500 1/min. As it is possible to see in Table 4-3, the velocity of the car at this point is 50km/h. The clutch is slipping till the vehicle speed 25,71 km/h, generating a lot of energetic losses. When the engine is switching off, its rotational speed decreases till 1800 1/min, and the vehicle speed decreases from 50 km/h to 15 km/h.

Starting the engine at 15 km/h and accelerating to 45km/h, the clutch slips between 15 km/h and 25,71 km/h, which causes also energetic losses.

This cycle is repeated about 30 times, which implies high energetic losses.

For reducing losses several options are available:

- Reducing the minimum rotational speed of the motor to 1500 [1/min]. With this solution, the time the clutch is slipping is lower, and the energy lost is smaller.
- Applying a different strategy. Reduce the minimum velocity of the car to, at least, 25,71 km/h in order to prevent slipping of the clutch.
- Use a switchable gear.

4.1.2. Clutch

The goal of the switchable clutch system is reducing losses as much as possible. It synchronizes in the engaged state. The problem (mentioned in Chapter 4.1.1) with this clutch is when the speed of the vehicle is between 15 km/h and 25,7 km/h. The fact that the clutch is not synchronized produces losses during this time, which is approximately 3 seconds in each cycle (the acceleration of the vehicle is approximately $1m/s^2$).

This clutch is totally integrated in the new power train of the vehicle.

In the Table 4-4 it is possible to see the technical data of the clutch:

Synchronization time:	1 s
Weight:	2,65 kg
Capacity requirement:	0,126 Ah
Maximal torque:	75 Nm
Maximum velocity:	600 1/min -> 56 km/h
Press testing:	3,6 mm -> 115 N

Table 4-4: Technical data of the clutch (47)

This clutch has a lot of different parts. Actuators, bearings, spindles or push rods. More information is available is the Master Thesis of Mr. Hack (47).

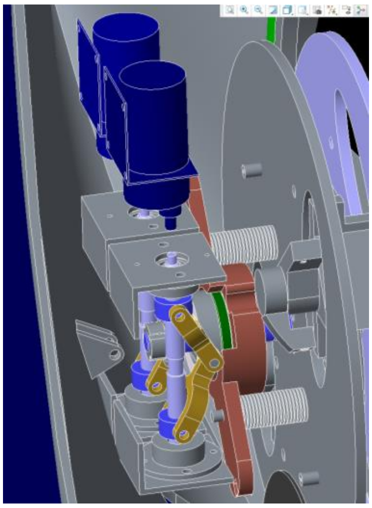


Figure 4-3: 3D Model of the clutch

4.1.3. Available environment for the design and other concept

In Figures 4-3 and 4-4 the available space for the drive train is represented. As the objective is to reach a maximum efficiency, the drive train should be as small as possible in order to reduce the weight. Taking into account the restrictions (for instance maximum chain force in X-direction), a balance/comparison between strength, weight and other criteria will be described in chapter 4.4.

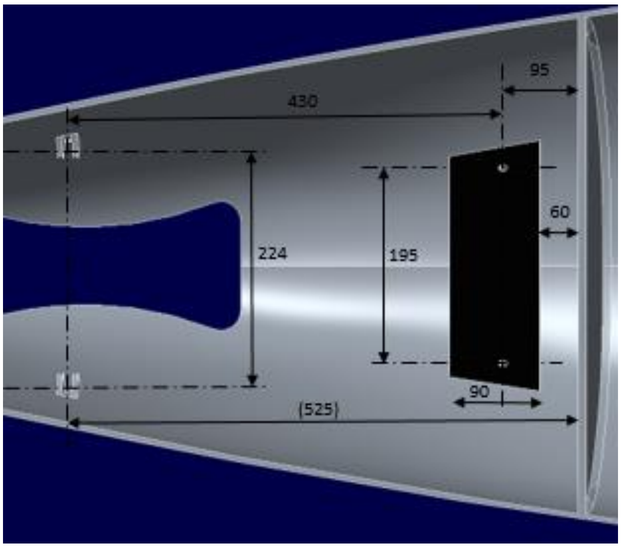


Figure 4-4: Top view of the back part of the vehicle and its dimensions.

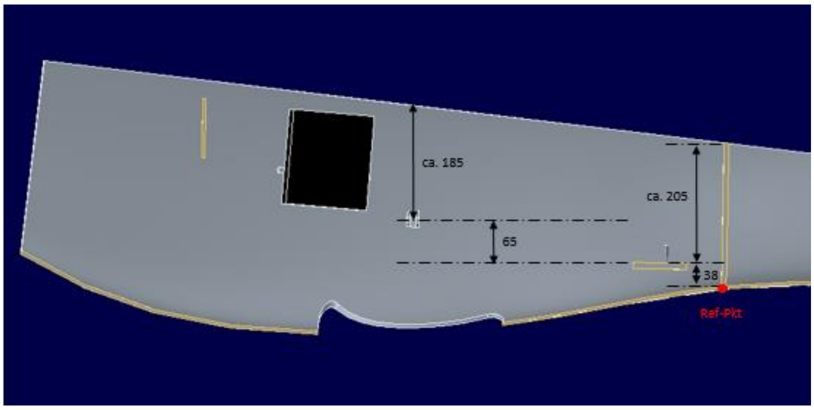


Figure 4-5: Front view of the back part vehicle and dimensions

Figure 4-6 shows the distribution of the forces (and distances) along the shaft of the wheel of the car.

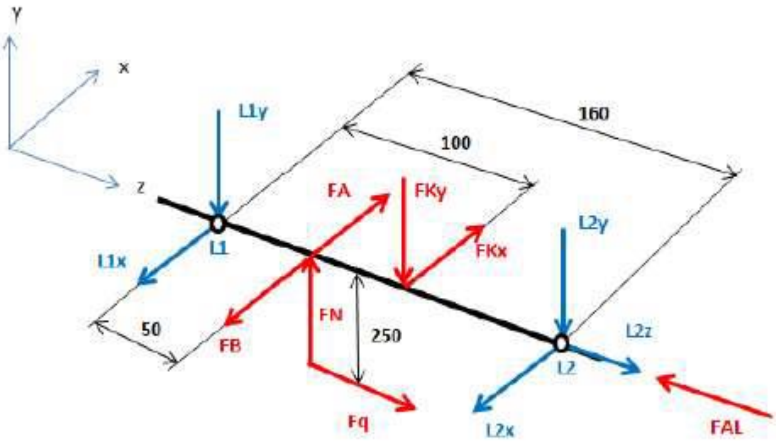


Table 4-5: Distribution of the forces (and distances) along the shaft of the wheel of the car (47).

L1	Bearing point of the floating bearing
L2	Bearing point of the fixed bearing
L1x	Bearing force of the floating bearing in X-direction
L1y	Bearing force of the floating bearing in Y-direction
L2x	Bearing force of the fixed bearing in X-direction
L2y	Bearing force of the fixed bearing in Y-direction
L2z	Bearing force of the fixed bearing in Z-direction
FA	Driving force (max: 300N)
FB	Brake force (max: 300N)
Fq	Shear force (max: 300N)
FKx	Chain Force in X-direction (max: 325N)
FKy	Chain Force in Y-direction (max: 131N)
FAL	Applied force (max: 580N)
FN	Normal force of the car, about 490N

Table 4-6: Forces on the shafts and the important points (47).

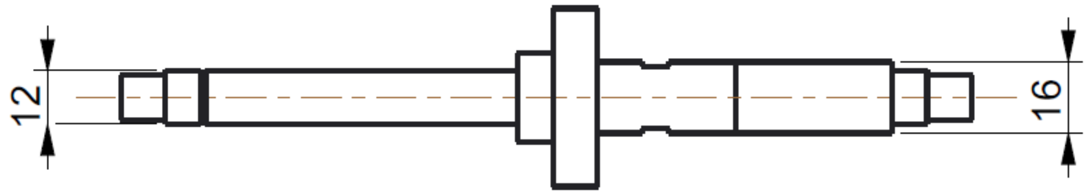
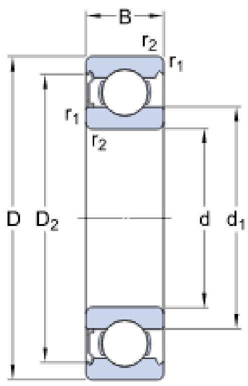


Figure 4-6: Main diameters of the shaft (47).

Bearing force on the floating bearing: Fr=806N, lifetime of 5364h and a safety factor of 2.9.

Bearing forces on the fixed bearing: Fa=580N and Fr=315N; lifetime of 8453h and safety factor of 5.

The bearing used is the following ones: SKF 6001. The characteristics of the bearings are shown in the following Figures 4-8 and 4-9:



d		12	mm
D		28	mm
B		8	mm
d_1	≈	17	mm
D_2	≈	24.72	mm
$r_{1,2}$	min.	0.3	mm

Figure 4-7: Geometrical characteristics of the bearings (48).

Calculation data

Basic dynamic load rating	C	5.4	kN
Basic static load rating	C_0	2.4	kN
Fatigue load limit	P_u	0.1	kN
Reference speed		60000	r/min
Limiting speed		38000	r/min
Calculation factor	k_r	0.025	
Calculation factor	f_0	13	

Mass

Mass bearing	0.0214	kg

Figure 4-8: Technical characteristics of the bearings (48).

All these calculations were done by Mr. Hack in his Master Thesis (47).

4.1.4. Brake System

The available brakes are from the brand Magura. It consists on a bicycle brake adapted to the car (external contacting shoe brake). In Figure 4-10, it is possible to see the Magura HS33 brake. This brake has as external fluid Mineral oil (Magura Royal Blood).

Figure 4-9: Magura HS 33 brake (49).

One rule of the competition says: “The vehicle will be placed on an incline with a 20 percent slope with the driver inside. The brakes will be activated each in turn. Each system alone must keep the vehicle immobile” (50).

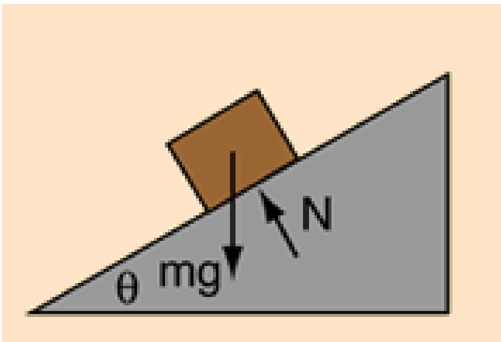


Figure 4-10: Sketch of decomposition of forces of a solid in an inclined plane.

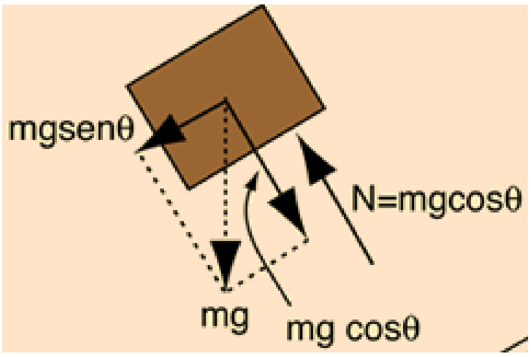


Figure 4-11: Sketch of forces of a solid in an inclined plane.

$\theta = \tan^{-1}\left(\frac{20}{100}\right) = 11{,}3$
$F = m \times g = 100 \times 9{,}8 = 980N$
$F_X = F \times \sin(\theta) = 980 \times \sin(11{,}3) = 192N$
$F_Y = F \times \sin(\theta) = 980 \times \cos(11{,}3) = 961N$ = N

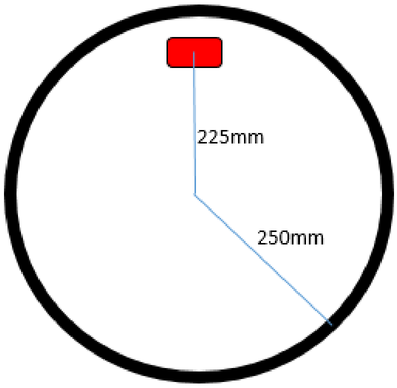


Figure 4-12: Shoe location on the wheel.

$$F_{BR} = 192N \times \frac{250}{225} = 213{,}33\ N$$

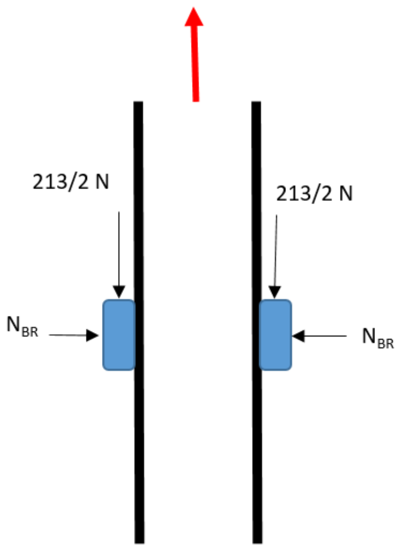


Figure 4-13: Forces of the shoe on the wheel.

The friction static coefficient μ is 0,5 (51)

$$\frac{213.33}{2} = \mu \times N_{BR}$$

$$\boldsymbol{N_{BR} = 213{,}33\ N}$$

4.2. List of requirements

The following table describes the requirements and facts to take into account in the development of this project.

Nr.	**Category**	**Demand**	**Value**	**Remark**
10		**General Information**		
10.10		Product	Drive train	
10.20		Environment	Shell Eco Marathon WIM 2	
20		**Concept**		
20.10	W	Number of stages	1	
30		**Forces, Energy**		
30.10	FF	Power	3,5 kW	
30.20	FF	Input shaft torque (maximum)	11 Nm	
30.30	W	Maximum weight	20 kg	As less as possible, including engine and rear wheel.
40		**Geometry**		
40.10	FF	Diameter of shaft of the motor	19 mm	
40.20	FF	Diameter of shaft of the wheel of the car	12 mm	
40.30	FF	Maximum Diameter of sprocket wheel 2	<500 mm	1 stage, not larger than rear wheel.
40.40	FF	Diameter of the wheel of the car	0,5 m	
50		**Kinematic**		
50.10	FF	Maximum Speed of the wheel of the car	530,4 1/min	(50 km/h)

50.20	FF	Minimum Speed of the wheel of the car	272,92 1/min	(15 km/h)
50.30	FF	i (gear ratio)	6.6	1 stage
50.40	FF	Maximum rotational Speed of the motor	3500 1/min	
50.50	FF	Minimum rotational Speed of the motor	1800 1/min	
60		**Operation and Lifetime**		
60.10		Lifetime	200 hours	2 years x 20 days/year x 5h/day
60.2		Safety Factor (SF)	>1,2	

Table 4-7: List of Requirements.

4.3. Overview of the different concepts

In chapter 2, the different elements that can be used for the power train (belt drives, chain drives and gear wheels) are described.

As the gear ratio should be constant, belt drives are not the best solution.

If the concept consists of gear wheels, the driven wheel should be greater than the wheel of the car, which makes no sense. This fact will be clarified with the following example:

Considering a centre distance between gear wheels of $a = 422{,}3mm$, a gear ratio $i = 6{,}6$, a fixed diameter of the wheel of the car $d = 500mm$ and applying these two equations the final result will be inconsistent:
$i = 6{,}6 = \frac{r_2}{r_1}$ and $r_1 + r_2 = 422{,}3mm$
Solving the system of equations, the final result is: $\boldsymbol{r_2 \approx 366{,}76mm}$ and $r_1 \approx 55{,}57\ mm.$

As mentioned before, the radius of the driven wheel ($\boldsymbol{r_2}$) is higher than the radius of the wheel of the car ($\boldsymbol{r_c = 250mm}$.).

With a chain drive it is possible to change the distance between centres and maintain the sprockets. This situation is not possible with gear wheels.

For these reasons and due to the geometrical restriction and the flexibility that they offer, chain drives are considered for the different proposals that are described in the chapter 4.4

4.4. Comparison of the different proposals: advantages and disadvantages of each concept

As mention in Chapter 4.3, all the proposals are based on chain drives. Each alternative is calculated according Decker book: Machine Elements (23).

All the different proposals have this common data:

Power (kW)	3,5
Maximum speed of the motor (min^{-1})	3500
Gear Ratio (i)	6,6

Table 4-8: Starting data of the proposals

As it is possible to see in Table 4-9, the values of each parameter do not differ so much each other. For instance, the difference between the minimum and maximum diameter of the small sprocket is 17,86 mm. The chain size of most of them is similar too. The most used is 06-B.

Alternative	D1 [mm]	Z1	D2 [mm]	Z2	Chain size
1	60,89	20	400,25	132	06-B
2	66,93	22	439,66	145	06-B
3	49,07	12	323,49	80	08-B
4	61,08	15	400,28	99	08-B
5	61,34	12	404,36	80	10-B
6	63,91	21	418,44	138	06-B

Table 4-9: Main characteristics of the proposals.

The material of the small sprockets is steel AISI 1010 and for the big wheels Aluminium Al 6061-T6.

AISI 1010 carbon steel is a plain carbon steel with 0.10% carbon content. This steel has relatively low strength but it can be quenched and tempered to increase strength. The density of this material is ρ=7800 kg/m^3 (52). Tables 4-20, 4-21, 4-22 show the most important properties of this material.

Element	Content
Iron, Fe	99,17-99,62%
Manganese, Mn	0,30-0,60%
Sulfur, S	≤ 0,050%
Phosphorous, P	≤0,040 %
Carbon, C	0,080-0,13 %

Table 4-10: Chemical Composition of AISI 1010 (52).

Properties	Value
Tensile strength	365 MPa
Yield strength	305 MPa
Elastic modulus	190-210 Gpa
Poisson's ratio	0,27-0,30
Hardness, Brinell	105

Table 4-11: Mechanical Properties of AISI 1010 (52).

Properties	Value
Thermal expansion co-efficient (@0.000-100°C)	12,2 μm/m°C
Thermal conductivity (typical for steel)	49,8 W/mK

Table 4-12: Thermal Properties of AISI 1010 (52).

This steel has also good formability and ductility, and can be easily formed using conventional methods. Forging can be performed. Some other manufacturing processes are tempering, hardening, etc. (52).

Aluminium alloy Al6061 is a precipitation-hardened alloy, developed in 1935. It is one of the most common alloys of aluminium for general-purpose use. The density of this material is ρ=2700 kg/m^3. Tables 4-23, 4-24 show the most important properties of this material (53):

Element	Content
Silicon	0,4-0,8%
Iron	≤0,7%
Copper	0,15-0,4%
Manganese	≤0,15 %
Magnesium	0,80-1,2 %
Chromium	0,04-0,35%
Zinc	≤0,25 %
Titanium	≤0,15 %
Aluminium	95,85-98,56%

Table 4-13: Chemical Composition of Al 6061-T6 (53).

Properties	Value
Tensile strength	260-310 MPa
Elastic modulus	70-80 Gpa
Poisson's ratio	0,33
Hardness, Brinell	95-97

Table 4-14: Mechanical Properties of AL 6061-T6 (53).

This material is highly weldable and used in the production of extrusions by pushing metal through a shaped die. Forging can be also performed.

For calculating the weight, a simplification has been used. Due to the differences in width of the suggested alternatives and fact that the big sprocket should fit in the clutch (diameter 300 mm), screws are used. For placing these screws an additional area is needed. This area may vary depending on the geometry of the alternatives, keeping the total volume of the sprocket approximately constant. This is the reason of using 280 mm as a "fictitious diameter". The weight of the different alternatives is detailed in table 4-25.

The forces of the sprockets on the shaft are calculated by using the following formulas:

$$F[N] = \frac{T[Nmm]}{R[mm]}$$

$$F_x[N] = F[N] \times \cos(\alpha)$$

$$F_y[N] = F[N] \times \sin(\alpha)$$

Where **F** is the total force, **Fx** is the force in the X direction, **Fy** the force in the Y direction, T the torque and **α** the degree between the horizontal plane and the force F.
The maximum force in X direction **Fx** should be 325 N, and the maximum force **Fy** 131 N. The forces on the table 4-26 are calculated with the maximum torque (at 2500 1/min), so they are the maximum forces that the sprockets will suffer during their operation.

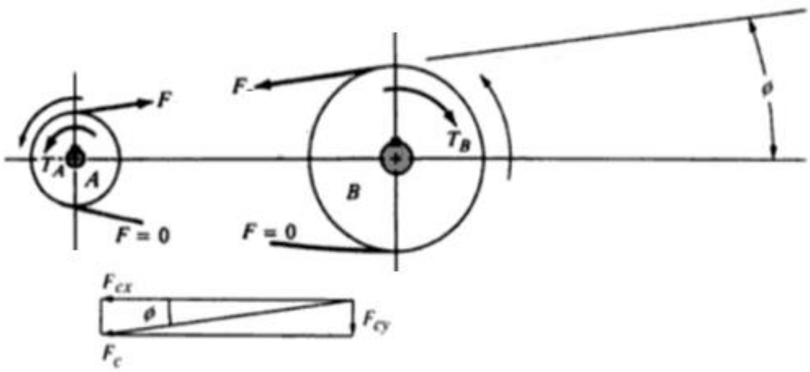


Figure 4-14: Sprocket force calculation (27).

ALTERNATIVE 1	D1 [mm]	60,89	d1 [mm]	19	Weight 1 [kg]	0,11	Total [kg]	1,03
	D2 [mm]	400,25	d2 [mm]	280	Weight 2 [kg]	0,92		
	b [mm]	5,32						
ALTERNATIVE 2	D1 [mm]	66,93	d1 [mm]	19	Weight 1 [kg]	0,13	Total [kg]	1,43
	D2 [mm]	439,66	d2 [mm]	280	Weight 2 [kg]	1,3		
	b [mm]	5,32						
ALTERNATIVE 3	D1 [mm]	49,07	d1 [mm]	19	Weight 1 [kg]	0,1	Total [kg]	0,5
	D2 [mm]	323,49	d2 [mm]	280	Weight 2 [kg]	0,4		
	b [mm]	7,21						
ALTERNATIVE 4	D1 [mm]	61,08	d1 [mm]	19	Weight 1 [kg]	0,15	Total [kg]	1,4
	D2 [mm]	400,28	d2 [mm]	280	Weight 2 [kg]	1,25		
	b [mm]	7,21						
ALTERNATIVE 5	D1 [mm]	61,34	d1 [mm]	19	Weight 1 [kg]	0,19	Total [kg]	1,84
	D2 [mm]	404,36	d2 [mm]	280	Weight 2 [kg]	1,65		
	b [mm]	9,17						
ALTERNATIVE 6	D1 [mm]	63,91	d1 [mm]	19	Weight 1 [kg]	0,11	Total [kg]	1,21
	D2 [mm]	418,44	d2 [mm]	280	Weight 2 [kg]	1,1		
	b [mm]	5,32						

Table 4-15: Weight of the different alternatives.

Alternative 1	D1 [mm]	400,25	R1 [mm]	200,125	α (rad)	0,385	Fx [N]	334,855
	D2 [mm]	60,89	R2 [mm]	30,445	F [N]	361,307	Fy [N]	135,702
	a [mm]	418,7	R1-R2 [mm]	169,68	α (deg)	22,072		
Alternative 2	D1 [mm]	439,66	R1 [mm]	219,83	α (rad)	0,419	Fx [N]	300,215
	D2 [mm]	66,93	R2 [mm]	33,465	F [N]	328,702	Fy [N]	133,850
	a [mm]	418	R1-R2 [mm]	186,365	α (deg)	24,042		
Alternative 3	D1 [mm]	323,49	R1 [mm]	161,745	α (rad)	0,314	Fx [N]	426,378
	D2 [mm]	49,07	R2 [mm]	24,535	F [N]	448,339	Fy [N]	138,600
	a [mm]	422,1	R1-R2 [mm]	137,21	α (deg)	18,017		
Alternative 4	D1 [mm]	400,28	R1 [mm]	200,14	α (rad)	0,387	Fx [N]	333,553
	D2 [mm]	61,08	R2 [mm]	30,54	F [N]	360,183	Fy [N]	135,922
	a [mm]	416,2	R1-R2 [mm]	169,6	α (deg)	22,182		
Alternative 5	D1 [mm]	404,36	R1 [mm]	202,18	α (rad)	0,383	Fx [N]	332,672
	D2 [mm]	61,34	R2 [mm]	30,67	F [N]	358,657	Fy [N]	134,030
	a [mm]	425,7	R1-R2 [mm]	171,51	α (deg)	21,955		
Alternative 6	D1 [mm]	418,44	R1 [mm]	209,22	α (rad)	0,397	Fx [N]	317,405
	D2 [mm]	63,91	R2 [mm]	31,955	F [N]	344,234	Fy [N]	130,657
	a [mm]	422,3	R1-R2 [mm]	177,265	α (deg)	22,782		

Table 4-16: Forces of the sprockets on the shaft.

Table 4-17 shows a comparison of the different solutions according to parameters such as weight, space used or assembly difficulty. The rated scale is: ++ (Very good), + (good), 0 (neutral), - (bad), -- (very bad).

	Alternative 1	Alternative 2	Alternative 3	Alternative 4	Alternative 5	Alternative 6
Weight	+	-	++	-	--	0
Space used	0	-	+	0	0	+
Strength	-	++	--	-	-	++
Easy assembly	+	-	+	+	-	+

Table 4-17: Comparison of the different alternatives.

According to this evaluation, the best solutions are alternative 3 and 6.

The space used and the weight of the alternative 3 are better than these values of the alternative 6, but the strength of the 3rd alternative is not enough to withstand the chain forces. For this reason, alternative 6 has been chosen.

The efficiency of this new drive train is higher than the previous one. The old version has three stages, which reduces its efficiency at least 6%. With this new concept, the efficiency is reduced around 2-3%. In order to keep the chain tauted, the previous solution uses additional wheels, which create friction and reduces more the efficiency. In this new solution it is possible to vary a little bit the distance between centres (using screws). With this small modification the efficiency is not decreased.

5. DETAILED DESCRIPTION OF THE FINAL SOLUTION AND CAD-DATA

The force of the chain on the sprockets is calculated according the following procedure (54):

$$\sigma_b = \frac{F}{b \times m} \times Y_{Fa} = \frac{344{,}2}{5{,}72 \times 3} \times 2.7 = 54{,}2\ MPa$$

$$\varepsilon = 2$$

$$Y_\varepsilon = 0.25 + \frac{0{,}75}{\varepsilon} = 0{,}625$$

$$\sigma_{F0} = \sigma_b \times Y_{SA} \times Y_\varepsilon = 54{,}2\ MPa$$

$$K_A = 1{,}8$$

$$K_V = 1 + \left(\frac{K_1}{K_A \times \frac{F}{b}} + K_2\right) \times K_3 = 1 + \left(\frac{9{,}6}{110} + 0{,}0193\right) \times 2{,}12 = 1{,}23$$

$$V_1 = \pi \times n_1 \times d_1 = 11{,}67\ m/s$$

$$K_3 = \frac{z_1 \times V_1}{100} \times \sqrt{\frac{i^2}{(1+i)^2}} = 2{,}12$$

$$\sigma_F = \sigma_{F0} \times K_V \times K_A = 120\ MPa$$

$$SF = \frac{\sigma_{zul}}{\sigma_F}$$

A good safety factor should vary between 2 and 3, so $\sigma_{zul} \geq 240\ MPa$.

The material of the small sprocket (steel AISI 1010), has a $\sigma_{zul} = 365\ MPa$. The safety factor is: $SF = \frac{365}{120} = 3$

The material of the big sprocket (Aluminium Al6061-T6), has a $\sigma_{zul} = 310\ MPa$. The safety factor is: $SF = \frac{310}{120} = 2{,}6$

In order to fix the big sprocket on the clutch, 4 screws are needed. The tolerance between the sprocket and the clutch is H7/n6. (The diameter is 300 mm).

	Hole	**Shaft**
Tolerance class	H7	n6
Lower limit deviation [µm]	0	34
Upper limit deviation [µm]	52	66

Table 5-1: Upper and lower deviation of tolerance H7/n6.

Minimum interference [µm]	-66
Maximum interference [µm]	18

Table 5-2: Maximum and minimum interference of tolerance H7/n6.

This combination is a transition fit. This fit is used for accurate location where greater interference is permissible. The Mounting of these parts are used pressing and applying a light force.

The hole quality is IT7, while the shaft has a quality immediately below the hole (IT6).

Some applications of this fit are: armatures of electric motors on shafts, gear rims, fixed plugs, driven bushings, flushed bolts.
Some of the selected preferred fits are: H7/n6, H8/n7, H8/p7, H7/m6 (55).

The calculation of the forces on the screws is the following (54):

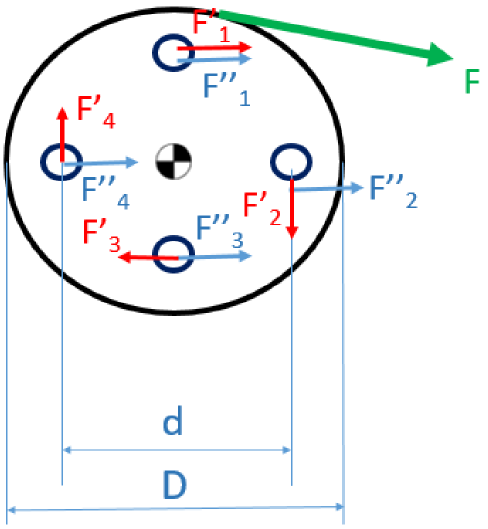


Figure 5-1: Disposition of the screws in the big sprocket

$$d = 275mm, D \cong 420mm, F = 344{,}2\ N$$

$$F'_1 = F'_2 = F'_3 = F'_4 = F' = \frac{1}{4} \times F \times \frac{D}{d} = \frac{1}{4} \times 344{,}2 \times \frac{420}{275} = 131{,}4N$$

$$F''_1 = F''_2 = F''_3 = F''_4 = F'' = \frac{1}{4} \times F = \frac{1}{4} \times 344{,}2 = 86{,}05N$$

$$F_{screw} = F' + F'' = 217{,}45\ N$$

Then the strength of the screw is calculated. This value has to be is equal or less than the yield strength. The screws have a diameter of 6 mm and a quality of 3.6 (using a safety factor of 3). Quality 3.6 means:

- Tensile strength: $\sigma_{\max screw} = 300\ MPa$
- Yield strength: $\sigma_y = 0{,}6 \times 300 = 180\ MPa$

$$\tau_{max} = \frac{F_{screw}}{\frac{\pi}{4} \times d_{screw}} = \frac{217{,}45\ N}{\frac{\pi}{4} \times 6} = 7{,}7\ MPa$$

$$\sigma_{max} = 2 \times \tau_{max} = 2 \times 7{,}7 = 15{,}4\ MPa$$

$$\sigma_{adm} = n_f \times \sigma_{max} = 3 \times 15{,}4\ MPa = 46{,}2\ MPa$$

$$\sigma_y = 0{,}6 \times 300 = 180\ MPa$$

$$46{,}2\ MPa\ < 180\ MPa.$$

As the stress in the screws is less than the maximum yield strength, the screws fulfil the requirements.

According DIN 960, in the Table 5-3 it is possible to see the screws needed (56). Nuts have to be used too. In the Table 5-3 there are needed nuts according DIN 934 (57):

Screw	Total	Nut	Total
M 6	4	M 6	4

Table 5-3: Screws and nuts needed for the big sprocket

The small sprocket needs a key. It is a machine element used to connect a rotating machine element to a shaft. The key prevents relative rotation between the two parts and may enable torque transmission. The key is the following one: DIN 6885-5-5-16-A.

$p_m = \frac{2 \times T \times K_\lambda}{d \times h\prime \times l\prime \times n \times \varphi} = \frac{2 \times 11000 \times 1{,}3}{19 \times 6 \times 10 \times 1 \times 1} = 25\ MPa \leq p_{zul}$ (54).

The material of the key is a non-alloy quality steel (Steel C45). Its p_{zul} is much greater than p_m. For this reason, it is possible to ensure that the key will not break (58).

The figure 5-1 shows the piece used to fix the small sprocket to the shaft. It has five screws. Four of them are used for the union with the sprocket, and the other one is for the union with the shaft.

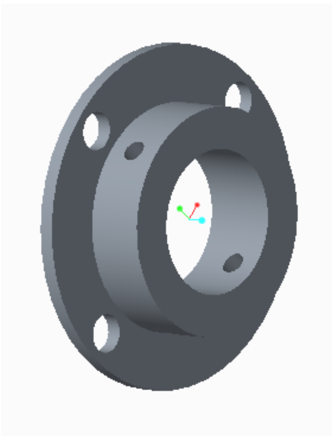


Figure 5-2: Piece used the union of the small sprocket and the shaft.

The calculation of the forces on the screws is the following (54):

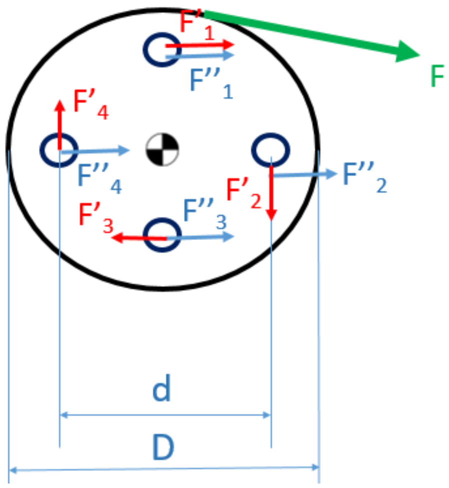


Figure 5-3: Disposition of the screws in the small sprocket.

$d = 55mm, D = 68mm, F = 344{,}2\ N$

$$F'_1 = F'_2 = F'_3 = F'_4 = F' = \frac{1}{4} \times F \times \frac{D}{d} = \frac{1}{4} \times 344{,}2 \times \frac{68}{55} = 106{,}4N$$

$$F''_1 = F''_2 = F''_3 = F''_4 = F'' = \frac{1}{4} \times F = \frac{1}{4} \times 344{,}2 = 86{,}05N$$

$$F_{screw} = F' + F'' = 192{,}5\ N$$

$$\tau_{max} = \frac{F_{screw}}{\frac{\pi}{4} \times d_{screw}} = \frac{192{,}5\ \ N}{\frac{\pi}{4} \times 6} = 6{,}8\ MPa$$

$$\sigma_{max} = 2 \times \tau_{max} = 2 \times 6{,}8 = 13{,}6\ MPa$$

$$\sigma_{adm} = n_f \times \sigma_{max} = 3 \times 13{,}6\ MPa = 40{,}8\ MPa$$

$$\sigma_y = 0{,}6 \times 300 = 180\ MPa$$

$40{,}8\ MPa\ < 180\ MPa.$

Following the same procedure, the quality of the other screw should be higher. Its quality is 8.8. As the stress in the screws is less than the maximum yield strength, the screws fulfil the requirements.

According DIN 960, in the table 5-4 it is possible to see the screws needed (56). Nuts have to be used too. In the Table 5-4 there are needed nuts according DIN 934 (57):

Screw	Total	Nut	Total
M 5	4	M 5	4
M 3	1	M 3	1

Table 5-4: Screws and nuts needed for the small sprocket.

The Roller Chain manufacturer is IWIS, and it is standardized according DIN 8187 (06-B size):

Parameter	Value
d_1 [mm]	6,35
b_1 [mm]	5,72
a_1 [mm]	12,9
P [mm]	9,525
g [mm]	8,2
X	176
Length [mm]	1676,4

Table 5-5: Geometrical details of the chain according DIN 8187 (59).

Parameter	Value
F_B [kN]	9
A [cm^2]	0,28
q [kg/m]	0,41
Total weight [kg]	0,687

Table 5-6: Technical details of the chain according DIN 8187 (59).

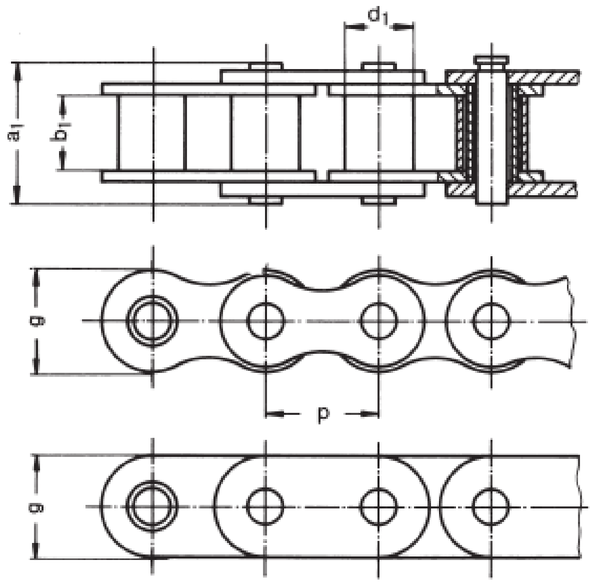


Figure 5-4: Roller chain 2D view (59).

The total weight of the sprockets and the chain is around 2 kg.

5.1. Frame

All the parts should be fixed to a frame, which is also fixed to the chassis. Figures 5-5 shows how all parts are fixed together. The frame consists of several components. Most of them at screwed or welded each other.

The use of screws in this design give the opportunity to exchange damaged components and, if necessary, replaced them with more rigid ones. Most of the components can be replaced or reworked. The general concept of frame is flexible for modifications.

For manufacturing the frame, aluminium sheets are used as at first step. It is possible to make some modifications on them applying different processes such as drilling (in most of the parts) or using a waterjet cutting machine in order to remove the material.

The material is: Aluminium 7075-T6. This material has a high ultimate strength (572 MPa) (60). With this value is possible to ensure that the frame structure is not going to break or suffer important damages.

The parts which are joined using welding process, Gas tungsten arc welding (GTAW) process is used. This kind of welding is also known as tungsten inert gas (TIG). It an arc welding process that uses a non-consumable tungsten electrode to produce the weld. It can be used on different types of metals, but most commonly aluminium is used in particular metals of a smaller thickness (61).

This process has some advantages (62):

- Is easily applied to thin materials.
- Produces very high-quality, superior welds.
- Welds can be made with or without filler metal.
- Provides precise control of welding variables (i.e. heat).
- Welding yields low distortion.

The base plate of the frame if the part 1, as it is possible to see in Figure 5-9. Part 2 and part 3 are directed screwed in part 1.

In Figure 5-5 and 5-6 it is possible to see the whole assembly of the drive train, including the motor, the frame, the clutch and the chain transmission.

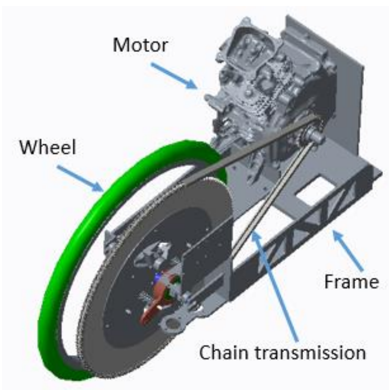


Figure 5-5: Assembly of the frame including the chain mechanism, wheel of the car and frame.

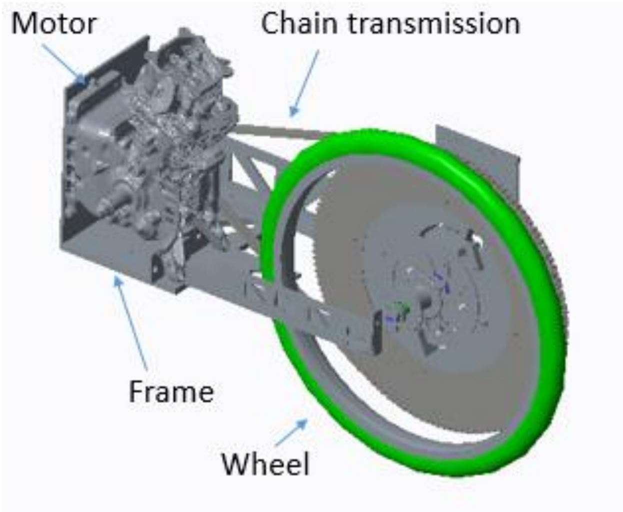


Figure 5-6: Assembly of the frame including the chain mechanism, wheel of the car and frame.

The base plate of the frame is the part 1, as it is possible to see in Figure 5-7. Part 2 and part 3 are directed screwed in part 1. More details about the screws used are available in the CAD Model.

All the frame parts, the power train and the clutch should be assembled outside the vehicle. The first step is joining parts 1 and 2 of the frame. Then, the clutch and the power train is mounted. Finally, part 3 of the frame is fixed in the whole assembly. Once it is finished, it is possible to locate in the vehicle.

In Figure 5-7 the assembly of the frame is shown:

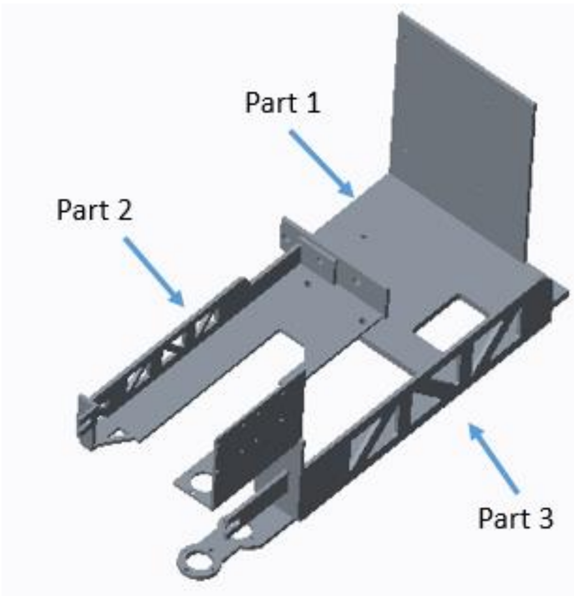


Figure 5-7: Assembly of the frame.

In Figure 5-8 it is possible to see the points where the frame is fixed to the chassis. More details are represented in CAD models.

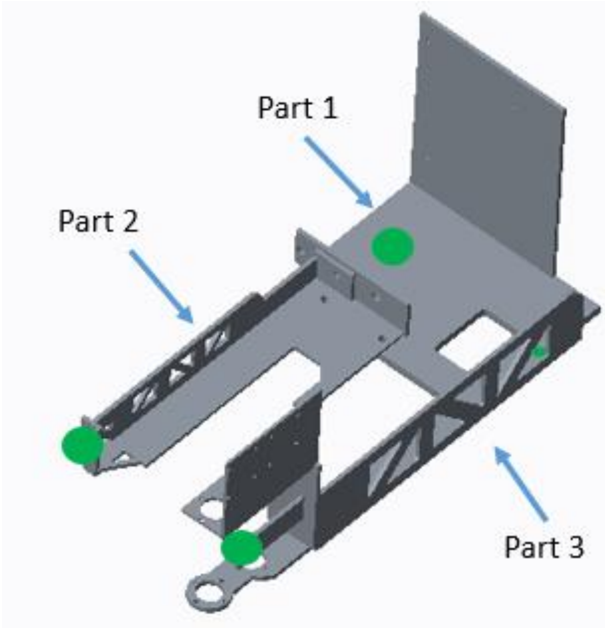


Figure 5-8: Assembly of the frame.

The material of the frame is aluminium, and the total volume is 1229,386 cm^3. Taking into account the density of the aluminium, the total weight of the frame is around 2,7 kg.

In the Table 5-7 it is possible to see the different screws according DIN 960 needed for screwed the frame (56). Nuts have to be used too. In the Table 5-7 there are needed nuts according DIN 934 (57):

Screw	Total	Nut	Total
M 3,4	8	M 3,4	8
M 4	5	M 4	5
M 5	8	M 5	8
M 6	11	M 6	11
M 8	7	M 8	7
M 10	2	M 10	2

Table 5-7: Screws and nuts needed for the frame

6. CONCLUSIONS

In the present Bachelor Thesis, a new concept for the power train of the vehicle WIM II of the High-Efficiency-Karlsruhe team has been developed. This concept will be built and mounted for the next Shell Eco-Marathon competition in 2017.

The final solution simplifies the previous one. It reduces losses (for instance it uses a chain instead of three, which makes the concept 4% more efficient). This new solution is also easier to repair during the race.

For this application, chain drives, gear wheels and belt drives have been considered. Due to the advantages that they offer, the different suggested concepts consist of chain drives. The different alternatives are compared by using several criteria such as the weight or strength. The weight of the module has to be as lower as possible and have enough strength. All the calculations have been done in the worst case, with a safety factor always higher than 1,5. With this high factor, safety is guaranteed.

Further improvements are possible. For instance, the design of a lighter frame or the use of an electric engine. These two modifications could make the efficiency increase.

Considering the problem of the slipping clutch, and follow the strategy mentioned in Chapter 4.1, it is possible that the results are not as successful as expected. As the motor is still in the testing phase, it is not possible to guarantee at this moment good results.

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APPENDIX

Alternative 1:

Technical data of the alternative 1:

Power	P (kW)	3,5
Maximum speed of the motor	$n_a = n_1$ (min^{-1})	3500
Gear Ratio	i	6,6
Initial distance between centres	a_0 (mm)	420
<u>Chain size and chain speed:</u>		
Service Factor	f_1	1,5
Number of teeth of the small wheel	z_1	20
Teeth factor	f_2	0,95
Required number of teeth of the big wheel	z_{2erf}	132
Selected number of teeth of the big wheel	z_2	132
Corrected Power	P_D (kW)	<u>4,988</u>
Roller chain number		06 B
Pitch of the chain	p (mm)	9,525
Breaking load on the chain	F_B (kN)	9
Pressed surface	A (cm^2)	0,28
Load per meter	q (kg/m)	0,41
Chain speed	v (m/s)	<u>11,113</u>
<u>Geometrical characteristics:</u>		
Factor based on the number of links	f_3	91,19
Number of links	X_0	142,28
Rounded number of links	X	<u>142</u>
Distance factor	f_4	92
Final distance between centres	a (mm)	<u>418,7</u>
<u>Forces</u>		
Static tensile force of the chain	F (kN)	0,315
Dynamic force of the chain	F_d (kN)	0,472

Centrifugal force	F_f (kN)	0,051
Total force	F_g (kN)	0,523
static breaking strength	S_B	28,6
Is the static breaking strength greater than 7?		***Yes!***
Dynamic fracture resistance	S_D	17,2
Is the dynamic fracture resistance greater than 5?		***Yes!***

Table A- 1: Technical data of the alternative 1 (23).

Geometrical data of the alternative 1:

Dimensions and technical data of the roller chain:				
Pitch of the chain	p (mm)	9,525		
Roller width	b1 (mm)	5,72		
Roller diameter	dR (mm)	6,35		
Max. height of the plate	g1 = g (mm)	8,2		
Simple-Double-Triple chain:				
Number of juxtaposed chains	Y	1		
Distance	e (mm)	10,24		
Detailed dimensions of gears for roller chains:				
Wheel chamfer radius	r4,min (mm)	0,2		
	r4,max (mm)	1		
Tooth width	B1 (mm)	5,32		
Number of teeth of the sprockets	**z**	**132**	**z**	**20**
Sprocket dimensions:				
Pitch angle	t (°)	2,7	t (°)	18,0
	t (rad)	0,048	t (rad)	0,314
Pitch diameter	d (mm)	400,25	d (mm)	60,89
Root diameter	d_f (mm)	393,90	d_f (mm)	54,54
Tip diameter	$d_{a,min}$ (mm)	403,54	$d_{a,min}$ (mm)	64,83
	$d_{a,max}$ (mm)	405,81	$d_{a,max}$ (mm)	66,44
Diameter of free rotation	$d_{s,min}$ (mm)	388.53	$d_{s,min}$ (mm)	48.53
	$d_{s,max}$ (mm)	390.13	$d_{s,max}$ (mm)	50.13

Roller bed radius	$r_{1,min}$ (mm)	3,21	$r_{1,min}$ (mm)	3,21
	$r_{1,max}$ (mm)	3,33	$r_{1,max}$ (mm)	3,33
Tooth flank radius	$r_{2,min}$ (mm)	102,11	$r_{2,min}$ (mm)	16,76
	$r_{2,max}$ (mm)	894,28	$r_{2,max}$ (mm)	29,46
Chamfer	c_{min} (mm)	0,95	c_{min} (mm)	0,95
	c_{max} (mm)	1,43	c_{max} (mm)	1,43
Roller bed angle	χ_{min} (°)	119,3	χ_{min} (°)	115,5
	χ_{max} (°)	139,3	χ_{max} (°)	135,5
Tooth height above pitch polygon	k_{min} (mm)	1,59	k_{min} (mm)	1,59
	k_{max} (mm)	2,84	k_{max} (mm)	3,16
Final width	B_Y (mm)	5,32	B_Y (mm)	5,32

Table A- 2: Geometrical data of the alternative 1 (23).

Alternative 2:

Technical data of the alternative 2:

Power	P (kW)	3,5
Maximum speed of the motor	$n_a = n_1$ (min^{-1})	3500
Gear Ratio	i	6,6
Initial distance between centres	a_0 (mm)	420
Chain size and chain speed:		
Service Factor	f_1	1,5
Number of teeth of the small wheel	z_1	22
Teeth factor	f_2	0,85
Required number of teeth of the big wheel	z_{2erf}	145,2
Selected number of teeth of the big wheel	z_2	145
Corrected Power	P_D (kW)	4,463
Roller chain number		06 B
Pitch of the chain	p (mm)	9,525
Breaking load on the chain	F_B (kN)	9
Pressed surface	A (cm^2)	0,28
Load per meter	q (kg/m)	0,41
Chain speed	v (m/s)	12,224

Geometrical characteristics:		
Factor based on the number of links	f_3	383,22
Number of links	X_0	181,33
Rounded number of links	X	182
Distance factor	f_4	98.5
Final distance between centres	a (mm)	418,0
Forces		
Static tensile force of the chain	F (kN)	0,286
Dynamic force of the chain	F_d (kN)	0,429
Centrifugal force	F_f (kN)	0,061
Total force	F_g (kN)	0,491
Static breaking strength	S_B	31,4
Is the static breaking strength greater than 7?		***Yes!***
Dynamic fracture resistance	S_D	18,3
Is the dynamic fracture resistance greater than 5?		***Yes!***

Table A- 3: Technical data of the alternative 2 (23).

Geometrical data of the alternative 2:

Dimensions and technical data of the roller chain:				
Pitch of the chain	p (mm)	9,525		
Roller width	b_1 (mm)	5,72		
Roller diameter	d_R (mm)	6,35		
Max. height of the plate	$g_1 = g$ (mm)	8,2		
Simple-Double-Triple chain:				
Number of juxtaposed chains	Y	1		
Distance	e (mm)	10,24		
Detailed dimensions of gears for roller chains:				
Wheel chamfer radius	$r_{4,min}$ (mm)	0,2		
	$r_{4,max}$ (mm)	1		
Tooth width	B_1 (mm)	5,32		
Number of teeth of the sprocket	**z**	**22**	**z**	**145**

Sprocket dimensions				
Pitch angle	t (°)	16,4	t (°)	2,5
	t (rad)	0,286	t (rad)	0,043
Pitch diameter	d (mm)	66,93	d (mm)	439,66
Root diameter	d_f (mm)	60,58	d_f (mm)	433,31
Tip diameter	$d_{a,min}$ (mm)	70,80	$d_{a,min}$ (mm)	442,94
	$d_{a,max}$ (mm)	72,49	$d_{a,max}$ (mm)	445,22
Diameter of free rotation	$d_{s,min}$ (mm)	54,64	$d_{s,min}$ (mm)	427,95
	$d_{s,max}$ (mm)	56,24	$d_{s,max}$ (mm)	429,55
Roller bed radius	$r_{1,min}$ (mm)	3,21	$r_{1,min}$ (mm)	3,21
	$r_{1,max}$ (mm)	3,33	$r_{1,max}$ (mm)	3,33
Tooth flank radius	$r_{2,min}$ (mm)	18,29	$r_{2,min}$ (mm)	112,01
	$r_{2,max}$ (mm)	33,73	$r_{2,max}$ (mm)	1077,21
Chamfer	c_{min} (mm)	0,95	c_{min} (mm)	0,95
	c_{max} (mm)	1,43	c_{max} (mm)	1,43
Roller bed angle	χ_{min} (°)	115,9	χ_{min} (°)	119,4
	χ_{max} (°)	135,9	χ_{max} (°)	139,4
Tooth height above pitch polygon	k_{min} (mm)	1,59	k_{min} (mm)	1,59
	k_{max} (mm)	3,12	k_{max} (mm)	2,83
Final width	B_Y (mm)	5,32	B_Y (mm)	5,32

Table A- 4: Geometrical data of the alternative 2 (23).

Alternative 3:

Technical data of the alternative 3:

Power	P (kW)	3,5
Maximum speed of the motor	$n_a = n_1$ (min^{-1})	3500
Gear Ratio	i	6,6
Initial distance between centres	a_0 (mm)	425
Chain size and chain speed:		
Service Factor	f_1	1,5
Number of teeth of the small wheel	z_1	12
Teeth factor	f_2	1.5
Required number of teeth of the big wheel	z_{2erf}	79,2
Selected number of teeth of the big wheel	z_2	80
Corrected Power	P_D (kW)	7,875
Roller chain number		08 B
Pitch of the chain	p (mm)	12,7
Breaking load on the chain	F_B (kN)	18
Pressed surface	A (cm^2)	0,5
Load per meter	q (kg/m)	0,7
Chain speed	v (m/s)	8,890
Geometrical characteristics:		
Factor based on the number of links	f_3	117,13
Number of links	X_0	116,43
Rounded number of links	X	116
Distance factor	f_4	70
Final distance between centres	a (mm)	422,1
Forces		
Static tensile force of the chain	F (kN)	0,394
Dynamic force of the chain	F_d (kN)	0,591
Centrifugal force	F_f (kN)	0,055
Total force	F_g (kN)	0,646
static breaking strength	S_B	45,7

Is the static breaking strength greater than 7?		***Yes!***
Dynamic fracture resistance	S_D	27,9
Is the dynamic fracture resistance greater than 5?		***Yes!***

Table A- 5: Technical data of the alternative 3 (23).

Geometrical data of the alternative 3:

Dimensions and technical data of the roller chain:				
Pitch of the chain	p (mm)	12,7		
Roller width	b_1 (mm)	7,75		
Roller diameter	d_R (mm)	8,51		
Max. height of the plate	$g_1 = g$ (mm)	11,8		
Simple-Double-Triple chain:				
Number of juxtaposed chains	Y	1		
Distance	e (mm)	13,92		
Detailed dimensions of gears for roller chains:				
Wheel chamfer	$r_{4,min}$ (mm)	0,3		
	$r_{4,max}$ (mm)	1,6		
Tooth width	B_1 (mm)	7,21		
Number of teeth of the sprocket	**z**	**12**	**z**	**80**
Sprocket dimensions				
Pitch angle	t (°)	30,0	t (°)	4,5
	t (rad)	0,524	t (rad)	0,079
Pitch diameter	d (mm)	49,07	d (mm)	323,49
Root diameter	d_f (mm)	40,56	d_f (mm)	314,98
Tip diameter	$d_{a,min}$ (mm)	54,95	$d_{a,min}$ (mm)	327,93
	$d_{a,max}$ (mm)	56,43	$d_{a,max}$ (mm)	330,85
Diameter of free rotation	$d_{s,min}$ (mm)	30,81	$d_{s,min}$ (mm)	306,65
	$d_{s,max}$ (mm)	33,41	$d_{s,max}$ (mm)	309,25
Roller bed radius	$r_{1,min}$ (mm)	4,30	$r_{1,min}$ (mm)	4,30
	$r_{1,max}$ (mm)	4,44	$r_{1,max}$ (mm)	4,44

Tooth flank radius	$r_{2,min}$ (mm)	14,30	$r_{2,min}$ (mm)	83,74
	$r_{2,max}$ (mm)	22,06	$r_{2,max}$ (mm)	447,97
Chamfer	c_{min} (mm)	1,27	c_{min} (mm)	1,27
	c_{max} (mm)	1,91	c_{max} (mm)	1,91
Roller bed angle	χ_{min} (°)	112,5	χ_{min} (°)	118,9
	χ_{max} (°)	132,5	χ_{max} (°)	138,9
Tooth height above pitch polygon	k_{min} (mm)	2,10	k_{min} (mm)	2,10
	k_{max} (mm)	4,53	k_{max} (mm)	3,81
Final width	B_Y (mm)	7,21	B_Y (mm)	7,21

Table A- 6: Geometrical data of the alternative 3 (23).

Alternative 4:

Technical data of the alternative 4:

Power	P (kW)	3,5
Maximum speed of the motor	$n_a = n_1$ (min^{-1})	3500
Gear Ratio	i	6,6
Initial distance between centres	a_0 (mm)	425
Chain size and chain speed:		
Service Factor	f_1	1,5
Number of teeth of the small wheel	z_1	15
Teeth factor	f_2	1,29
Required number of teeth of the big wheel	z_{2erf}	99
Selected number of teeth of the big wheel	z_2	99
Corrected Power	P_D (kW)	6,773
Roller chain number		08 B
Pitch of the chain	p (mm)	12,7
Breaking load on the chain	F_B (kN)	18
Pressed surface	A (cm^2)	0,5
Load per meter	q (kg/m)	0,7
Chain speed	v (m/s)	11,113

Geometrical characteristics:		
Factor based on the number of links	f_3	178,73
Number of links	X_0	130
Rounded number of links	X	130
Distance factor	f_4	73
Final distance between centres	a (mm)	430
Forces		
Static tensile force of the chain	F (kN)	0,315
Dynamic force of the chain	F_d (kN)	0,472
Centrifugal force	F_f (kN)	0,086
Total force	F_g (kN)	0,559
static breaking strength	S_B	57,2
Is the static breaking strength greater than 7?		***Yes!***
Dynamic fracture resistance	S_D	32,2
Is the dynamic fracture resistance greater than 5?		***Yes!***

Table A- 7: Technical data of the alternative 4 (23).

Geometrical data of the alternative 4:

Dimensions and technical data of the roller chain:				
Pitch of the chain	p (mm)	12,7		
Roller width	b_1 (mm)	7,75		
Roller diameter	d_R (mm)	8,51		
Max. height of the plate	$g_1 = g$ (mm)	11,8		
Simple-Double-Triple chain:				
Number of juxtaposed chains	Y	1		
Distance	e (mm)	13,92		
Detailed dimensions of gears for roller chains:				
Wheel chamfer	$r_{4,min}$ (mm)	0,3		
	$r_{4,max}$ (mm)	1,6		
Tooth width	B_1 (mm)	7,21		
Number of teeth of the sprocket	**z**	**15**	**z**	**99**

Sprocket dimensions				
Pitch angle	t (°)	24,0	t (°)	3,6
	t (rad)	0,419	t (rad)	0,063
Pitch diameter	d (mm)	61,08	d (mm)	400,28
Root diameter	d_f (mm)	52,57	d_f (mm)	391,77
Tip diameter	$d_{a,min}$ (mm)	66,63	$d_{a,min}$ (mm)	404,67
	$d_{a,max}$ (mm)	68,45	$d_{a,max}$ (mm)	407,64
Diameter of free rotation	$d_{s,min}$ (mm)	43,16	$d_{s,min}$ (mm)	383,49
	$d_{s,max}$ (mm)	45,76	$d_{s,max}$ (mm)	386,09
Roller bed radius	$r_{1,min}$ (mm)	4,30	$r_{1,min}$ (mm)	4,30
	$r_{1,max}$ (mm)	4,44	$r_{1,max}$ (mm)	4,44
Tooth flank radius	$r_{2,min}$ (mm)	17,36	$r_{2,min}$ (mm)	103,14
	$r_{2,max}$ (mm)	27,57	$r_{2,max}$ (mm)	679,51
Chamfer	c_{min} (mm)	1,27	c_{min} (mm)	1,27
	c_{max} (mm)	1,91	c_{max} (mm)	1,91
Roller bed angle	χ_{min} (°)	114,0	χ_{min} (°)	119,1
	χ_{max} (°)	134,0	χ_{max} (°)	139,1
Tooth height above pitch polygon	k_{min} (mm)	2,10	k_{min} (mm)	2,10
	k_{max} (mm)	4,36	k_{max} (mm)	3,79
Final width	B_Y (mm)	7,21	B_Y (mm)	7,21

Table A- 8: Geometrical data of the alternative 4 (23).

Alternative 5:

Technical data of the alternative 5:

Power	P (kW)	3,5
Maximum speed of the motor	$n_a = n_1$ (min^{-1})	3500
Gear Ratio	i	6,6
Initial distance between centres	a_0 (mm)	425
Chain size and chain speed:		
Service Factor	f_1	1,5
Number of teeth of the small wheel	z_1	12
Teeth factor	f_2	1,64
Required number of teeth of the big wheel	z_{2erf}	79,2
Selected number of teeth of the big wheel	z_2	80
Corrected Power	P_D (kW)	8,610
Roller chain number		10 B
Pitch of the chain	p (mm)	15,875
Breaking load on the chain	F_B (kN)	22,4
Pressed surface	A (cm^2)	0,67
Load per meter	q (kg/m)	0,95
Chain speed	v (m/s)	11,113
Geometrical characteristics:		
Factor based on the number of links	f_3	117,13
Number of links	X_0	104,5
Rounded number of links	X	104
Distance factor	f_4	58
Final distance between centres	a (mm)	425,7
Forces		
Static tensile force of the chain	F (kN)	0,315
Dynamic force of the chain	F_d (kN)	0,472
Centrifugal force	F_f (kN)	0,117
Total force	F_g (kN)	0,590
static breaking strength	S_B	71,1

Is the static breaking strength greater than 7?		***Yes!***
Dynamic fracture resistance	S_D	38,0
Is the dynamic fracture resistance greater than 5?		***Yes!***

Table A- 9: Technical data of the alternative 5 (23).

Geometrical data of the alternative 5:

Dimensions and technical data of the roller chain:				
Pitch of the chain	p (mm)	15,875		
Roller width	b_1 (mm)	9,65		
Roller diameter	d_R (mm)	10,16		
Max. height of the plate	$g_1 = g$ (mm)	14,7		
Simple-Double-Triple chain:				
Number of juxtaposed chains	Y	1		
Distance	e (mm)	16,59		
Detailed dimensions of gears for roller chains:				
Wheel chamfer radius	$r_{4,min}$ (mm)	0,3		
	$r_{4,max}$ (mm)	1,6		
Tooth width	B_1 (mm)	9,17		
Number of teeth of the sprocket	**z**	**12**	**z**	**80**
Sprocket dimensions				
Pitch angle	t (°)	30,0	t (°)	4,5
	t (rad)	0,524	t (rad)	0,079
Pitch diameter	d (mm)	61,34	d (mm)	404,36
Root diameter	d_f (mm)	51,18	d_f (mm)	394,20
Tip diameter	$d_{a,min}$ (mm)	69,17	$d_{a,min}$ (mm)	410,39
	$d_{a,max}$ (mm)	71,02	$d_{a,max}$ (mm)	414,04
Diameter of free rotation	$d_{s,min}$ (mm)	39,61	$d_{s,min}$ (mm)	384,41
	$d_{s,max}$ (mm)	42,21	$d_{s,max}$ (mm)	387,01
Roller bed radius	$r_{1,min}$ (mm)	5,13	$r_{1,min}$ (mm)	5,13
	$r_{1,max}$ (mm)	5,28	$r_{1,max}$ (mm)	5,28
Tooth flank radius	$r_{2,min}$ (mm)	17,07	$r_{2,min}$ (mm)	99,97

	$r_{2,max}$ (mm)	26,33	$r_{2,max}$ (mm)	534,82
Chamfer	c_{min} (mm)	1,59	c_{min} (mm)	1,59
	c_{max} (mm)	2,38	c_{max} (mm)	2,38
Roller bed angle	χ_{min} (°)	112,5	χ_{min} (°)	118,9
	χ_{max} (°)	132,5	χ_{max} (°)	138,9
Tooth height above pitch polygon	k_{min} (mm)	2,86	k_{min} (mm)	2,86
	k_{max} (mm)	5,90	k_{max} (mm)	5,00
Final width	B_Y (mm)	9,17	B_Y (mm)	9,17

Table A- 10: Geometrical data of the alternative 5 (23).

Alternative 6:

Technical data of the alternative 2:

Power	P (kW)	3,5
Maximum speed of the motor	$n_a = n_1$ (min^{-1})	3500
Gear Ratio	i	6,6
Initial distance between centres	a_0 (mm)	425
Chain size and chain speed:		
Service Factor	f_1	1,5
Number of teeth of the small wheel	z_1	21
Teeth factor	f_2	0,85
Required number of teeth of the big wheel	z_{2erf}	138,6
Selected number of teeth of the big wheel	z_2	138
Corrected Power	P_D (kW)	4,463
Roller chain number		06 B
Pitch of the chain	p (mm)	9,525
Breaking load on the chain	F_B (kN)	9
Pressed surface	A (cm^2)	0,28
Load per meter	q (kg/m)	0,41
Chain speed	v (m/s)	11,668

Geometrical characteristics:		
Factor based on the number of links	f_3	346,75
Number of links	X_0	175,55
Rounded number of links	X	176
Distance factor	f_4	95,5
Final distance between centres	a (mm)	422,3
Forces		
Static tensile force of the chain	F (kN)	0,300
Dynamic force of the chain	F_d (kN)	0,450
Centrifugal force	F_f (kN)	0,056
Total force	F_g (kN)	0,506
static breaking strength	S_B	30,0
Is the static breaking strength greater than 7?		***Yes!***
Dynamic fracture resistance	S_D	17,8
Is the dynamic fracture resistance greater than 5?		***Yes!***

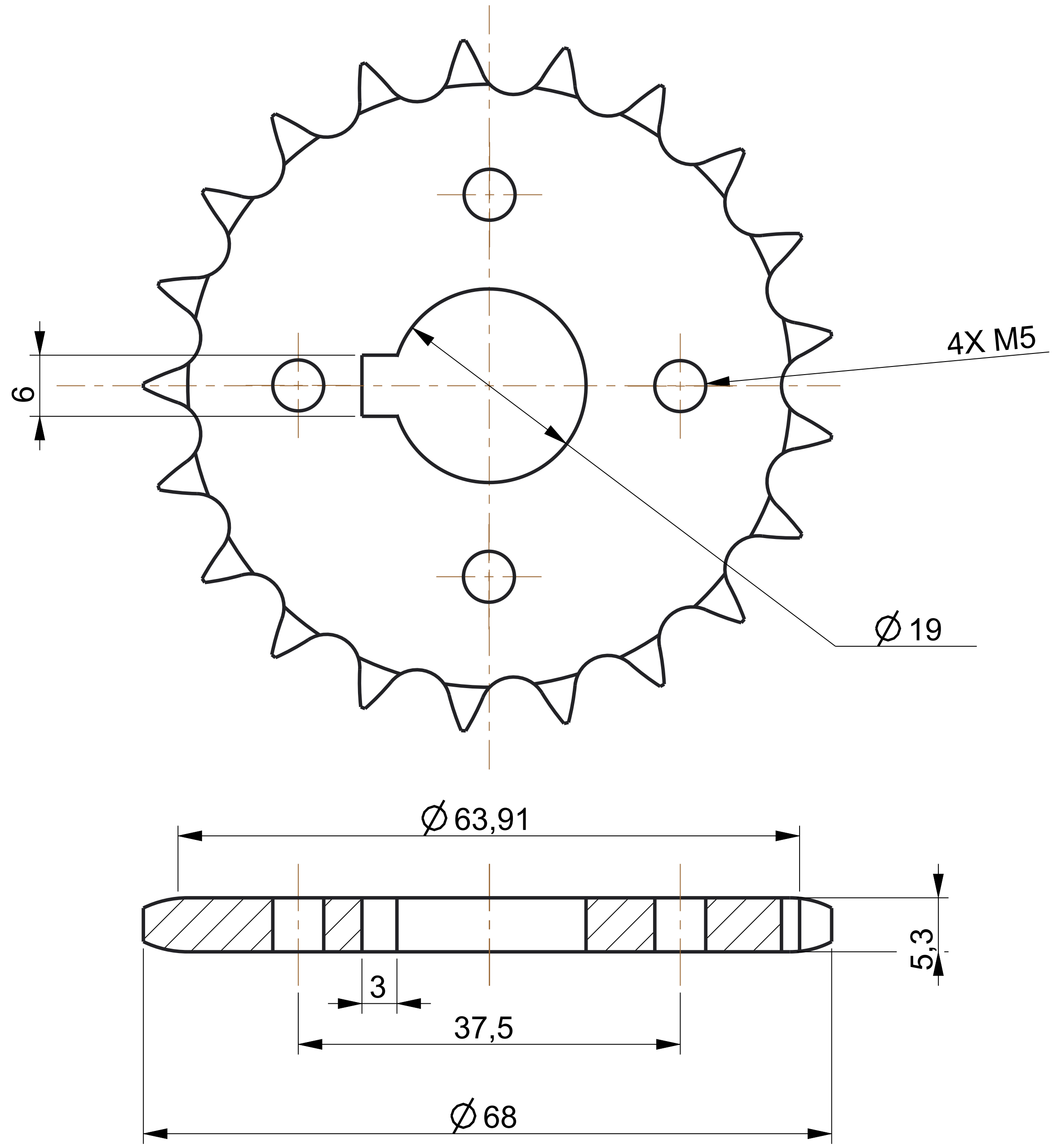
Table A- 11: Technical data of the alternative 6 (23).

Geometrical data of the alternative 6:

Dimensions and technical data of the roller chain:				
Pitch of the chain	p (mm)	9,525	p (mm)	9,525
Roller width	b_1 (mm)	5,72	b_1 (mm)	5,72
Roller diameter	d_R (mm)	6,35	d_R (mm)	6,35
Max. height of the plate	$g_1 = g$ (mm)	8,2	$g_1 = g$ (mm)	8,2
Simple-Double-Triple chain:				
Number of juxtaposed chains	Y	1	Y	1
Distance	e (mm)	10,24	e (mm)	10,24
Detailed dimensions of gears for roller chains:				
Wheel chamfer	$r_{4,min}$ (mm)	0,2	$r_{4,min}$ (mm)	0,2
	$r_{4,max}$ (mm)	1	$r_{4,max}$ (mm)	1
Tooth width	B_1 (mm)	5,32	B_1 (mm)	5,32

Number of teeth of the sprocket	z	21	z	138
Sprocket dimensions				
Pitch angle	t (°)	17,1	t (°)	2,6
	t (rad)	0,299	t (rad)	0,046
Pitch diameter	d (mm)	<u>63,91</u>	d (mm)	<u>418,44</u>
Root diameter	d_f (mm)	57,56	d_f (mm)	412,09
Tip diameter	$d_{a,min}$ (mm)	67,81	$d_{a,min}$ (mm)	421,72
	$d_{a,max}$ (mm)	69,46	$d_{a,max}$ (mm)	423,99
Diameter of free rotation	$d_{s,min}$ (mm)	51,58	$d_{s,min}$ (mm)	406,72
	$d_{s,max}$ (mm)	53,18	$d_{s,max}$ (mm)	408,32
Roller bed radius	$r_{1,min}$ (mm)	3,21	$r_{1,min}$ (mm)	3,21
	$r_{1,max}$ (mm)	3,33	$r_{1,max}$ (mm)	3,33
Tooth flank radius	$r_{2,min}$ (mm)	17,53	$r_{2,min}$ (mm)	106,68
	$r_{2,max}$ (mm)	31,55	$r_{2,max}$ (mm)	976,58
Chamfer	c_{min} (mm)	0,95	c_{min} (mm)	0,95
	c_{max} (mm)	1,43	c_{max} (mm)	1,43
Roller bed angle	χ_{min} (°)	115,7	χ_{min} (°)	119,3
	χ_{max} (°)	135,7	χ_{max} (°)	139,3
Tooth height above pitch polygon	k_{min} (mm)	1,59	k_{min} (mm)	1,59
	k_{max} (mm)	3,14	k_{max} (mm)	2,83
Final width	B_Y (mm)	<u>5,32</u>	B_Y (mm)	<u>5,32</u>

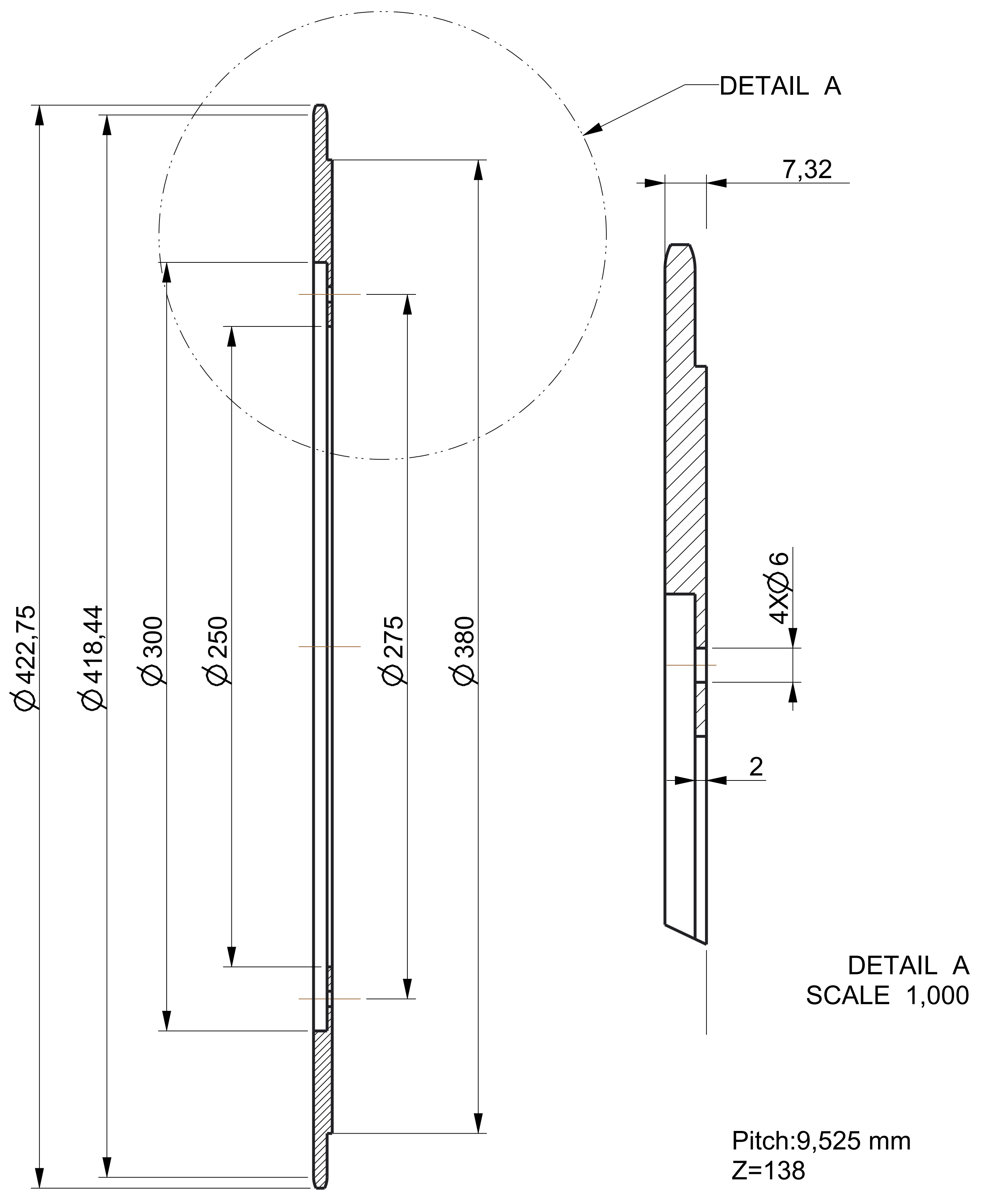
Table A- 12: Geometrical data of the alternative 6 (23).



Pitch: 9,525 mm
Z=21

description:

Layout notes:	model: Sprocket 21 teeth	type: PART	weight [kg]:	date: Jul-05-16
	file: Sprocket 21 teeth	sheet n°/n°: 1/1 A4	scale: 1,500	
	HSKA	drawn by: Jose Luis Areces		
		part number:	rev.:	final:



description:				
Layout notes:	model: Sprocket 138 teeth	type: PART	weight [kg]:	date: Jul-05-16
	file: Sprocket 138 teeth	sheet n°/n°: 1/1 A4	scale: 0.45	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:

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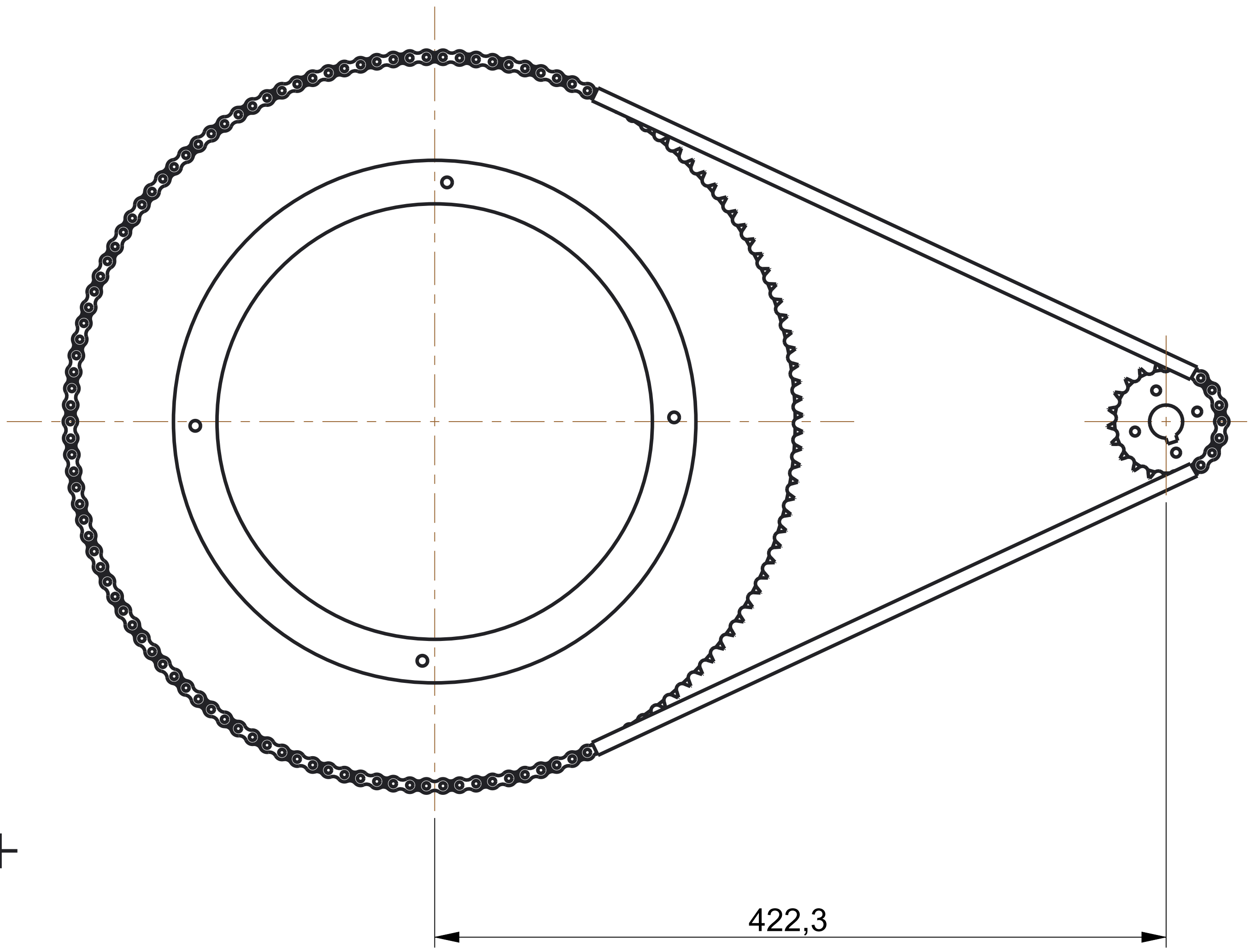
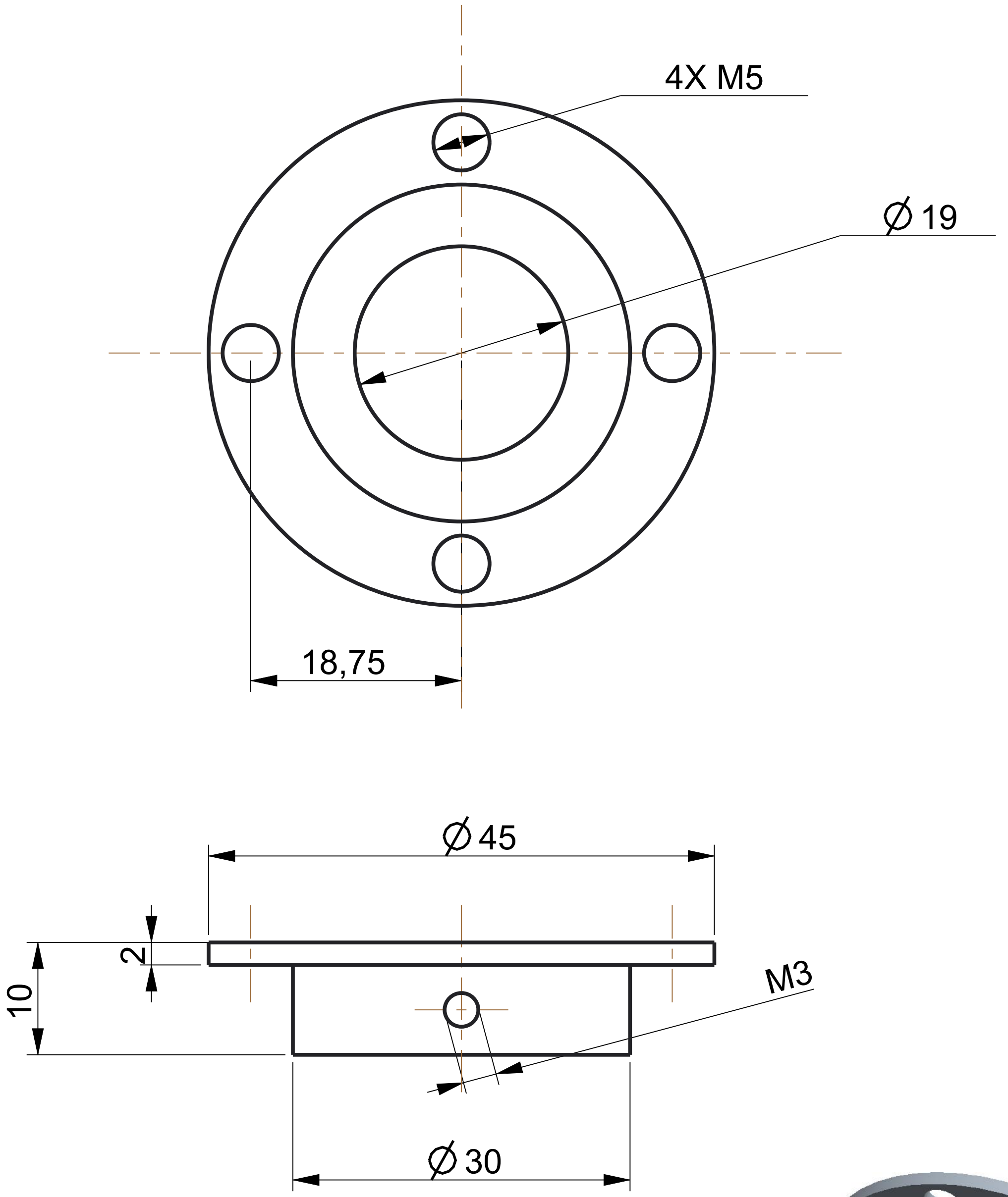
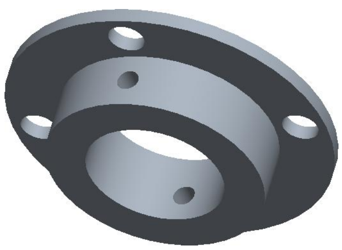


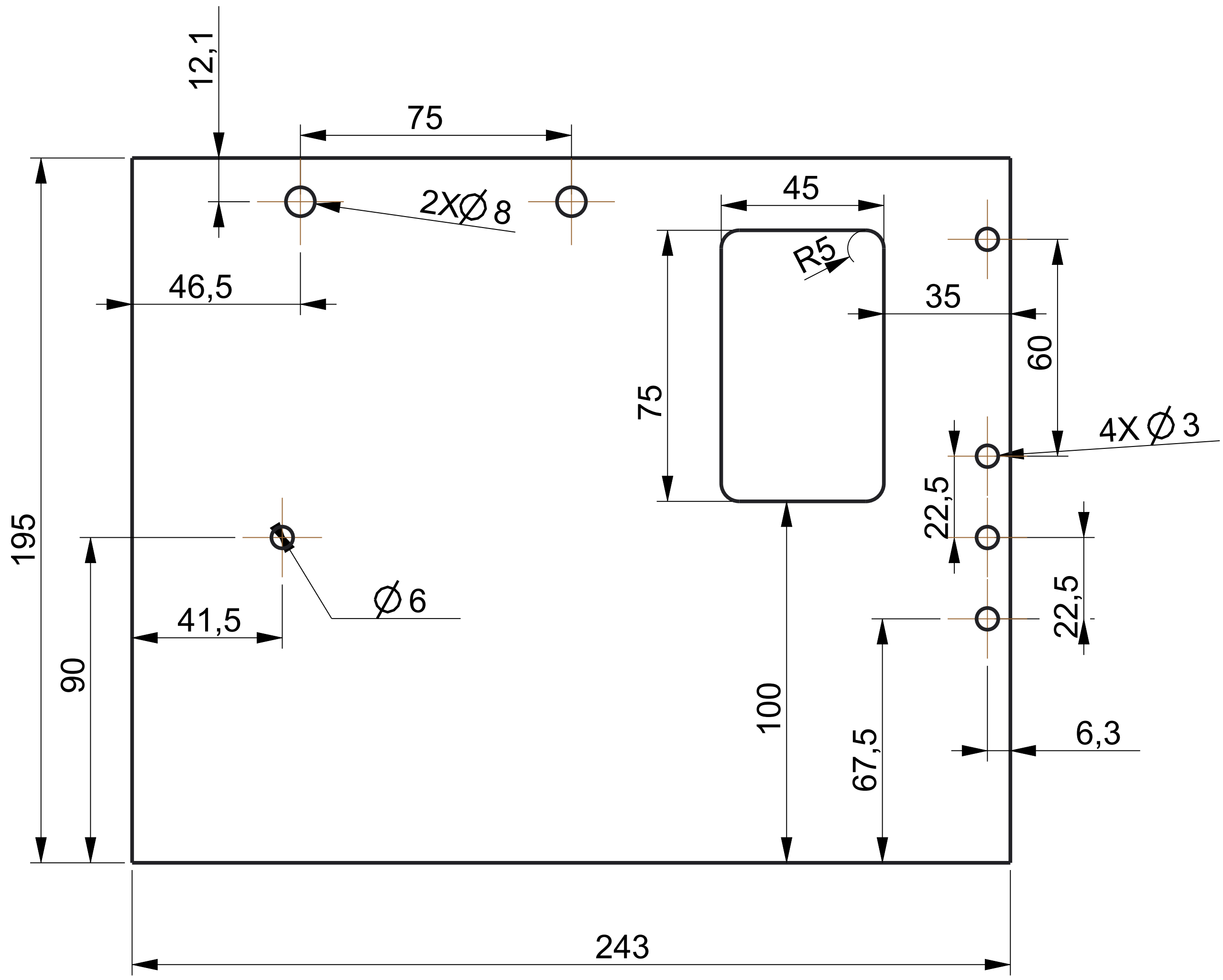
description:

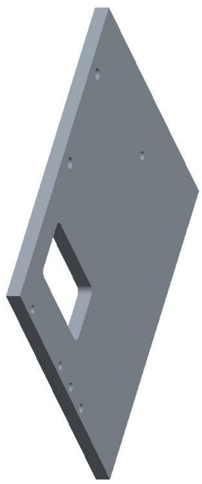
Layout notes:	model: ASSEMBLY_SPROCKETS_CHAIN	type: ASSEM	weight [kg]:	date: Jul-06-16
	file: ASSEMBLY_SPROCKETS_CHAIN	sheet n°/n°: 1/1 A4	scale: 0.250	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:





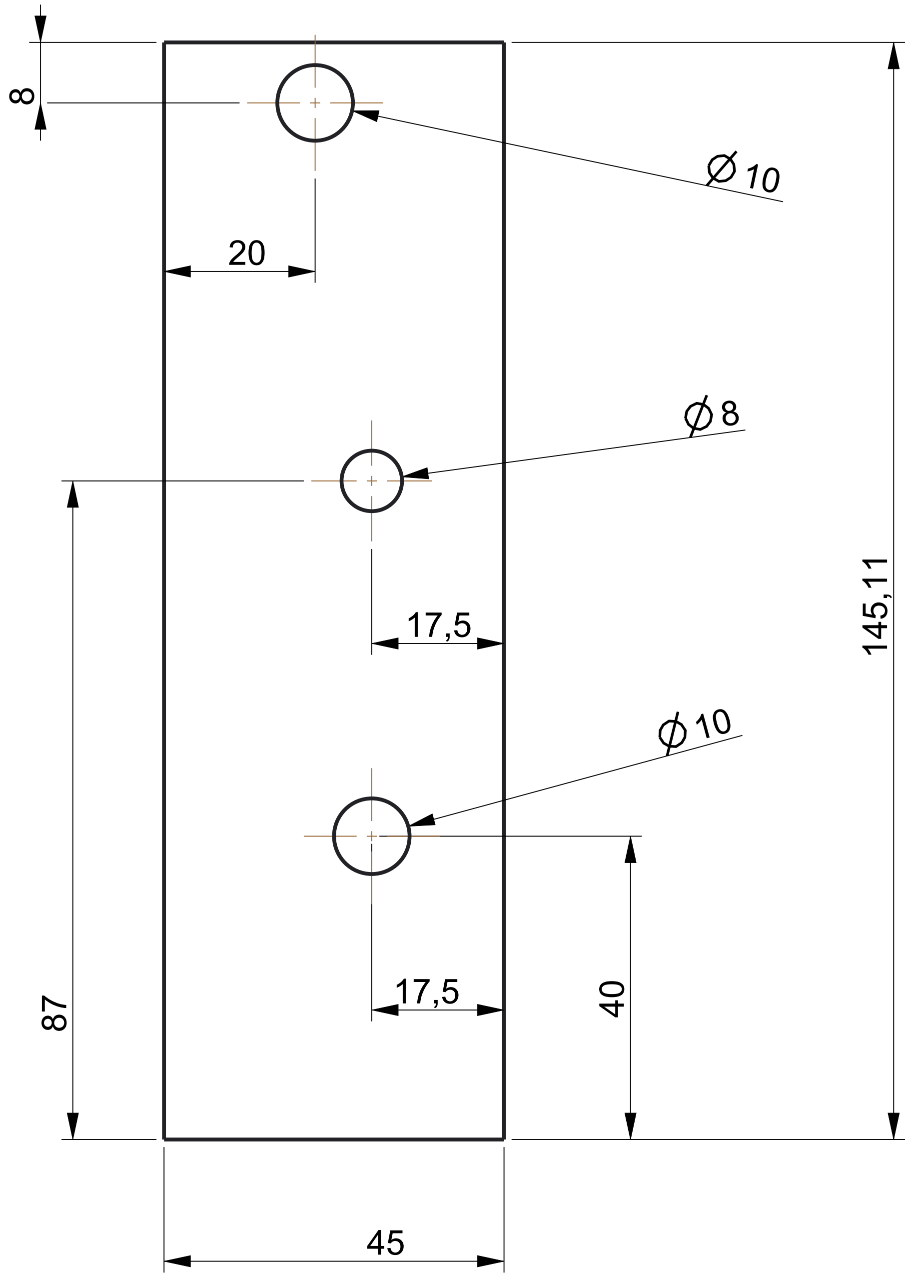
description:				
Layout notes:	model: FIX_SPROCKET	type: PART	weight [kg]:	date: Jul-06-16
	file: FIX_SPROCKET	sheet n°/n°: 1/1 A4	scale: 1.5	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:

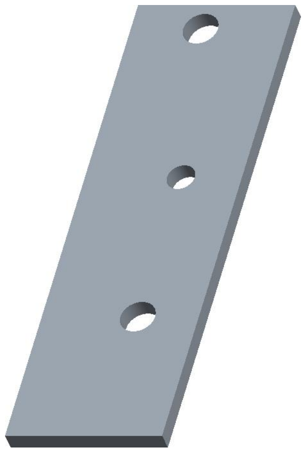




Width:10 mm

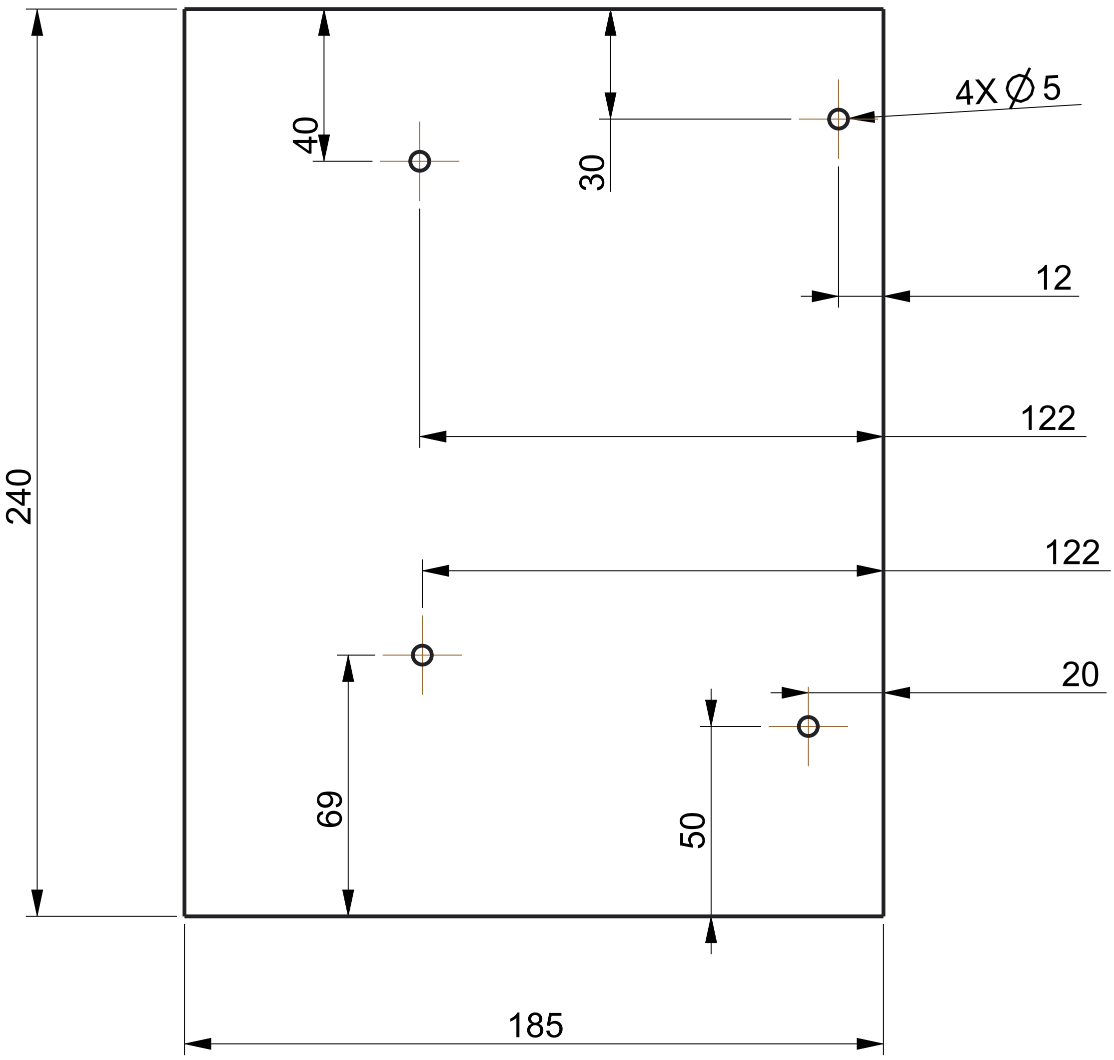
description:				
Layout notes:	model: FRAME_PART1_PIECE_1	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART1_PIECE_1	sheet n°/n°: 1/1 A4	scale: 0.5	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:



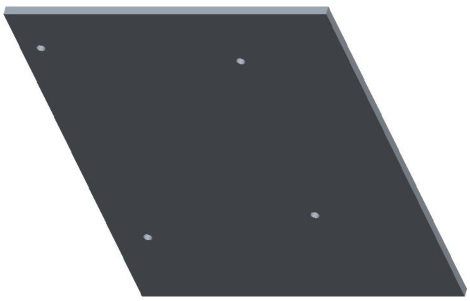


Width:5 mm

description:				
Layout notes:	model: FRAME_PART1_PIECE_2	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART1_PIECE_2	sheet n°/n°: 1/1 A4	scale: 1	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:



Width: 5 mm



description:				
Layout notes:	model: FRAME_PART1_PIECE_3	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART1_PIECE_3	sheet n°/n°: 1/1 A4	scale: 0.5	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:

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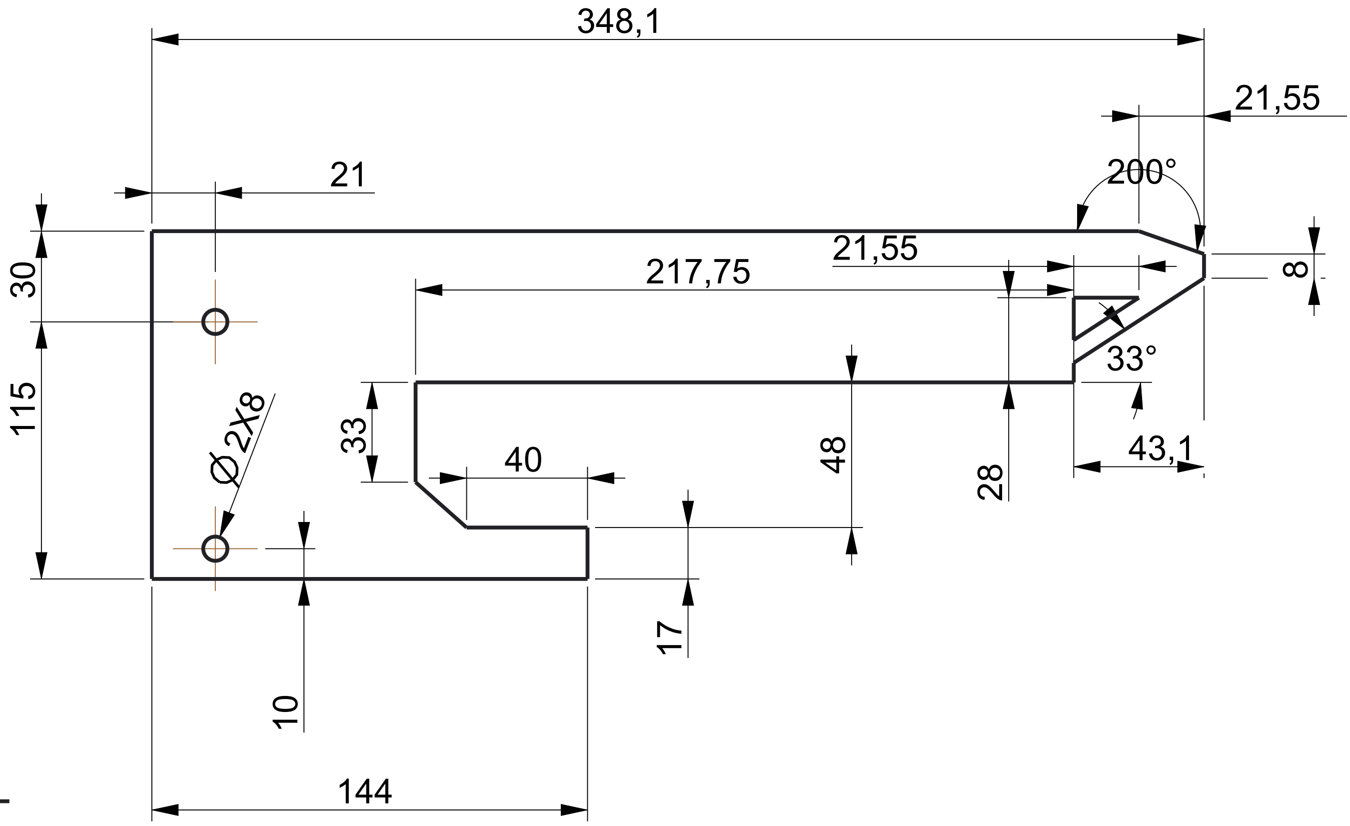
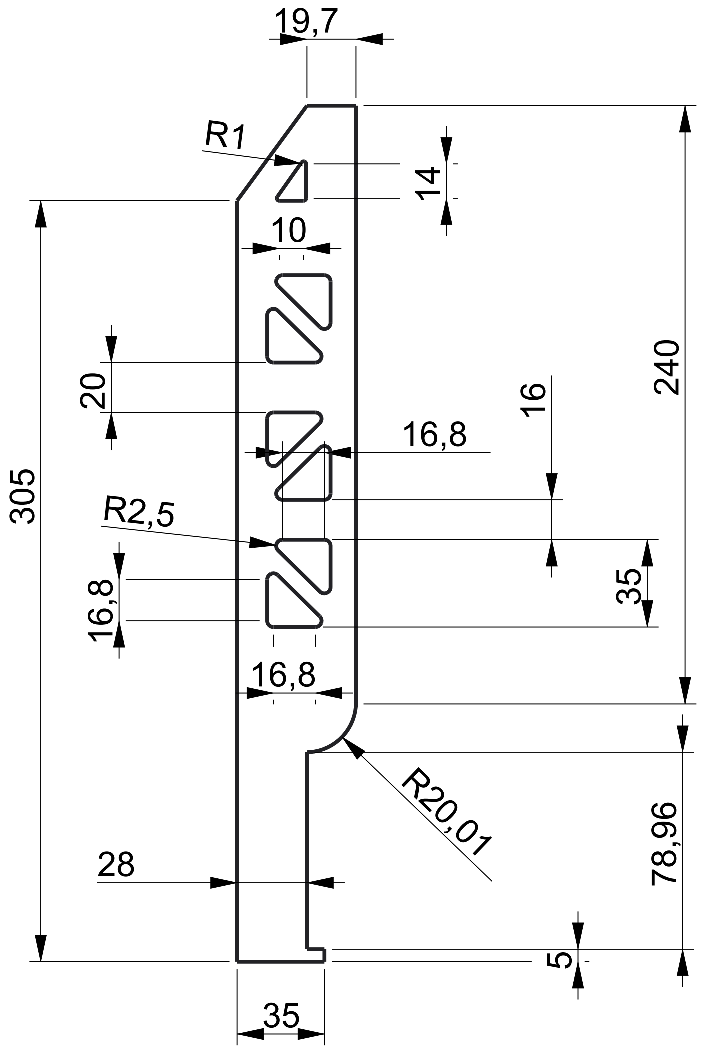
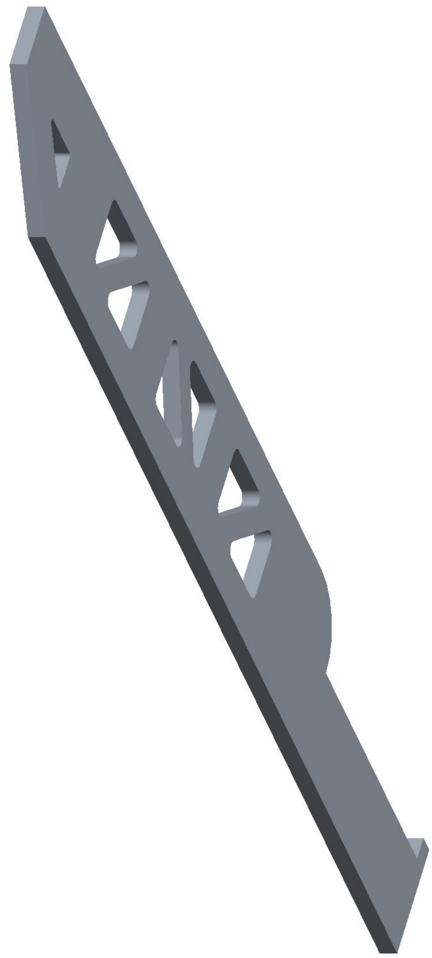


Width: 5 mm

description:

Layout notes:	model: FRAME_PART2_PIECE_1	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART2_PIECE_1	sheet n°/n°: 1/1 A4	scale: 0.4	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:

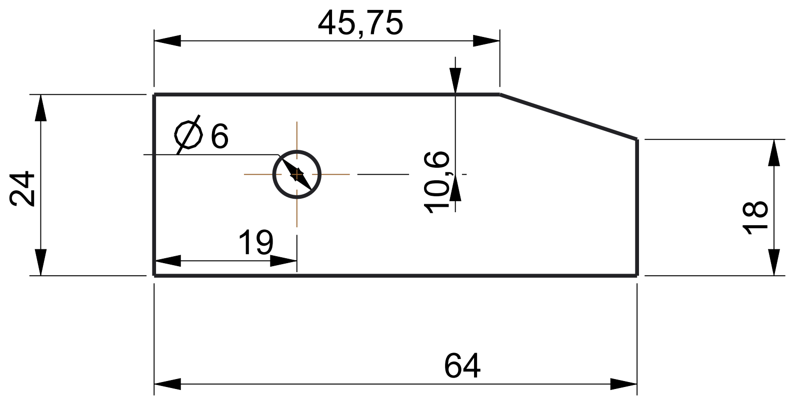


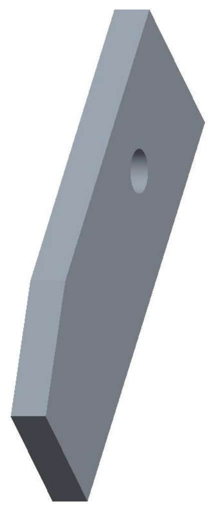


Width: 5 mm

description:

Layout notes:	model: FRAME_PART2_PIECE_2	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART2_PIECE_2	sheet n°/n°: 1/1 A4	scale: 0.33	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:





Widht: 5 mm

description:

Layout notes:	model: FRAME_PART2_PIECE_3	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART2_PIECE_3	sheet n°/n°: 1/1 A4	scale: 1,000	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:

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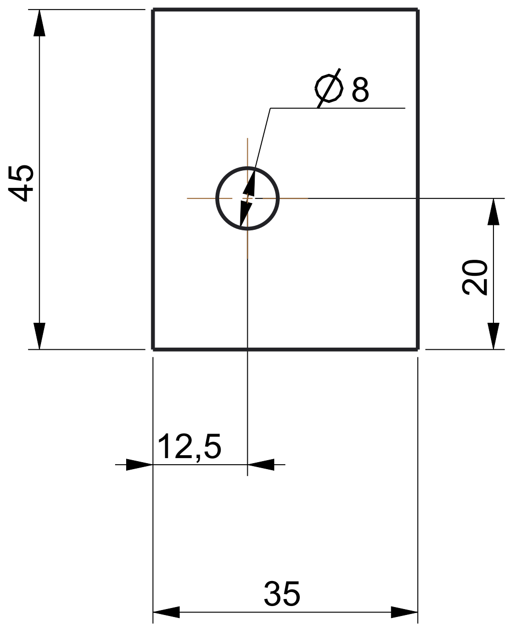

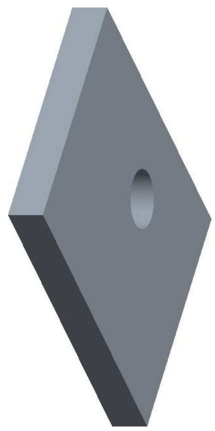

Width: 5 mm

description:				
Layout notes:	model: FRAME_PART2_PIECE_4	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART2_PIECE_4	sheet n°/n°: 1/1 A4	scale: 1,000	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:

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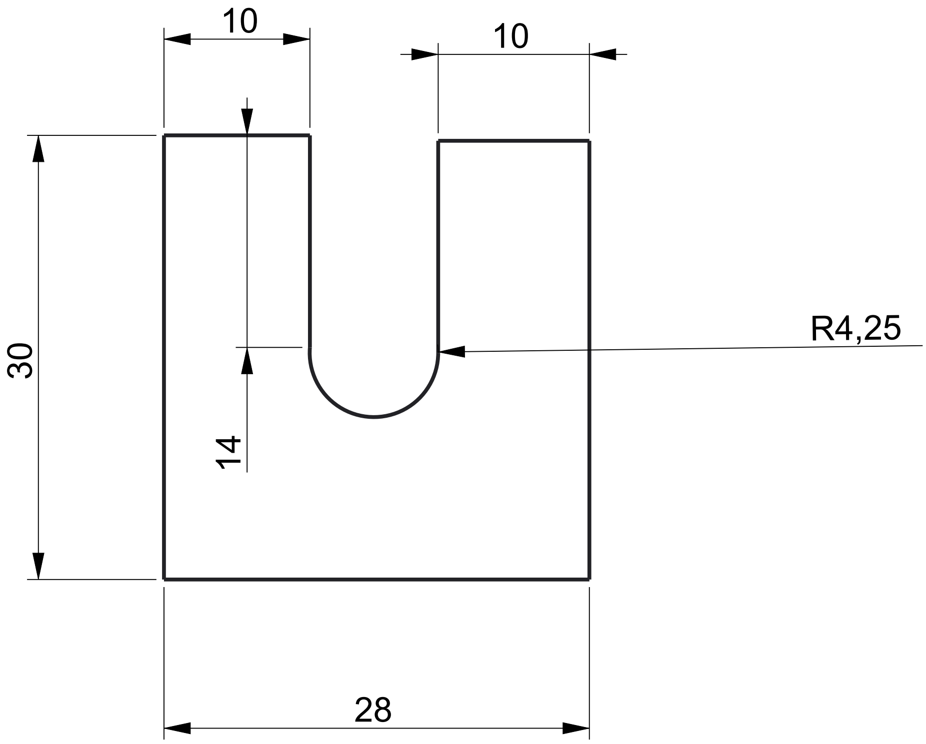
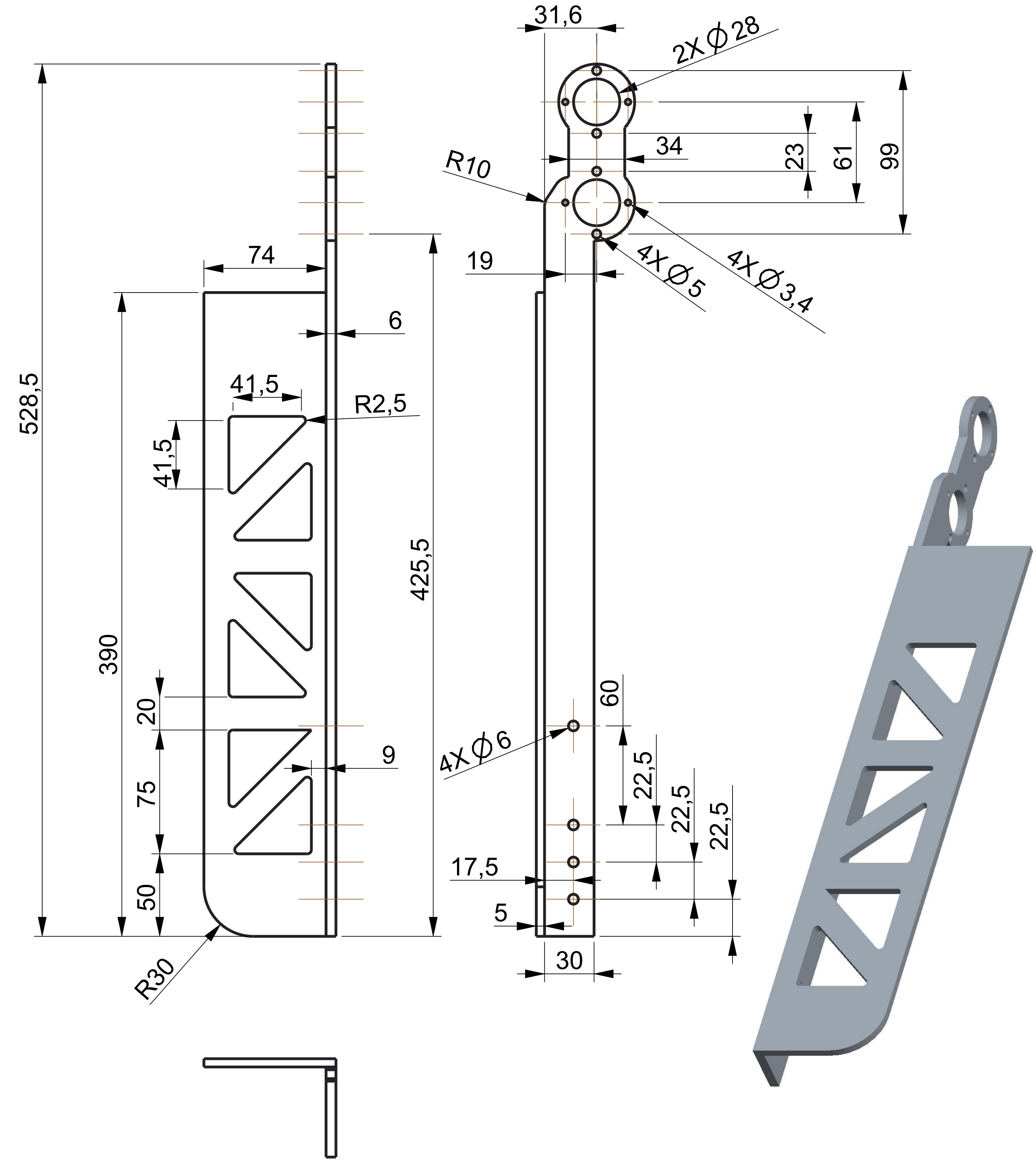


Width: 5 mm

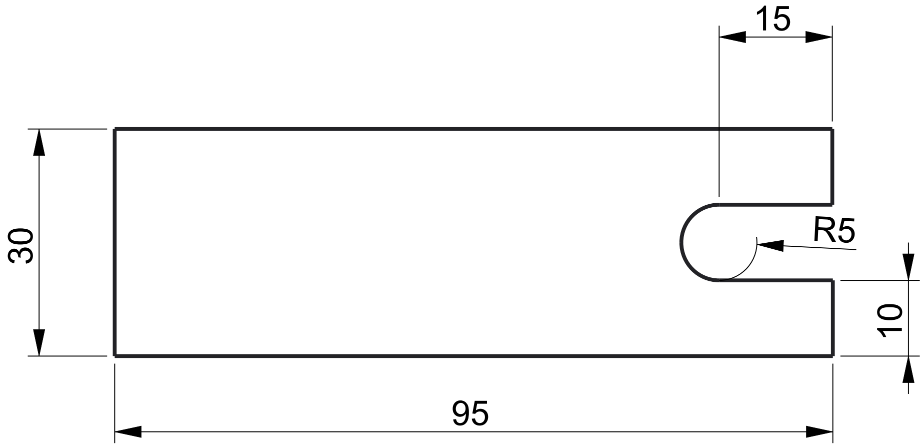
description:				
Layout notes:	model: FRAME_PART2_PIECE_5	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART2_PIECE_5	sheet n°/n°: 1/1 A4	scale: 2,000	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:



description:

Layout notes:

model:	type:	weight [kg]:	date:
FRAME_PART3_PIECE_1	PART		Jul-06-16
file: FRAME_PART3_PIECE_1	sheet n°/n°: 1/1 A4	scale: 0.3	
HSKA	drawn by: JOSE LUIS ARECES		
	part number:	rev.:	final:



Width: 5 mm

description:				
Layout notes:	model: FRAME_PART3_PIECE_2	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART3_PIECE_2	sheet n°/n°: 1/1 A4	scale: 1,000	
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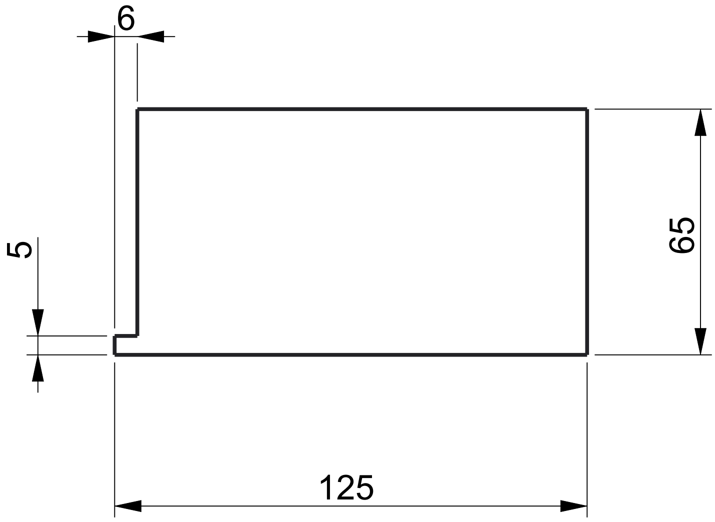
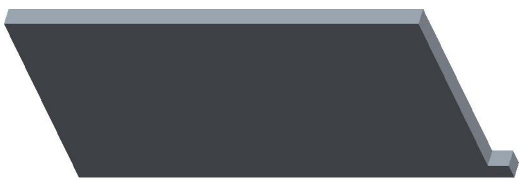
		part number:	rev.:	final:





Width: 5 mm

description:				
Layout notes:	model: FRAME_PART3_PIECE_3	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART3_PIECE_3	sheet n°/n°: 1/1 A4	scale: 0,500	
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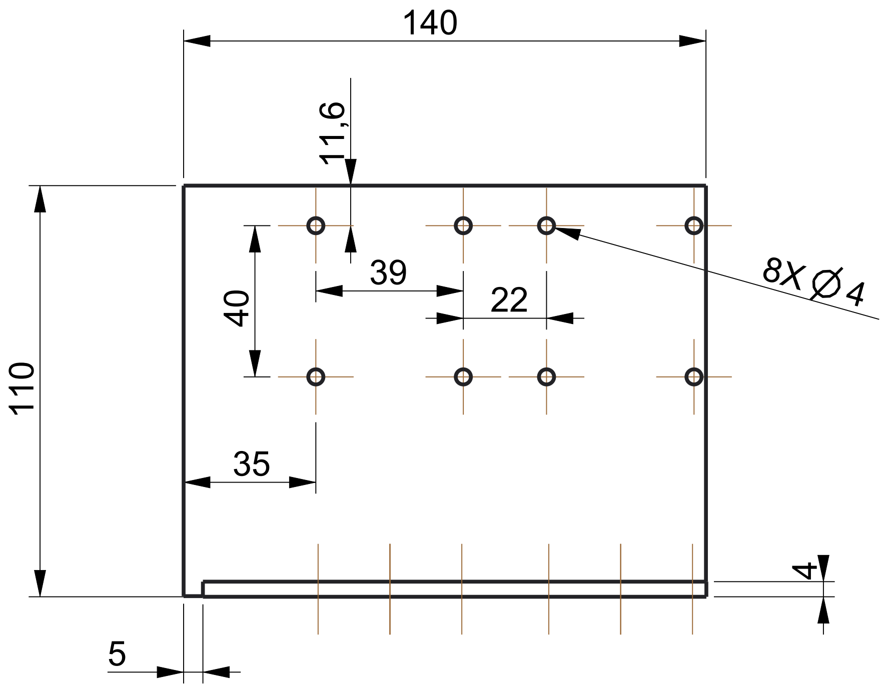

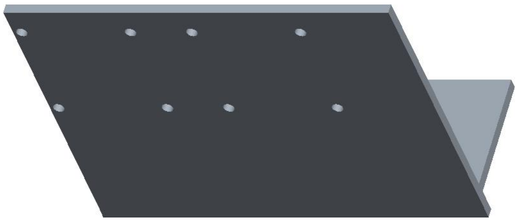

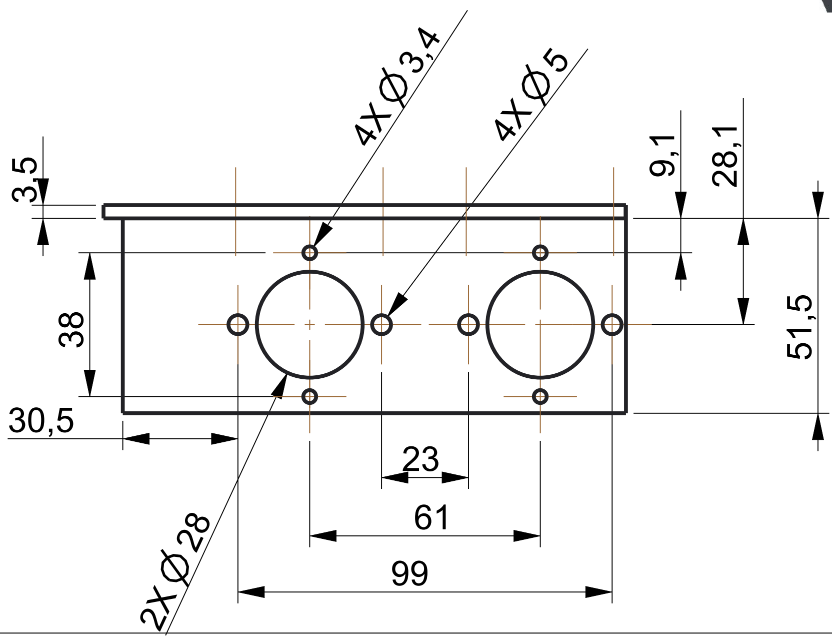

description:				
Layout notes:	model: FRAME_PART3_PIECE_4	type: PART	weight [kg]:	date: Jul-06-16
	file: FRAME_PART3_PIECE_4	sheet n°/n°: 1/1 A4	scale: 0,500	
	HSKA	drawn by: JOSE LUIS ARECES		
		part number:	rev.:	final:

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