HO CHI MINH CITY UNIVERSITY OF
TECHNOLOGY AND EDUCATION
FACULTY OF INTERNATIONAL EDUCATION

FINAL PROJECT

DESIGN AND CALCULATION FOR THE CLUTCH OF TOYOTA FORTUNER 2021

SUBJECT ID: VEDE320231E_23_1_02FIE

SEMESTER 1 – ACADEMIC YEAR 2023-2024

Implementation: Group 1

Guiding lecturer: Ph.D. Nguyễn Mạnh Cường

Ho Chi Minh City, November, 2023

HO CHI MINH CITY UNIVERSITY OF TECHNOLOGY AND EDUCATION VEHICLE DESIGN	THE SOCIALIST REPUBLIC OF VIETNAM Independence - Freedom - Happiness

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GROUP ASSIGNMENT FOR VEHICLE DESIGN FINAL PROJECT SEMESTER 1 - ACADEMIC YEAR 2023-2024

1. Subject ID: *VEDE320231E_23_1_ 02FIE*

2. Guiding lecturer: Ph.D. Nguyễn Mạnh Cường

3. Project: *Design and calculation for The Clutch of Toyota Fortuner 2021.*

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Novemver, 2023

GUIDING LECTURER

CONTENT

Abstract

This research study focuses on the design and calculation of the clutch system for the Toyota Fortune 2021 model. The clutch plays a crucial role in transmitting torque from the engine to the power transmission system efficiently and reliably. The objective of this research is to develop an optimized clutch design that meets the performance requirements and durability standards set by Toyota.

The research methodology involves a comprehensive analysis of the clutch system, considering factors such as torque capacity, frictional characteristics, engagement and disengagement speed, and overall system integration. Various design parameters, including clutch disc materials, pressure plate specifications, and actuation work, are taken into account to ensure optimal performance and longevity.

The research findings provide valuable insights into the design and calculation aspects of the clutch system for the Toyota Fortune 2021 model. The optimized clutch design not only ensures efficient power transmission but also enhances the driving experience by enabling smooth gear shifting and reducing wear on transmission components. The research outcomes contribute to the improvement of clutch systems in automotive engineering, leading to enhanced performance, durability, and customer satisfaction.

The link drive below includes AutoCAD files (dwg, pdf), Project Report (PDF, Word, PPT), Presentation video (mp4).

AUTOCAD - DESIGN AND CALCULATION FOR THE CLUTCH OF TOYOTA FORTUNER 2021

A. INTRODUCTION

1. Overview of The Clutch

The clutch is a vital component in mechanical systems that allows for the disconnection and connection of the input and output shafts. Its primary function is to control the transfer of power between the engine and the transmission or other working works. By engaging or disengaging the clutch, the driver or operator can control the transfer of rotational energy from the engine to the wheels or other driven components.

1.1. Functions of The Clutch

The clutch serves several important functions in mechanical systems, particularly in vehicles and machinery. Here are the primary functions of a clutch:

a. Power Transfer

The clutch facilitates the transfer of power from the engine (or motor) to the transmission or other driven works. When engaged, it allows the rotational energy generated by the engine to be transmitted to the wheels or other components, enabling movement or operation.

b. Gear Changes

In vehicles with manual transmissions, the clutch enables smooth gear changes by temporarily disengaging the engine from the transmission. This allows the driver to shift between different gears without causing gear grinding or damage to the transmission.

c. Smooth Engagement

The clutch enables smooth and gradual engagement of power between the engine and transmission. It helps prevent sudden jolts or jerks when starting a vehicle or engaging machinery, ensuring a more comfortable and controlled operation.

d. Safety and Protection

The clutch provides a safety work by allowing the driver to disengage the power source quickly in emergency situations. It helps prevent damage to the drivetrain or machinery by disconnecting the input and output shafts when necessary.

1.2. General composition of The Clutch

The construction of a clutch consists of three main parts:

a. The driving part includes: the flywheel, clutch cover, pressure plate, and the cover support.

b. The driven part includes: the friction disc and the driven shaft.

c. The control work is used to disengage the clutch when needed, and it includes: the pedal, connecting rod, sliding joint, levers, and release bearing.

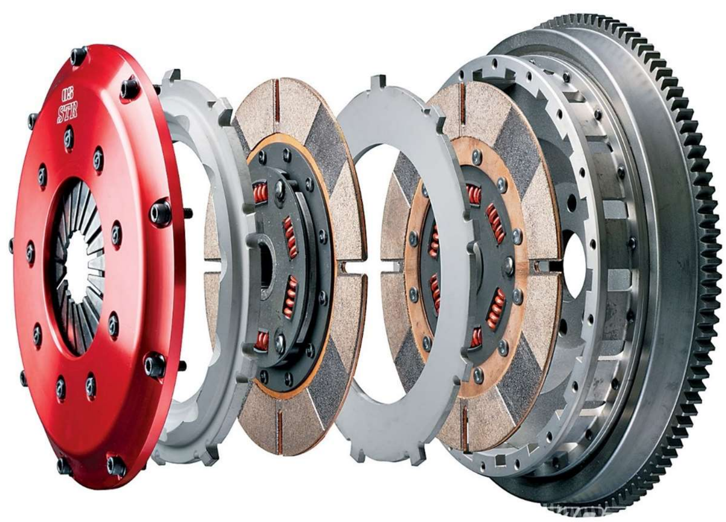


Figure 1: Clutch Construction

a. Clutch Disc

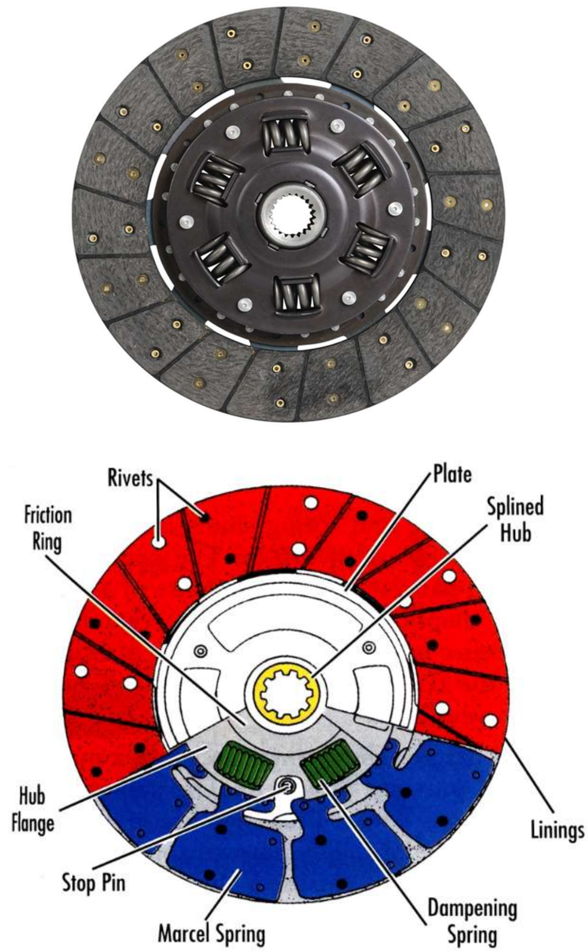


Figure 2;3: Friction disc

Also known as the friction disc or clutch plate, the clutch disc is a flat, circular plate typically made of friction material having a high coefficient of friction. It is mounted on the driven shaft and located between the flywheel (connected to the engine) and the pressure plate (connected to the transmission). The clutch disc has friction surfaces on both sides that engage with the flywheel and pressure plate, allowing for power transfer.

b. Flywheel

Figure 4: Flywheel

The flywheel is a heavy, disc-shaped component bolted to the engine's crankshaft. It is mounted on the crankshaft keeps on running as long as the engine keeps running and also provides a rotating mass that helps smooth out engine vibrations and maintains rotational momentum. The flywheel also has a friction surface where the clutch disc engages.

c. Pressure Plate

Figure 5: Pressure Plate

The pressure plate is a spring-loaded component bolted to the flywheel. It exerts pressure on the clutch disc, sandwiching it between the pressure plate and the flywheel. The pressure plate's role is to create frictional force and maintain a tight connection between the clutch disc and the flywheel.

d. Diaphragm Spring

Figure 7: Diaphram spring

The diaphragm is a sort of spring that has a circular shape. It aids in the retention of pressure on a friction plate. The outside part of the spring pushes outwards and presses the friction plate on the flywheel when the release bearing presses the inside half of the spring.

e. Clutch Release Bearing

Figure 9: Clutch release bearing

Also known as the throw-out bearing, the clutch release bearing is a cylindrical bearing that applies force to the pressure plate's release fingers. When the clutch pedal is pressed, the release bearing moves the pressure plate away from the clutch disc, disengaging the clutch.

f. Clutch Fork

Figure 10: Clutch fork

The clutch fork is a lever connected to the clutch pedal on one end and the clutch release bearing on the other. When the clutch pedal is depressed, the clutch fork transfers the force from the pedal to the release bearing, initiating clutch disengagement.

g. Clutch Pedal

Figure 6: Clutch pedal

The clutch pedal plays a role in generating hydraulic pressure in the master cylinder. This pressure acts on the clutch release cylinder and results in clutch engagement and disengagement. The free stroke of the clutch pedal refers to the distance the pedal can move until the clutch release bearing presses against the clutch disc spring.

h. Clutch Actuator Cylinder

Figure 8: Clutch actuator cylinder

The clutch release cylinder receives hydraulic pressure from the master cylinder to control the movement of the piston, thereby controlling the direction of clutch disengagement through the push rod.

1.3. Classification of The Cluch

Following these criteria, the clutch can be classified based on different characteristics and features, serving specific applications and requirements.

1.3.1. Based on the torque transmission

According to the method of transmitting torque from the engine crankshaft to the power transmission system, clutches are classified into the following types:

a. Friction Clutch

Figure 11: Friction Clutch

This type of clutch uses friction surfaces to transmit torque. It consists of friction components such as clutch disc and pressure plate. When the friction surfaces come into contact and press against each other, torque is transmitted from the engine crankshaft to the power transmission system.

b. Hydraulic Clutch

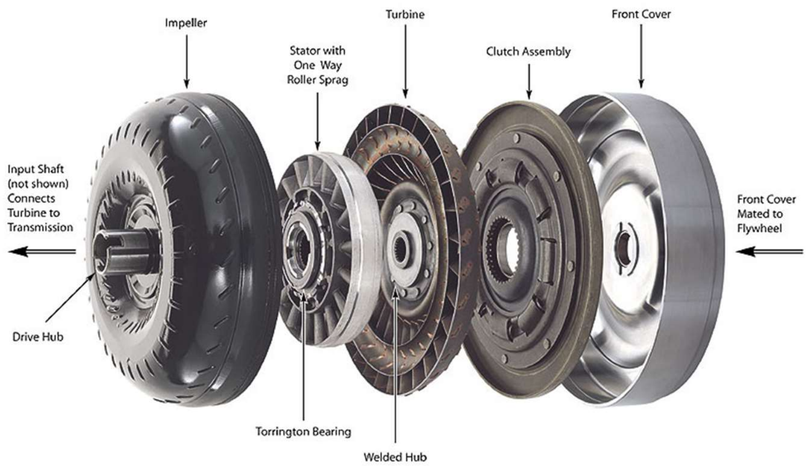


Figure 12: Hydraulic clutch

This type of clutch uses the energy of a fluid to transmit torque. It includes a fluid system, comprising a pump, cylinder, and control valve, to generate pressure and control the fluid within the clutch. When pressure is generated, the fluid transmits torque from the engine crankshaft to the power transmission system.

c. Electromagnetic Clutch

Figure 13: Electromagnetic clutch

This type of clutch utilizes the action of an electromagnetic field to transmit torque. It consists of electromagnetic magnets and electrical coils to generate the magnetic field. When the coils are activated and create the magnetic field, the interaction between the magnets and the field transmits torque.

1.3.2. Based on the operating condition

Clutches can be classified according to their operating states into the following types:

a. Disengaged Clutch

When the clutch is disengaged, there is no connection between the clutch disc and the pressure plate. This allows the torque from the engine to not be transmitted to the power transmission system.

b. Engaged Clutch

When the clutch is engaged, the clutch disc is tightly compressed between the pressure plate. This creates frictional force and transmits the torque from the engine to the power transmission system.

1.3.3. Based on the methods of generating pressure on the friction disc

a. The Spring-type Clutch

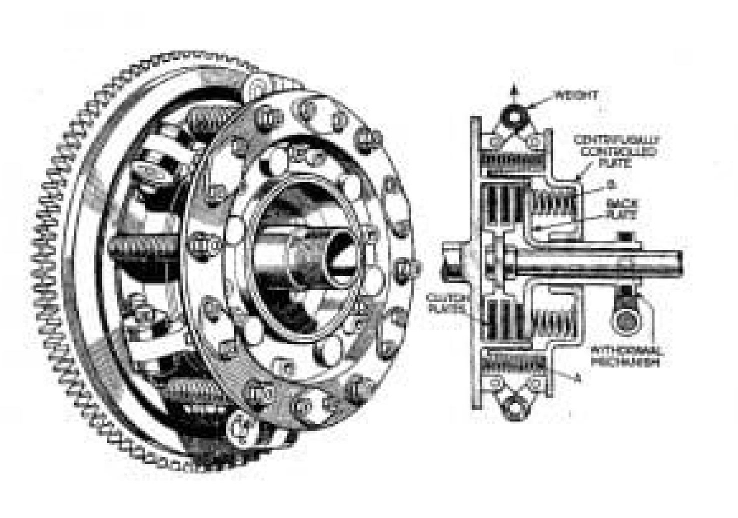


Figure 14: Spring-type clutch

The spring-type clutch is used to engage and disengage power transmission by utilizing the force exerted by springs. When the force from the springs is reduced, the clutch disc and pressure plate separate, disengaging the power transmission. When the force from the springs is increased, the clutch disc and pressure plate come into contact, engaging the power transmission.

b. The Semi-centrifugal Clutch

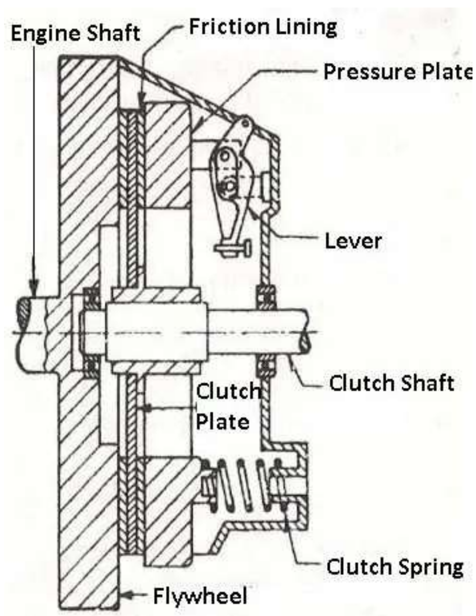


Figure 15: Semi-centrifugal clutch

The semi-centrifugal clutch, also known as a centripetal clutch, is a type of clutch that combines friction clutch and centrifugal clutch principles. The semi-centrifugal clutch provides automatic engagement and disengagement, allowing the rider to control the vehicle conveniently and safely *c. The Centrifugal Clutch*

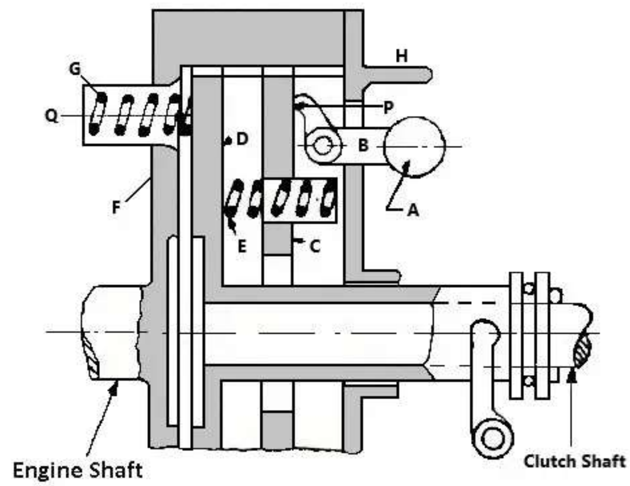


Figure 16: Centrifugal Clutch

This type of clutch operates based on centrifugal force. It is commonly used in applications where there is a need for automatic engagement and disengagement of the clutch.

1.3.4. Based on the number of plates

a. Single Plate Clutch

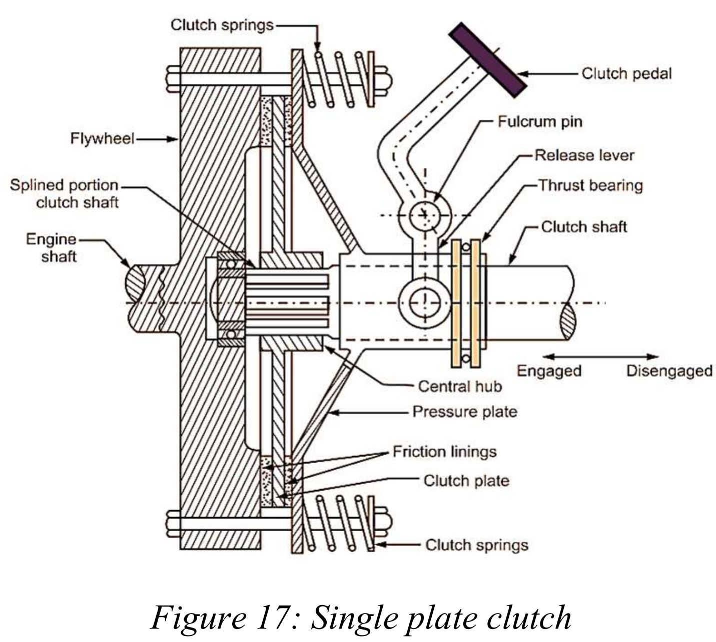


Figure 17: Single plate clutch

Single plate clutch is one of the most commonly used types of clutches used in most modern light vehicles. The clutch helps to transmit torque from the engine to the transmission input shaft. As the name states it has only one clutch plate.

It consists of a clutch plate, friction plate, pressure plate, flywheel, bearings, clutch spring and nut-bolts arrangement.

The single-plate clutch has only one plate which is attached on splines of the clutch plate. Single plate clutch is one of the main components of the clutch. The clutch plate is simply thin metallic disc which has both side friction surfaces.

The flywheel is attached on the engine crankshaft and rotates with it. A pressure plate is bolted to flywheel through clutch spring, which provides the axial force to keep the clutch engaged position, and is free to slide on the clutch shaft when the clutch pedal is operated.

A friction plate which is fixed between the flywheel and pressure plate. The friction lining is provided on both sides of the clutch plate.

Working principle

In a vehicle, we operate the clutch by pressing the clutch to peddle for disengagement of gears. Then springs get compressed and the pressure plate moves backwards. Now the clutch plate becomes free between the pressure plate and flywheel. Due to this now the clutch is getting disengaged and able to shift the gear.

This makes flywheel to rotate as long as the engine is running and the clutch shaft speed reduces slowly and then it stops rotating. As long as the clutch peddle is pressed, the clutch is said to be disengaged, otherwise, it remains engaged due to the spring forces. After releasing the clutch pedal the pressure plate comes back to its original position and clutch is again engaged.

b. Multi-Plate Clutch

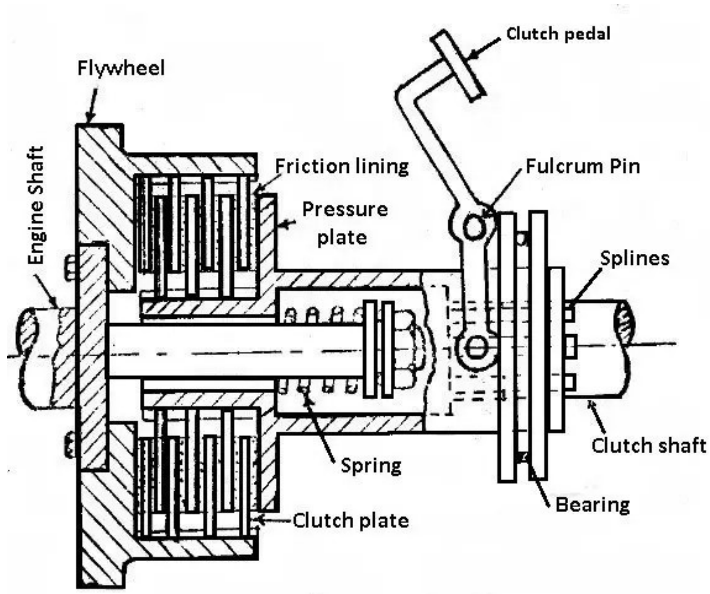


Figure 18: Multi-plate clutch

The multi-plate clutch is shown in the figure. These types of clutches use multiple clutches to make frictional contact with a flywheel of the engine. This makes transmit power between the engine shaft and the transmission shaft of a vehicle. The number of clutches means more friction surface.

The increased number of friction surfaces also increases the capacity of the clutch to transmit torque. The clutch plates are fitted to the engine shaft and gearbox shaft.

They are pressed by coil springs and assembled in a drum. Each of the alternate plates slides in grooves on the flywheel and the other slides on splines on the pressure plate. Hence, each different plate has an inner and outer spline.

The working principle of multiple clutches is the same as the working of the single-plate clutch. The clutch is operated by pressing the clutch pedal. The multiple clutches are used in heavy commercial vehicles, racing cars, and motorcycles for transmitting high torque.

The multiple clutches have two characters dry and wet. If the clutch is operated in an oil bath, it is known as a wet clutch. If the clutch is operated dry without oil, it is known as a dry clutch. The wet clutches are commonly used in connection with, or as a part of the automatic transmission.

1.3.5. Based on the clutch driving method

a. Mechanical-driving Clutch

This type of clutch uses power from the engine to transmit the drive through mechanical works. A common example is the friction clutch, where the clamping force is generated by the friction clutch work between the clutch disc and the pressure plate.

b. Hydraulic-driving Clutch

This type of clutch uses power from the flow of fluid (usually hydraulic oil) to transmit the drive. A hydraulic clutch includes components such as a pump, cylinder, and control valve to regulate the flow and pressure of the fluid, thereby operating the clutch.

c. Reinforced Clutch

- *Reinforced Mechanical-driving Clutch with Pneumatic Actuation*

This type of clutch uses compressed air as the power source to transmit the drive. It includes a pneumatic system to generate the necessary clamping force for engaging and disengaging the clutch.

- *Reinforced Hydraulic-driving Clutch with Pneumatic Actuation*

This type of clutch also uses compressed air but in a hydraulic environment. It utilizes a pneumatic system to generate clamping force and control the fluid in the hydraulic clutch for drive transmission.

1.4. Technical requirements

The clutch must be able to transmit the maximum torque of the engine under any working conditions. In other words, the friction torque of the clutch must always be greater than the maximum torque of the engine.

The process of disengaging the clutch must be decisive and quick. This means that when the clutch is disengaged, the passive part must be completely separated from the active part. This eliminates the inertial torque induced by the crankshaft and the engine torque from the clutch system when shifting gears, avoiding difficulties in gear shifting.

When engaging the clutch, a smooth operation is required. This means that the friction torque generated in the clutch must increase gradually when the clutch is engaged. This prevents jerking and tooth impact between the gears in the gearbox and other transmission components in the power transmission system.

The moment of inertia of the passive components of the clutch must be minimized to reduce the impact forces on the gear engagement (in cases without a synchronizer). This also reduces the working conditions of the synchronizer and speeds up the gear shifting time.

2. Some common clutch design options

2.1. Dry-friction clutch with a cylindrically-arranged spring-loaded passive disc

a. Composition

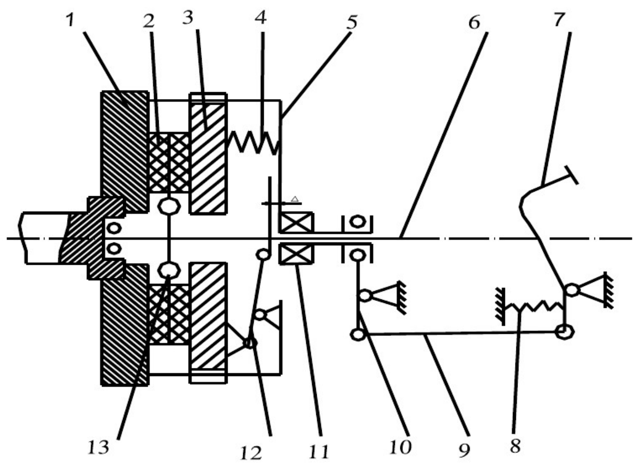


Figure 19: Dry-friction clutch with a cylindrically-arranged spring-loaded passive disc

1 – Flywheel	*8 – Torsion spring*
2 – Friction disc	*9 – Puller*
3 – Pressure plate	*10 – Wrench*
4 – Compression spring	*11 – T-ball*
5 – Clutch cover	*12 – Pliers*
6 – Open bushing	*13 – Dampers*
7 – Clutch pedal	

The active components includes: the flywheel, clutch cover, pressure plate, pliers, and compression springs. When the clutch is fully disengaged, the components in the active group will rotate together with the flywheel.

The passive components includes the driven disc (friction disc) and the driven shaft. When the clutch is fully disengaged, the components in the passive group will remain stationary.

According to the structural diagram in the illustration, the clutch cover is fixed to the flywheel using bolts, pressure plate can undergo translational movement within the housing and has a component for transmitting torque from the clutch cover to the pressure plate.

b. Working principle

When the clutch is disengaged, the thrust bearing presses against the lever, and the other end of the lever lifts the pressure plate away from the flywheel. This compresses the springs, causing the friction disc to move away from the flywheel, and power cannot be transmitted to the gearbox.

When the clutch is engaged, the thrust bearing moves away from the lever. As a result, the springs on the pressure plate push the pressure plate towards the rotating flywheel. This brings the friction disc closer to the flywheel, allowing power to be transmitted to the gearbox.

c. Advantages and disadvantages

About the composition, the dry-friction clutch with a cylindrically-arranged spring-loaded passive disc is simple and easily affordable. However, the size of the clutch cover is quite large, which is proportional to the torque of the engine.

About engaging and disengaging phase, the engaging phase is occurred completely and smoothly, and also releases heat well which ensures the lifespan of the clutch assembly. However, the disengaging phase is not smooth and cannot transmit large torque.

This type of clutch is commonly used for most of the cars nowadays, included stroke cars and light trucks because of the simple construction.

2.2. Dry-friction clutch with Diaphgram spring

a. Composition

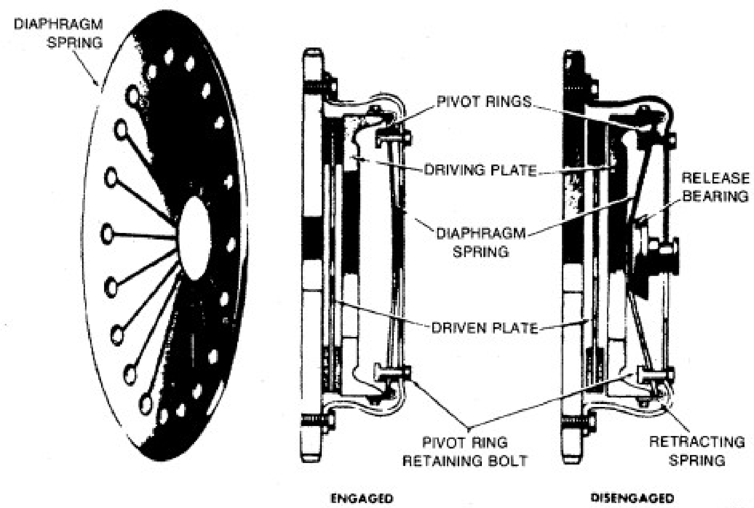


Figure 20: Diaphrgram clutch

The pressure plate commonly utilizes the diaphgram spring. Diaphgram spring functions similarly to coil springs or cylindrical spring. The diaphgram spring is a large curved spring that is either bent upwards or downwards and is divided into specific leaves connected from the outer edge to the inner hole. This spring is attached inside the pressure plate, with the outer edge tightly fitted against the rear surface of the pressure plate. A circular plate is attached behind the leaf spring to provide positioning for the outer edge of the spring.

b. Working principle

The diaphragm spring works when the center of is is pushed into the engine, the outer edge of the diaphragm spring moves in the opposite direction, away from the engine. This action separates the friction disc and the sliding plate from the flywheel.

When the center of the spring is released, the spring returns to its normal state. At that time, the outer edge of the pressure plate pushes the surface of the diaphragm spring towards the friction disc, engaging the clutch.

c. Advantages

Reducing size, weight, and simplifying the structure of the clutch assembly has several advantages. Without external assembly parts, balancing becomes easier. Additionally, centrifugal forces that reduce the pressure on the friction disc at high speeds are eliminated. The force acting on the disc remains consistent across all operating modes.

2.3. Dry-friction clutch with two cylindrically-arranged spring-loaded passive disc

a. Construction

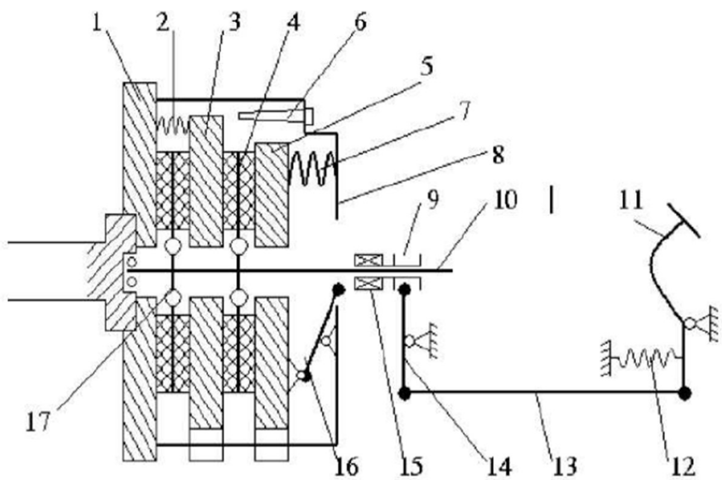


Figure 21: Dry-friction clutch with two cylindrically-arranged spring-loaded passive disc

1 – Flywheel	*7 – Compression spring*
2 – Intermediate plate spring	*8 – Clutch cover*
3 – Intermediate passive disc	*9 – Open bushing*
4 – Friction disc	*10 – Clutch shaft*
5 – Outer pressure disc	*11 – Clutch pedal*
6 – Restricter bolt	*12 – Torsion spring*

13 – Puller

14 – Wrench

15 – T-ball

16 – Pilers

17 – Dampers

When there is a need to transmit high torque with a compact layout, multiple-disc clutches are commonly used. In the case of heavy-duty trucks, a twin-disc clutch is often encountered.

b. Working principle

When the clutch is engaged, the pressure springs tightly compress the pressure plates and the driven plates against the flywheel. At this point, the springs are also compressed, and there are clearances between the ends of the adjustable pliers and the ball bearings on the release lever to ensure complete engagement of the clutch.

When the clutch is disengaged, the drive components will cause the pressure plate to move to the left, releasing the first driven plate. As the plate begins to make contact with the bolts, the first driven plate is completely separated. The pressure plate continues to move to the left, releasing the second driven plate. Only when both driven plates are fully released is the clutch fully disengaged.

c. Advantages and disadvantages

About the composition, the dry-friction clutch with two cylindrically-arranged spring-loaded passive disc is complicated because of having more friction discs. However, the size of the clutch cover is smaller as the result of the diameter of friction discs is smaller.

About engaging and disengaging phase, the operation is not occurred completely and smoothly due to more friction discs which need to be disengaged, and also leads to the delay of the operation. Moreover, it cannot release heat effectively.

This type of clutch is commonly used for race cars and heavy trucks.

2.4. Hydraulic clutch

The hydraulic clutch is a type of fluid coupling used to transmit rotational energy from the engine to the gearbox. The position of the hydraulic clutch is located between the engine and the gearbox, serving as a clutch in a manual transmission. The main function of the hydraulic clutch is to allow the separation of the load from the primary power source coming from the engine.

a. Composition

Pump Wheel/ Impeller

The pump blades are directly attached to the torque converter shell, and the torque converter shell is connected to the engine shaft. These are curved blades with an angled edge. The pump wheel assembly includes the transmission fluid, rotating with the engine speed. As it rotates with the engine, centrifugal force causes the fluid to move outward.

The blades of the pump wheel are designed to direct the fluid towards the turbine blades. It functions as a centrifugal pump, drawing fluid from the automatic transmission and delivering it to the turbine.

Stator

The main function of the stator is to redirect the fluid flow from the turbine back to the pump wheel, allowing the fluid to enter the pump wheel in the direction of its rotation. When the fluid enters in the direction of the pump wheel's rotation, it increases torque several times.

The stator changes the direction of the fluid by up to nearly 90 degrees. It is attached with a one-way clutch, allowing rotation in only one direction. The turbine is connected to the power transmission system, while the stator is located between the pump wheel and the turbine.

Turbine

The turbine is connected to the input shaft of the automatic transmission. It also consists of curved blades with an angled edge. The turbine blades are designed to change the direction of the fluid entering the pump wheel. When the turbine rotates, the input shaft of the transmission also rotates, causing the vehicle to move.

The turbine also has a lock-up clutch, which engages when the converter reaches the connection point between the engine and the transmission (at this point, the converter is considered a coupling). This helps improve the efficiency of the torque converter.

Figure 22: Hydraulic clutch

b. Working principle

The pump wheel plays the role of the first fan connected to the engine, while the turbine functions as the second fan. When the engine is in operation, the pump wheel rotates, and centrifugal force directs the fluid inside the torque converter toward the turbine.

As it comes into contact with the turbine blades, the turbine begins to spin. This action sets the transmission system in motion, causing the vehicle's wheels to move.

When the engine stops, the turbine also ceases to rotate, but the working wheel connected to the engine continues to move, preventing the engine from stalling.

There are three stages of operation for the hydraulic torque converter:

Vehicle at a standhill

While the vehicle is stationary, the engine continues to drive the pump wheel, but the turbine cannot operate. This occurs when the vehicle is at a standstill, and the driver keeps the brake pedal pressed to prevent the vehicle from moving. In this case, the torque amplification is at its maximum. When the driver releases the brake pedal and applies the accelerator, the pump wheel starts to move faster, driving the turbine into action, and the vehicle starts to move. During this stage, there is a significant difference between the speed of the pump wheel and the turbine.

Acceleration

During acceleration, the speed of the turbine continuously increases, but there is still a difference between the pump wheel and the turbine. As the speed of the turbine increases, the torque amplification decreases (less than in the stationary/turbine not rotating condition).

Lock-up point

In this case, the turbine speed is approximately 90% (usually at 60 km/h) compared to the pump wheel, and this point is called the lock-up point. The torque amplification gradually approaches 0, and the torque converter becomes a simple fluid coupling. At the lock-up point, the lock-up clutch engages the turbine with the pump wheel and the torque converter, causing the pump wheel and turbine to rotate at the same speed, and the stator also begins to rotate in the direction of the pump wheel and turbine.

c. Advantages and disadvantages

The advantage of a hydraulic clutch system is that it provides a smoother and more precise clutch engagement compared to mechanical linkage systems. It also reduces the effort required to operate the clutch pedal, making it more comfortable for the driver. Hydraulic clutches are commonly used in modern vehicles, especially in cars with automatic transmissions.

Due to its inability to vary torque, hydraulic clutches have limited applications in automobiles. They exhibit low efficiency in the working range with low transmission ratios. Additionally, their high sensitivity can negatively impact the combined performance with internal combustion engines.

2.5. Electromagnetic clutch

2.5.1. Composition and working principle

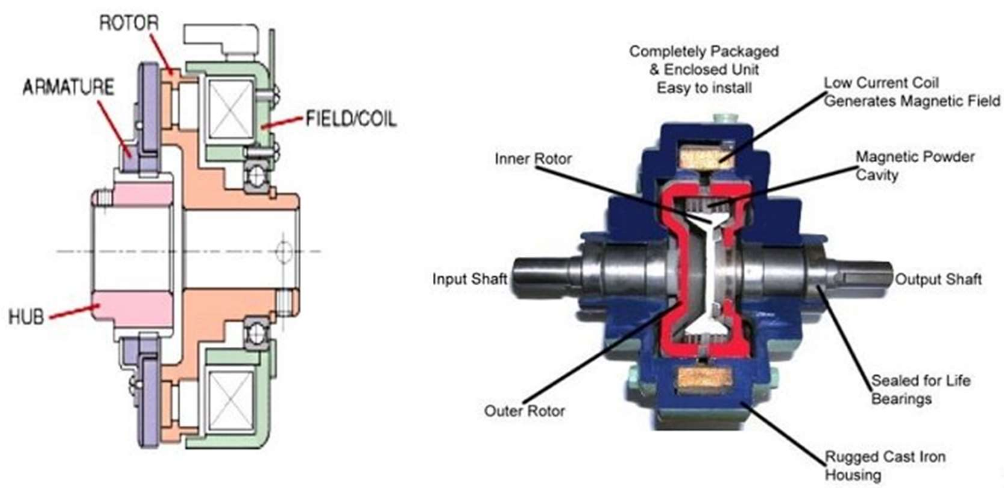


Figure 23: Electromagnetic Friction Clutch

An electromagnetic clutch is a type of clutch that uses electromagnetic force to engage and disengage the clutch. It consists of two main components: the driving part and the driven part.

The driving part contains an electromagnet that, when energized, creates a magnetic field. The driven part has a ferromagnetic disc or plate that is attracted to the magnetic field when the electromagnet is activated.

There are 3 types of electromagnetic clutch:

- *Electromagnetic Friction Clutch* uses frictional forces to engage and disengage the clutch. When the electromagnet is energized, it creates a magnetic field that attracts and compresses friction surfaces together, generating frictional forces for torque transmission. When the electromagnet is de-energized, the clutch disengages and the friction surfaces separate.

- *Electromagnetic Toothed Clutch* utilizes a toothed design on the driving and driven components. When the electromagnet is activated, it creates a magnetic field that attracts and aligns the teeth, allowing torque transmission. Disengagement occurs when the electromagnet is de-energized, and the teeth disengage, separating the components.

- *Electromagnetic Delayed-Action Clutch* has a built-in time delay before complete clutch engagement or disengagement. When the electromagnet is energized, it gradually builds up magnetic force, leading to a delayed engagement. Similarly, when the electromagnet is de-energized, there is a delayed release of magnetic force, resulting in a gradual disengagement.

3. Clutch control drivetrain options

The clutch drivetrain system in an automobile is crucial for power transmission and vehicle operation. It engages or disengages the engine from the transmission, serving multiple functions such as gear shifting, starting and stopping the vehicle, and protecting the engine and drivetrain.

There are two main types of clutch drivetrain systems: mechanical and hydraulic. Mechanical clutches offer high reliability, torque transmission, and ease of repair. However, they require manual operation and lack the convenience and efficiency of automatic transmissions, making them less suitable for stop-and-go traffic. Hydraulic clutches, on the other hand, are widely used in passenger and commercial vehicles due to their compactness, strong pedal force, integration ease, quick response, and fast operation.

In order to reduce driver pedal force, the clutch drive system can incorporate a mechanical, hydraulic, pneumatic, or vacuum power assist component. The most commonly used in vehicles, especially passenger cars, is the hydraulic drive system combined with power assistance. Passenger cars typically utilize vacuum power assist, while commercial vehicles often rely on pneumatic power assist due to the availability of compressed air tanks.

There are some clutch control drivetrain options:

- Mechanical drivetrain.
- Pneumatic-assisted mechanical drivetrain.
- Mechanical drivetrain with vacuum assistance.
- Hydraulic drivetrain.
- Hydraulic drivetrain with pneumatic assistance.
- Hydraulic drivetrain with vacuum assistance.

3.1. Mechanical drivetrain

a. Structure

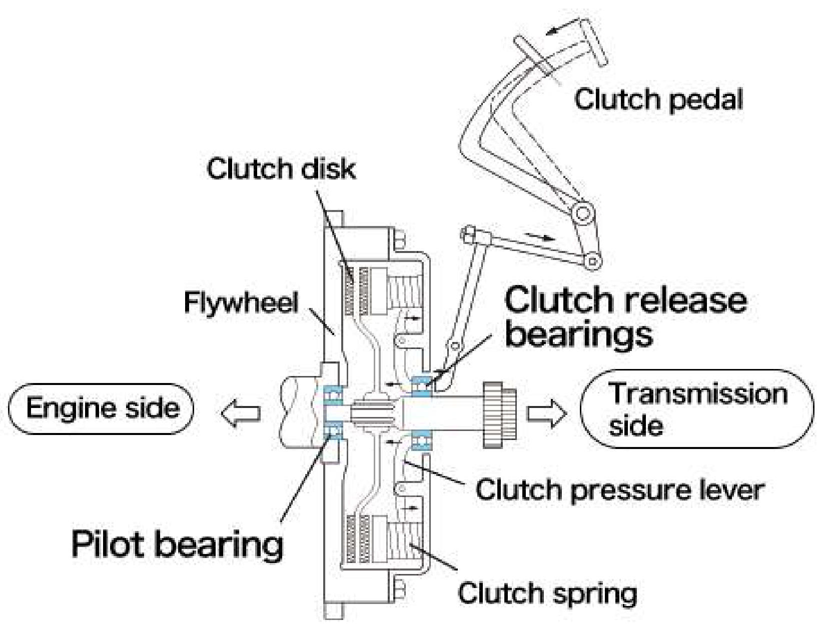


Figure 24: Mechanical drivetrain diagram

b. Working principle

Whenever the clutch is disengaged, the pressure plate and driven plate are separated from the working surfaces by the T-ball , which is moved to the left by the force on the pedal and transferred by puller and open pushing.

When the clutch is engaged, the pedal return spring pulls the pedal back to its starting position as the driver stops exerting effort on it. The torsion spring moves the ball to the right and stops pressing against the release lever at the same time. The spring pressure then re-engages the driven plate and pressure plate, restoring them to their initial operational condition.

c. Advantages and disadvantages

About the advantages of mechanical drivetrain, it has a simple design which leads to easy manufacturing and maintenance, as well as repair. Moreover, the engaging operation is swift and forceful which provides good driving experience. The price of the mechanical drivetrain is affordable and cost-effective.

Turning to its disadvantages, the disengaging phase is not swift and smooth. The friction force between components is quite high, which makes the pedal heavy and hard to press. However, this can be solved by using pressure assistance.

3.2. Pneumatic-assisted mechanical drivetrain

a. Structure

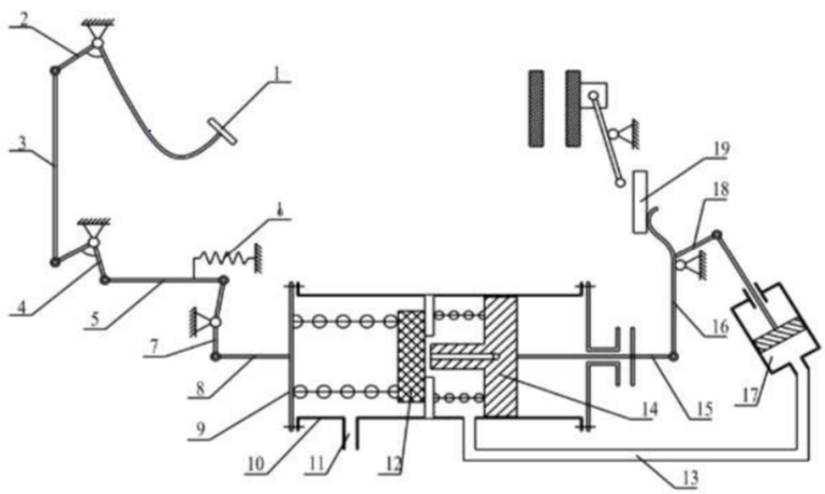


Figure 25:Diagram of Pneumatic-assisted mechanical drivetrain

1–Clutch pedal;	*12-Seal of the distribution valve;*
2,4,7,8,18–Drive lever;	*13-Air line;*
3,5–Linkage;	*14-Piston of the distribution valve;*
6–Torsion spring;	*15-Piston rod;*
9 -Flange of the distribution cylinder	*16-Clutch release fork;*
10–Distribution valve body;	*17-Actuator cylinder;*
11-Air inlet;	*19-Release bearing*

b. Working principle

The motorized link (2) rotates around O1 when the driver applies force Q to the clutch pedal (1). The driven link (4) rotates around O2 when the driver applies force Q to the connecting rod (3), and the driven link (7) rotates around O3 when the driver delivers force Q to the connecting rod (5). The distribution valve body 10 is being pushed to the right (in the direction of the arrow) by the driven link (8) and the distribution valve cam (9). The release work's bearing (16) rotates around O4 when the right side of the distribution valve body makes contact with the piston rod's stroke-limiting screw (15), pushing the release fork (19) to the left (in the arrowhead direction). Clutch disengagement. The distribution valve piston's end (14) meets the valve seal (12) and opens the valve at the same moment that the right side of the distribution valve body makes contact with the piston rod's stroke-limiting screw (15). At this point, compressed air pushes the cylinder (17) as it strokes through the compressed air line (13) and through the valve (12) into chamber B, causing the driving link (18) to revolve around O4. In addition, as the release bearing (16) rotates, the release fork (19) is pushed to the left. Clutch disengagement.

The clutch pedal (1) returns to its initial position when the driver releases it assisted by the torsion spring (6). The distribution valve body (10) is simultaneously pulled to the left by the driven link (8), and when the right end of the piston (14) makes contact with the valve body's right flange, the piston (14) is pushed to the left, spinning the release bearing (16) and moving the release fork (19) to the right. The distribution valve (12)'s valve seal is simultaneously pushed under the force of the return spring, completely sealing the valve. The clutch is now fully engaged at this stage.

c. Advantages and disadvantages

About the advantages, the mechanical drive system can still control the clutch in case the pneumatic actuation is broken, making the drive system dependable and reliable.

Turning to its disadvantages, when the pneumatic actuation is broken, it makes the pedal force more heavy. Therefore, this type of drivetrain is suitable for vehicles equipped with an air compressor.

3.3. Hydraulic drivetrain

a. Structure

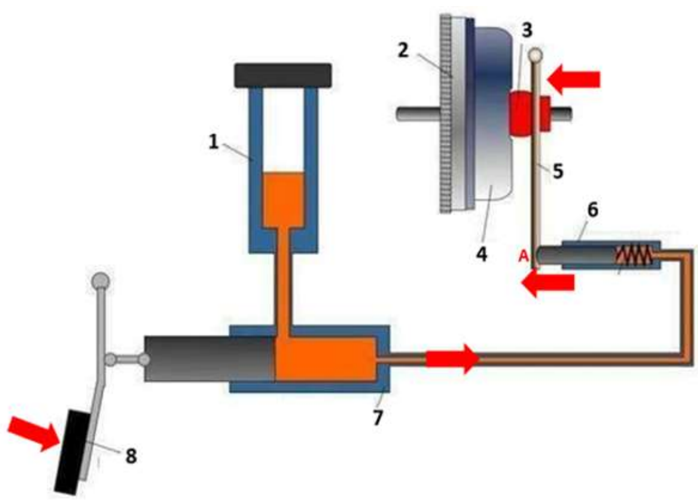


Figure 26: Diagram of Hydraulic drivetrain

1 – Oil tank	*5 – Clutch release lever*
2 – Flywheel	*6 – Slave cylinder*
3 – T-ball	*7 – Master cylinder*
4 – Pressure plate	*8 – Pedal*

b. Working principle

When engaging the clutch, the clutch disc attached to the clutch pedal begins to revolve when the driver depresses the clutch pedal to start the action. The pressure plate and the flywheel are in contact with the friction surfaces of the clutch disc. At that time, the pressure plate presses against the clutch disc and also makes contact with the diaphragm spring. The hydraulic clutch is then ready for use when the pressure plate, diaphragm spring, friction surfaces, clutch disc, and flywheel have all been assembled.

When disengaging the clutch, the driver presses the clutch pedal, the torsion spring rotates backward to disengage the clutch disc's friction with the pressure plate and the flywheel. The clutch disc's rotational motion now steadily decreases until it stops altogether.

c. Advantages and disadvantages

About the advantages of the hydraulic drivetrain, it is easy to arrange and has a simple design. Moreover, it possesses smooth operation and require minimal pedal pressure to disengage the clutch.

Turning to the disadvantages, the maintenance and repairs sections are difficult because the hydraulic system and cylinders need to be sealed to avoid air infiltration.

3.4. Vacuum-assisted hydraulic drivetrain

a. Structure

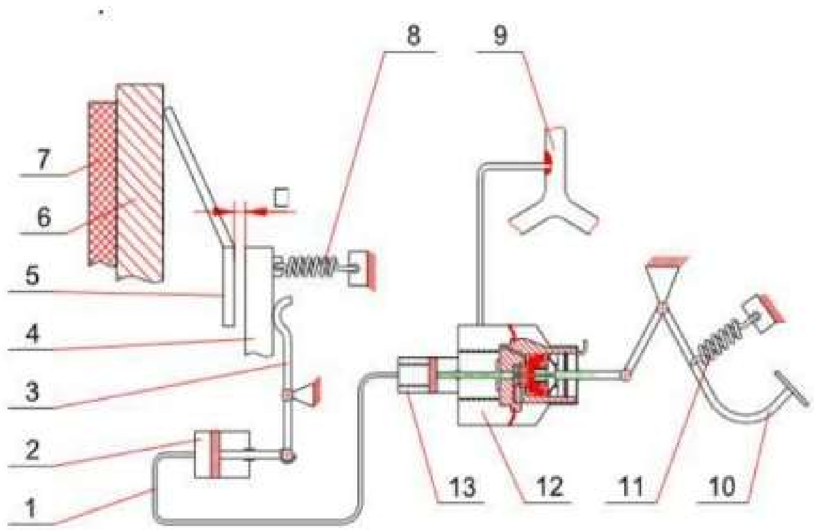


Figure 28: The diagram of Vacuum-assisted hydraulic drivetrain

1 – Oil line	*7 – Passive disc*
2 – Slave cylinder	*8 – Torsion spring of T-ball*
3 – Release fork	*10 – Clutch pedal*
4 – Thrust bearing	*11 – Torsion sping of clutch pedal*
5 – Throw-out bearing	*12 – Hydraulic clutch booster*
6 – Pressure plate	*13 – Master cylinder*

b. Principle Operation

When pushing the pedal, it is crucial to move the air push valve to the left by controlling it. The air push valve is also moved to the left until it comes into touch with the vacuum valve by the control valve's spring. The vacuum chamber and the air chamber are then separated, leaving one chamber. The piston moves to the left as a result of the pressure difference between the constant pressure and variable pressure chambers, pushing the reaction plate and the booster to the left and increasing the applied force.

When the pedal is released, the spring's restoring force will push the pusher to the left and lock the valves at the same time, completing the functioning procedure.

c. Advantages and disadvantages

About the advantages of Vacuum-assisted hydraulic drivetrain, it is straightforward to control and only requires a minimalpedal pressure. Moreover, it doesn't need fuel or power for the vacuum booster. The clutch can continue to work even if the power assistance system malfunctions.

Turning to its disadvantages, the manufacturing, maintenance, and repair sections are difficult because of the complicated design. It also requires high standards for sealing to stop gas and oil leaks. Because the vacuum level is not high, by expanding the diaphragm will make the construction bulky in order to achieve sufficient power assistance.

4. Compression spring design options

When designing a compression spring for a clutch system, several important factors need to be considered. Clutches commonly utilize compression springs to facilitate the engagement and disengagement of the transmission. During engagement, the driver does not apply pressure to the pedal. Instead, a compression spring exerts force to push the pressure plate against the flywheel, thereby pressing the friction disc against it. Through friction, the torque generated by the engine is transferred from the flywheel and pressure plate to the passive friction plates, then to the clutch hub via splines, and ultimately to the clutch shaft. In this manner, the clutch efficiently transmits engine torque to the power transmission system.

In the free state, the deformation characteristics (the relationship between force F and deformation Δ) of types of cylindrical, torsion, and disc springs are depicted on the graph:

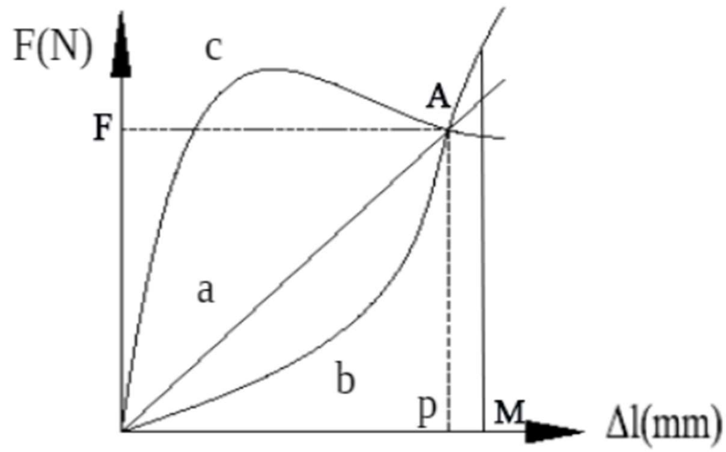


Figure 29: The characteristics of different types of compression spring

a. Cylindrical spring — *F: Compression force (N)*

b. Torsion spring — Δ: Deformation of springs

c. Disc spring

4.1. Torsion spring

A torsion sprin is a type of spring specifically designed to exert torque or rotational force when twisted.

Unlike traditional springs that generate linear force, the primary function of a torsion spring is to store rotational mechanical energy as it is twisted, and upon release, it exerts a restoring torque, bringing the spring back to its original position. The amount of torque exerted is directly proportional to the angle of twist, acting in the opposite direction.

a. The advantages

- The force applied to the large spring is often used in vehicles with a torque greater than 500 Nm.
- It is possible to reduce the space of the structure because the spring can compress until it lies on a flat surface.

b. The disadvantages

- The space near the clutch axis is tight and challenging to arrange for the clutch release bearing.

- When using a conical spring, the pressure from the spring acts on the pressure plate through pressure points. Therefore, adjusting the clutch becomes complex.

- Conical springs have a linear form in the small working range, and then, as the spring coils begin to engage, the stiffness of the spring increases very quickly. Therefore, it requires generating a significant force to disengage the clutch, and when the friction disc wears, the spring pressure decreases rapidly.

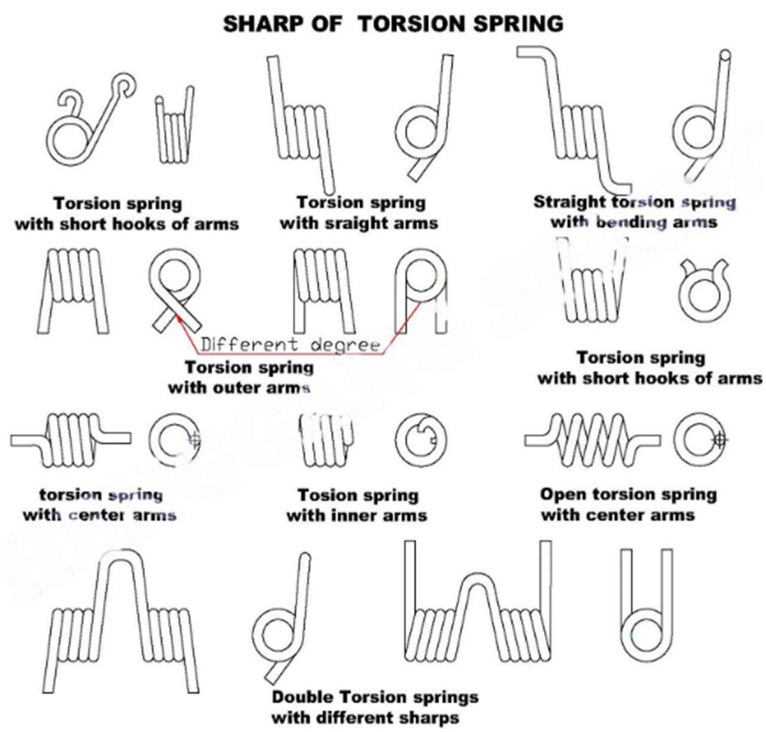


Figure 30: Types of Torsion spring

4.2. Cylindrical spring

Figure 31 Cylindrical spring

Cylindrical springs are often arranged in a circular pattern on the pressure plate. To position the springs and reduce their deformation under the centrifugal force, cups or raised bosses are typically used on the pressure plate or on the clutch housing.

a. The advantages

As there is a lot of pressure pressing against the friction disc, the structure is small and takes up little room. By positioning the springs symmetrically in relation to one another and the release levers, you may provide even pressure on the friction surfaces. Moreover, the cylindrical spring also maintains linear properties over the whole operating range with simple and inexpensive manufacture.

b. The disadvantages

Since the forces applied by the springs may change after some time of operation, they frequently cannot guarantee the same characteristics. Since unequal pressure might cause friction disc wear, it is crucial that the springs are manufactured precisely.

Non-uniformity and susceptibility to warping.

4.3. Disc spring

A disc spring is a conical-shaped spring that can be used to apply a predictable and precise force. It can also withstand high loads and operate in small spaces

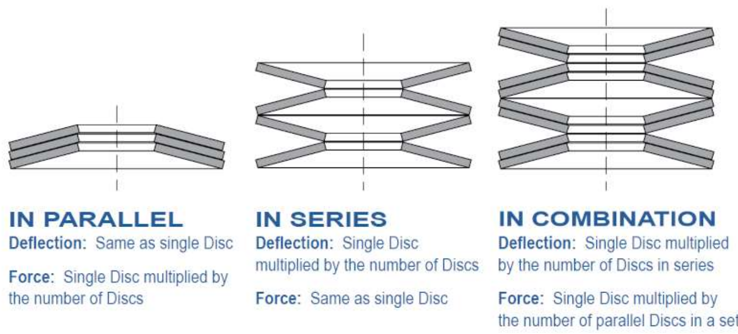


Figure 31: Types of Disc spring

a. The advantages

The release bearing is replaced with disc springs, creating a small and straightforward structure.

Disc springs have the benefit of operating reasonably within their operating range and experiencing little characteristics changes with deformation. As a result, the disengagement force needed to release the friction disc is not great, and the pressure holds steady even after the friction disc wears out.

b. The disadvantages

The disc spring has difficult manufacturing process.

5. Passive disc of the clutch

The passive disc, also known as the driven disc or driven plate, is a component of a clutch assembly used in vehicles with manual transmissions. It transmits power from the engine to the transmission, allowing for smooth engagement and disengagement of the clutch.

Clutch disc NKK NW-7829AF on Ebay

5.1. Structure of clutch passive disc

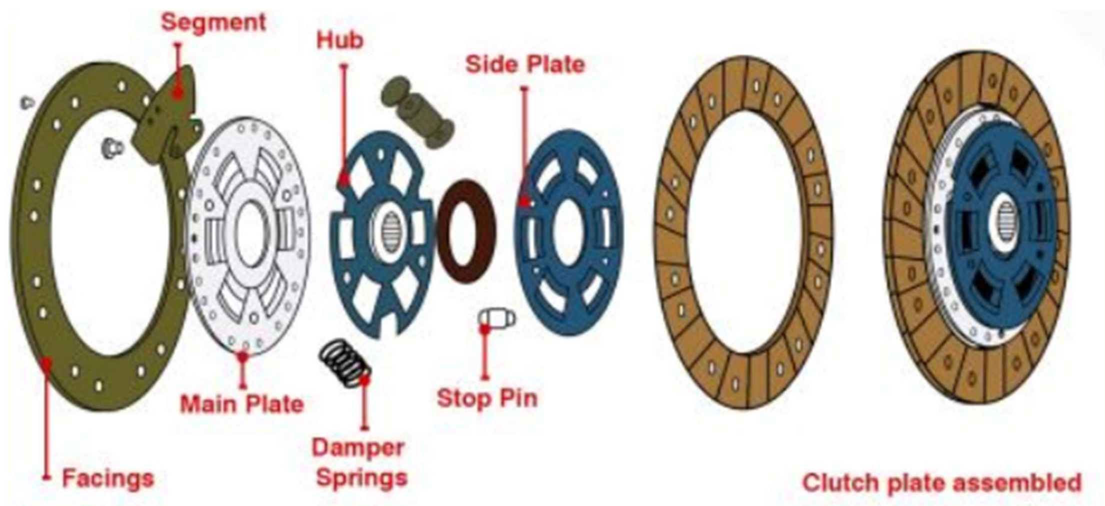


Figure 32: Structure of a clutch passive disc

The structure of a passive disc typically includes the following components:

a. Friction Facing

The passive disc has a friction-facing surface that comes into contact with the clutch pressure plate. This friction-facing material is usually made of a high-friction material such as organic, ceramic, or metallic material. It provides the necessary friction to transfer torque from the engine to the transmission.

b. Main Plate

This part will be attached to the friction facing. It includes holes to connect the segments throughout the outer ring.

c. Disc Hub

The hub is the central part of the passive disc where it connects to the transmission input shaft. It is typically made of steel and provides the structural support for the friction-facing material.

d. Damper Springs

Passive discs may also feature springs and dampers to enhance their performance. Springs help to absorb shock and vibration during clutch engagement, providing smoother operation.

e. Splines

The hub of the passive disc often incorporates splines on its inner diameter. These splines engage with corresponding splines on the transmission input shaft, creating a positive connection that allows the disc to rotate with the shaft.

f. Stopping Pin

This is a small detail but plays a relatively important role. It helps the system to be fixed to perform the power transmission process.

g. Side Plate

This part is located at the outermost position of the clutch passive disc. This is the component that keeps the parts of the disc fixed, especially the springs.

5.2. Operation

The working principle of a passive disc in a clutch assembly involves its engagement and disengagement to transmit or interrupt power between the engine and the transmission.

a. Power Transmission

When the clutch pedal is released, the clutch assembly is engaged. The pressure plate, which is actuated by the clutch work, applies clamping force to the passive disc. The clamping force brings the passive disc into contact with the flywheel, which is connected to the engine's crankshaft. This allows the passive disc to rotate with the engine.

b. Frictional Contact

The passive disc has friction linings on both sides, which come into contact with the flywheel and the pressure plate. The frictional contact between the passive disc and the flywheel enables power transmission from the engine to the clutch assembly.

c. Torque Transfer

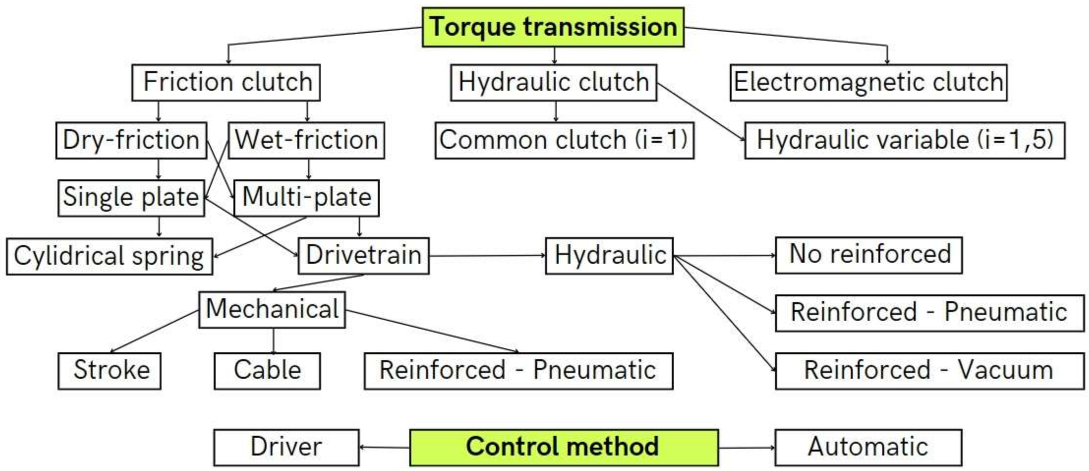
As the engine rotates, the torque generated by the engine is transferred through the passive disc. The friction linings provide the necessary grip to transmit the torque from the engine to the transmission input shaft, which is connected to the hub of the passive disc. This allows the rotational motion to be transferred to the transmission, enabling the vehicle to move.

d. Clutch Disengagement

When the clutch pedal is depressed, the clutch assembly is disengaged. The pressure plate releases the clamping force on the passive disc, separating it from the flywheel. This interrupts the frictional contact between the passive disc and the flywheel, thereby disconnecting the engine's power from the transmission.

6. Selection of design options

6.1. General analysis



a. Power and Torque Capacity

The clutch design must be capable of handling the power and torque output of the engine. The clutch components, such as the clutch disc, pressure plate, and flywheel, need to be appropriately sized and engineered to withstand the forces generated by the engine.

b. Vehicle Weight and Intended Use

The design of the clutch may vary depending on the weight of the vehicle and the intended use. A heavier vehicle or a vehicle intended for towing or off-road applications may require a clutch with higher torque capacity and durability.

c. Engagement Feel and Characteristics

Vehicle manufacturers aim to provide a smooth and predictable engagement feel for the driver. The design of the clutch components, such as the diaphragm springs in the pressure plate, can influence the engagement characteristics, including pedal effort, engagement point, and smoothness.

d. Longevity and Reliability

Clutch components should be designed for durability and longevity, with materials and construction that can withstand the demands of regular use. This includes selecting suitable friction materials for the clutch disc, ensuring proper heat dissipation, and minimizing wear on the components.

e. Cost and Manufacturing Considerations

The design of the clutch should also take into account manufacturing costs and feasibility. Manufacturers seek to balance performance and quality with cost-effective production methods and components.

In addition, the clutch system must meet the following criteria:

- Easy control with a light pedal effort.
- High and stable friction coefficient.
- Good heat dissipation.
- High operational efficiency.
- Simple structure for easy repair and maintenance.

6.2. Determine the option design

After carefully considering the general analysis with several design options of the clutch, advantages and disadvanges of each component, we may draw a conclusion to design a clutch with the features which can optimize performance and meet specific requirements and also easily adapt with the driving conditions that the drivers demand.

About the clutch assembly, we decide to choose the *Dry-friction Single-plate Clutch with a Diaphgram spring* by following those advantages:

- Most common type: as it is widely used in various vehicles, thus the components for replacements and repair are always available with a high quantity. This makes the repair and maintenance of the clutch easier and convenient

- Reducing size, weight, and simplifying the structure of the clutch compared to other types of the clutch.

- Without external assembly parts, balancing becomes easier.

- Additionally, centrifugal forces that reduce the pressure on the friction disc at high speeds are eliminated. The force acting on the disc remains consistent across all operating modes.

About the clutch control drivetrain system, we choose *the hydraulic drivetrain* by following these features:

- Advantages: Hydraulic drivetrain has high operational efficiency, easy installation. It has the ability to limit disc displacement when the clutch engages suddenly, reducing dynamic loads.

B. DESIGN AND CALCULATION FOR THE CLUTCH ASSEMBLY

1. Fundamental parameters of Toyota Fortuner 2021

Technical Characteristics	**Parameters**
Manufacterer Code	GUN156L-SNFSXN 3W
Engine	
Number of cylinders	4
Engine type	Cylinder in line
Fuel type	Diesel
Displacement (cc)	2755
Max power (kW/rpm)	150/3400
Max power HP/rpm	204/3400
Max torque Nm	420/1400-3400
Dimensions	
Ground clearance (mm)	279
Wheelbase (mm)	2745
Dimensions (Lxwxh) in mm	4795 x 1855 x 1835
Transmission	
Transmission	Part time manual 4x4
Gearbox	Manual
Rear differential	Manual locking
Weight/ Capacity	
Curb weight (kg)	2135
Gross vehicle weight (kg)	2735
Number of seats	7
Gear ratio	

Main transmission	4,53
First gear	4,12
Reverse gear	4,12
Tyres	
Tyre dimension	265/65 R17

Table 1: Technical characteristics of Toyota Fortuner 2021

2. Design calculation of the clutch assembly

2.1. Determine the friction torque

A clutch must be designed to transmit the full engine torque and, at the same time, protect the power transmission system from overloading. With these two requirements, *the frictional torque of the clutch is calculated using the formula:*

$$M_c = \beta . M_{emax} \qquad (1)$$

Note:

- M_c: the neccesary friction torque of the clutch
- M_{emax}: maximum engine torque (according to the technical characteristics, we have Memax = 420 N.m)
- β: the expected coefficient of the clutch

The coefficient β must be greater than 1 to ensure full transmission of the engine's torque in all cases (when the friction surface is covered with grease, when the pressure springs have reduced elasticity, reducing the frictional moment of the clutch, when the friction plates are worn). However, the coefficient β should not be chosen too large to avoid increasing the size of the passive disc and preventing the transmission system from being overloaded to ensure the function of the safety work. The coefficient β is chosen experimentally.

Vehicle type	β value
Passenger cars	1,35 1,75
Trucks, passengers, transport tractors (no mooc trailer)	1,50 2,25
Trucks with trailer	1,80 3,00
Usually Closed Clutch-Style NN Tractors	2,00 2,50

Table 2: The expected Coefficient of the clutch in different types of cars

Because Toyota Fortuner is a 7-seat passenger car so we choose β = 1.5.

Substitute into (1), we have the friction torque of the clutch needs to be transmitted:

$$M_c = \beta . M_{emax} = 1{,}5.420 = 630\ (N.m)$$

Calculate and check the clutch working conditions:

The parameters representing the load mode of the clutch are the specific sliding work ω_μ determined when the car starts on the spot and the temperature increase Δt of the active disc after one clutch engagement.

2.2. Determine fundamental dimensions and detailed designs

2.2.1. The way slipping work is generated during the clutch disengagement

When starting: The active clutch disc is rotating along the crankshaft while the passive disc has not rotated (there is a difference in speed). Before the passive disc rotates with the active disc into a solid block, there is always slippage. When shifting gears or when braking, the driver often opens the clutch and then closes it, the speed difference between the passive and active discs always causes slippage before they rotate into a continuous system. In short, any cause that causes a difference in speed between the two discs mentioned above will cause slippage.

The sliding phenomenon creates frictional work, frictional work turns into heat, heating the clutch parts beyond normal working temperature and especially the spring can be annealed at such high temperatures that it can Loss of pressing ability, causing rapid wear and tear of parts (friction discs, pressure discs...).

We calculate the slip work generated by slowly releasing the pedal. This calculation takes into account the actual process of closing the clutch, including two stages:

- Increase clutch torque M, from 0 value to Ma value at start close the clutch, the car will now start in place.

- Increase the torque of clutch M to the value that no more causes clutch slippage.

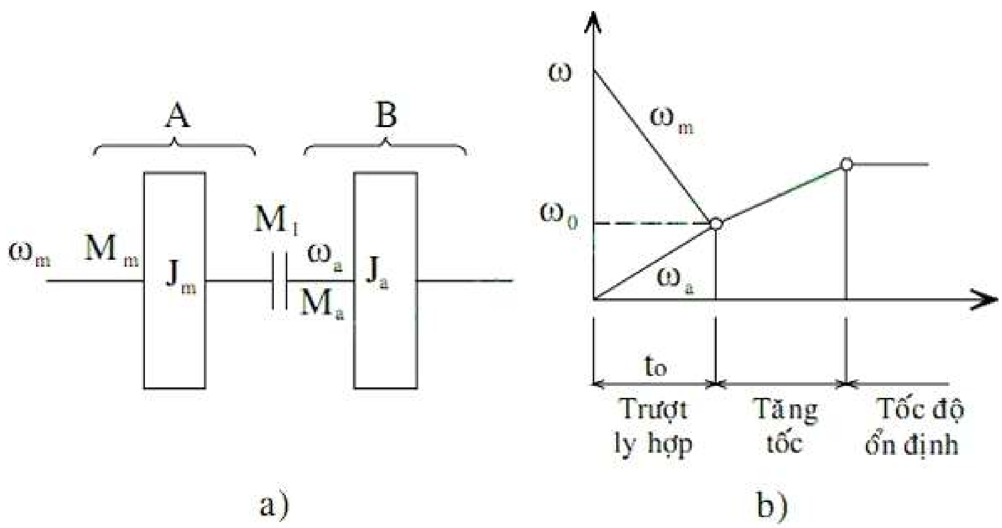


Figure 32: Diagram of Sliding calculation

a) Calculation model *b) Angular variation graph*

The sliding work in the initial stage L_1 is determined as follows:

$$L_1 = M_a.\frac{w_m - w_a}{2}.t_1 \quad (2)$$

The sliding work in the second stage L_2 is determined according to the following formula:

$$L_2 = \frac{1}{2}.J_a.(w_m - w_a)^2 + \frac{2}{3}.t_2 \quad (3)$$

The sliding work is determined according to the following formula:

$$L = L_1 + L_2 = M_a.(w_m - w_a).\left(\frac{t_1}{2} + \frac{2.t_2}{2}\right) + \frac{1}{2}.J_a.(w_m - w_a)^2 \ \text{(J)}$$

Note:

M_a is the motion resistance moment referred to the shaft of the clutch, calculated according to the formula:

$$M_a = [(G + G_m).\Psi + \mathrm{KF}v^2].\frac{r_w}{i_0.i_{h1}.\eta_{tl}} \quad (N.m)$$

Here:

- G: total weight of the car; G = M.g = 2735.9,81 = 26830,35 (N)

- α: slope angle, assumed α = 0

- Therefore: Ψ = f

- f: is the rolling resistance coefficient of the road. Choose f = 0,03 (according to Theory of Vehicle)

- Ψ = f = 0,03

- KFv^2: Air resistance, when starting on the spot V = 0

Therefore, KFv^2 = 0

- i_0: the transmission ratio of the main transmission, $i_0 = 4{,}53$

- i_h: the transmission ratio of the main gearbox, $i_h = i_{h1} = 4{,}12$

- η_{tl}: the efficiency of the powertrain, $\eta_{tl} = 0{,}93$

- r_w: the rolling radius of the wheel takes into account the deformation of the tire, $r_w = \lambda.\ r_w$ with $\lambda = 0{,}93$ is a coefficient that takes into account the deformation of the tire and the air pressure in the tire, taking tires with low pressure.

- B: Layer width: B = 265 mm

- d: wheel rim diameter; d = 17 inch

- r_0: the design radius.

We have, $r_0 = \lambda.\left(B + \frac{d}{2}.25{,}4\right) = 0{,}93.\left(265 + \frac{17}{2}.25{,}4\right) = 447{,}237$ (mm)

Therefore,

$$r_w = 0{,}93.447{,}237 = 416\ (mm) = 0{,}416\ (dm)$$

$$M_a = (G.\Psi + \mathrm{KF}v^2).\frac{r_w}{i_0.i_{h1}.\eta_{tl}}$$

$$= (26830{,}35.0{,}03 + 0).\frac{0{,}41}{4{,}53.4{,}12.0{,}93} = 19{,}01\ (N.m)$$

- w_m: angular speed of the engine referred to the maximum torque of the engine (rad/s).

$$w_m = \frac{\pi.n_M}{30} = \frac{3{,}14.3400}{30} = 355{,}87\ (rad/s)$$

- w_a: angular speed of the clutch shaft (rad/s). $w_a = 0$ as the car starts in place.

- J_a: The inertial moment of the car referred to the clutch shaft is determined according to the formula:

$$J_a = \frac{G}{g}.\frac{r_b^2}{(i_0.i_h)^2} = \frac{26830.(0{,}41)^2}{9{,}81.(4{,}53.4{,}12)^2} = 1{,}32\ (kg.m.s^2)$$

- t_1: the initial stage of engaging the clutch, determined by the formula:

$$t_1 = \frac{M_a}{K}$$

- t_2: the second stage of engaging the clutch, determined by the formula:

$$t_2 = \frac{A}{\sqrt{k}}$$

Note:

- k: the ratio coefficient of the increase in torque $M_1$when engaging the clutch. For passenger car, $k = 50 - 150\ N.m/s$.

We select $k = 150\ N.m/s$

- A: simplified expression calculated according to the formula:

$$A = \sqrt{2.J_a.(\omega_m - \omega_a)} = \sqrt{2.1,32.314} = 28,79$$

Therefore,

$$t_1 = \frac{M_a}{k} = \frac{19,01}{150} = 0,127\ (s)$$

$$t_2 = \frac{A}{\sqrt{k}} = \frac{28,79}{\sqrt{150}} = 2,35\ (s)$$

$$t_0 = t_1 + t_2 = 0,127 + 2,35 = 2,47\ (s)$$

To sum up, we have:

$M_a = 19,01\ N.m$	$J_a = 1,32\ kg.m.s^2$
$w_m = 314\ rad/s$	$t_1 = 0,127\ s$
$w_a = 0\ rad/s$	$t_2 = 2,35\ s$

Substitute into (2) and (3), we obtain:

$$L_1 = M_a.\frac{w_m - w_a}{2}.t_1 = 19,01.\frac{314 - 0}{2}.0,127 = 379,04\ (J)$$

$$L_2 = \frac{1}{2}.J_a.(w_m - w_a)^2 + \frac{2}{3}.t_2 = \frac{1}{2}.1,32.314^2 + \frac{2}{3}.2,35 = 65075\ (J)$$

So the total sliding work:

$$L = L_1 + L_2 = 379,04 + 65075 = 65454\ (J)$$

2.2.2. Determine the fundamental size of the clutch

Determine the average friction radius of the passive disc

Clutch friction moment is determined according to the formula:

$$M_c = \beta . M_{emax} = \mu . P_{\Sigma} . R_{avg} . i$$

Note:

- μ: friction coefficient

- P_{Σ}: total pressure on the friction discs (kg)

- i: number of pairs of friction surfaces

- R_{avg}: average friction radius (cm)

Outside diameter of friction plate according to empirical formula:

$$D_2 = 2. R_2 = 3{,}16. \sqrt{\frac{M_{emax}}{C}}$$

Here:

- M_{emax}: maximum torque of the engine (Nm)

- D_2: outside diameter of friction disc (cm)

- C: experience coefficient. With cars we choose C = 4,7

Therefore,

$$D_2 = 2. R_2 = 3{,}16. \sqrt{\frac{M_{emax}}{C}} = 3{,}16. \sqrt{\frac{420}{4{,}7}} = 29{,}8(cm) = 298\ (mm)$$

Compare the outer diameter of the friction disc with the diameter of the engine flywheel taken from the reference vehicle: $D_{fw} = 365\ mm$.

We see that $D_2 < D_{fw}$. Therefore, we choose $D_2 = 298$ mm.

→ Outer radius of friction disc: $R_2 = 149\ mm$.

The inner radius of the friction disc is calculated according to the outer radius:

$R_1 = (0{,}53 \div 0{,}75).R_2 = (0{,}53 \div 0{,}75).149 = (78{,}97 \div 111{,}75)$ mm

Choose $R_1 = 120\ mm$

Inferred: The average friction radius is calculated according to the formula:

$$R_{avg} = \frac{2}{3}.\frac{R_2^3 - R_1^3}{R_2^2 - R_1^2} = \frac{2}{3}.\frac{149^3 - 100^3}{149^2 - 100^2} = 126\ (mm) = 12{,}6\ (cm)$$

Check the pressure generated on the friction disc surface.

Choose $i = 2$, w have:

The number of pairs of friction surfaces calculated according to the formula:

$$i = \frac{M_c}{\mu.P_{\Sigma}.R_{tb}} = \frac{M_c}{2.\pi.R_{tb}^2.b.\mu.[q]}$$

Note:

- M_c: the friction moment of the clutch

- b: the width of the friction plate mounted on the passive disc

$$b = R_2 - R_1 = 149 - 100 = 49\ (mm) = 4{,}9\ (cm)$$

- $[q]$: Allowable specific pressure on friction surface

By looking up table 3 of the manual "Design of automobile clutch systems", we can determine the allowable specific pressure.: $[q]$ = 100 ÷250 kN/m^2

→ We choose $[q]$ = 250 $kN/m2$ = 2,5 $kg/cm2$

- μ: Friction coefficient, select μ = 0,3

⇒ *Check the friction surface pressure according to the formula:*

$$q = \frac{M_c}{2.\pi.R_{avg}^2.b.\mu.i} = \frac{64,24.100}{2.3,14.12,6^2.4,9.0,3.2} = 2,19\ (kg/cm^2)$$

So q < [q] = 2,5 (kG/cm2). The friction surface ensures allowable durability.

2.2.3. Check the specific sliding work

Slip work when the car starts in place:

This formula was built according to the experience of HAHM Institute:

$$L = \frac{5,6\ .\ G\ .\ M_{emax}.\left(\frac{n_o}{100}\right)^2.r_w^2}{i_0.i_h.i_f.(0.95.M_{emax}.i_c - G.r_w.\Psi)}$$

Note:

- L: clutch slippage when starting on the spot (kg.m)

- G: weight of entire car, $G = 2735$ kg

- M_{emax}: maximum torque of the engine, $M_{emax} = 42,8\ kg.m$

- n_0: number of engine revolutions when the car is used on site (rpm)

We take $n_0 = 0,75.n_{emax} = 0,75.3400 = 2550\ (rpm)$

With With n_{emax} is the maximum number of revolutions of the engine.

- r_{tire}: tire working radius, $r_{tire} = 427\ mm = 0,43\ m$

- λ: tire deformation coefficient, select $\lambda = 0,935$ (low pressure tires)

$$r_{tire} = \lambda.r_0 = 0,935.447,237 = 418,17(mm) = 0,418\ (m)$$

- i_0: Transmission ratio of main transmission, $i_0 = 4,53$

- i_h: Gear ratio of main gearbox, $i_h = i_{h1} = 4,12$

- i_f: Transmission ratio of the auxiliary gearbox, $i_f = 1$

- i_c: overall transmission ratio of the drive system, $i_c = i_0.i_h.i_f = 18,66$

- Ψ: Total resistance coefficient of the road surface, select Ψ = 0,16

So the sliding work of the clutch when starting on the spot is:

$$L = \frac{5{,}6.G.M_{emax}.\left(\frac{n_o}{100}\right)^2.r_{tire}^2}{i_0.i_h.i_f.(0.95.M_{emax}.i_c - G.r_{tire}.\Psi)} =$$

$$= \frac{5{,}6.2735\,.\,42{,}8.\left(\frac{2550}{100}\right)^2.0{,}418^2}{4{,}53.4{,}12.1.(0.95.42{,}8.18{,}66 - 2735.0{,}418.0{,}16)}$$

$$= 6930\ (kg.m)$$

Specific sliding work.

To evaluate the wear of the friction disc, the specific sliding work must be determined according to the formula:

$$l_o = \frac{L}{F.i} \leq [l_o]$$

Note:

- L: clutch slippage (kg.m)

- F: friction ring surface area, $F = \pi.(R_2 - R_1)^2\ (cm^2)$

- i: number of pairs of friction surfaces, we have $i = 2$

- $[i_0]$: specific sliding work allowed

Choose $[l_0]$= 1000 ÷1200 KJ/ m^2 = 10 ÷ 12 kg.m/cm^2

The allowable specific sliding force is determined by the type of car:

$$l_0 = \frac{L}{F.i} = \frac{L}{\pi.({R_2}^2 - {R_1}^2).i} = \frac{6930}{3{,}14.2.(14{,}9^2 - 10^2)}$$

$$= 10{,}65437\ (kg.m/cm^2) < [l_0]$$

Conclusion: The specific sliding work satisfies the allowable conditions.

2.2.4. Check the specific sliding work

To check the average temperature increase after each clutch engaging and disengaging when the vehicle starts in place. Ignoring the amount of heat transferred to the surrounding environment, it is considered that sliding work generates heat to heat parts such as springs, pressure plates, etc. Therefore, the temperature of parts must be checked by determining the temperature increase according to the formula:

$$\Delta T = \frac{\gamma . L}{C . m_t} = \frac{\gamma . L}{427 . c . G_t}$$

Note:

- L: slipping work is generated when the clutch slips (J)

- T: detailed temperature calculation ($°C$)

- γ: coefficient to determine the sliding work used to heat the part to be calculated with external pressure plate

- $\gamma = \frac{1}{2} . n$, with n is the number of passive disks

→ $\gamma = \frac{1}{2} . n = \frac{1}{2} . 1 = 0,5$

- C: specific heat capacity of the part. For cast iron C = 0,115 kcal/kG ^oC

- m_t: mass of heated parts. With Fortuner, $mt = 8kg$

- G_t: weight of heated components (kg)

We have:

$$\Delta T = \frac{\gamma . L}{C . m_t} = \frac{\gamma . L}{427 . c . G_t} = \frac{0,5.6930}{427.0,115.6} = 8,82\ °C$$

The allowable temperature increase of the calculation details for each vehicle start-up is less than 10°C. Therefore, it meets the allowable conditions.

2.2.5. Design the clutch drivetrain system

The clutch drivetrain system has the function of transmitting the driver's pedal force to the clutch to perform clutch engaging and disengaging.

Clutch drive is usually mechanical or hydraulic. Mechanical drives have the common advantage of simple structure. They are easy to manufacture, but they also have the disadvantage that the pedal force often has to be large and is difficult to arrange for cars with engines located far away from the driver. Mechanical drives are often used on some cars and trucks because cars require small pedal force and trucks often have compressed air tanks, so the arrangement of power steering is convenient and hydraulic drives are available today. Used on most types of cars and passenger cars due to its great advantages of being compact, creating great pedal force, being placed on the car and having a quick impact time.

To reduce the force the driver exerts on the pedals, the drive system may have mechanical, hydraulic, pneumatic or vacuum assist. Currently, the most commonly used in cars is hydraulic drive combined with power steering. Power assistance on passenger cars can be vacuum assistance, while trucks often use a pneumatic assistance system due to the availability of compressed air tanks.

The purpose of designing the clutch drivetrain system is to be easy to arrange, easy to control, ensure reliability and ensure economy. Therefore, the drive plan must meet the requirements of the drivetrain system mentioned above.

Commonly used drive options are:

- Mechanical drive.
- Pneumatically assisted mechanical drive
- Vacuum assisted mechanical drive.
- Hydraulic drive.

- Hydraulic drive with pneumatic assistance.

- Vacuum assisted hydraulic drive.

2.2.6. Mechanical drivetrain without reinforcement

The mechanical clutch drivetrain diagram without reinforcement is shown on the following diagram:

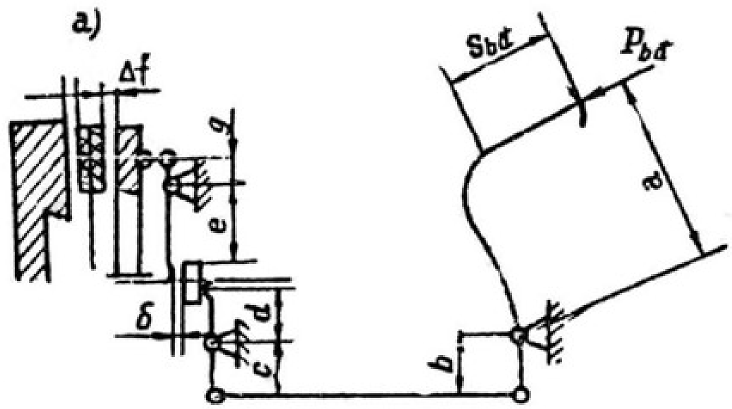


Figure 34: The diagram of mechanical drivetrain without reinforcement

For mechanical drives, the gear ratio of the drive system and pedal stroke are determined as follows:

$$S_{pe} = (\delta_m . Z_{ms} . \delta_{dis}) . i_{co} + \delta_0 . \frac{a}{b} . \frac{c}{d} . \frac{e}{f} + (\delta_1 . \delta_2) . \frac{a}{b}$$

Note:

- δ_m: clearance between each pair of friction surfaces when the clutch is opened (mm)

$$Z_{ms} = 2 \; and \; \delta_m = 0{,}75 \; mm$$

- δ_{dis}: the required displacement of the pressure plate is due to the elasticity of the passive plate (mm)

$$\delta_{dh} = 1 \; mm$$

- δ_0: the necessary clearance gap between the opening lever and the opening bush (mm)

$$\delta_0 = 2\ mm$$

- δ_1: the required clearance gap between pedal and drive system (mm)

$$\delta_1 = 1{,}5\ mm$$

- δ_2:distance to open the oil compensation hole in the master cylinder (mm).

$$\delta_2 = 0{,}5\ mm$$

- $\frac{a}{b}$: Pedal gear ratio, i_{pe}

- $\frac{c}{d}$: Intermediate transmission ratio, i_{inter}

$$i_{inter} = 1$$

- $\frac{e}{f}$: Transmission ratio of the opening fork, i_{of}

$$i_{of} = 2$$

- i_{co}: overall transmission ratio of the control system; equal to the product of the transmission ratios of the components involved in the control system

$$i_{co} = i_{pe}.i_{inter}.i_{of}.i_{ac}$$

With i_{ac}: transmission ratio of the actuator, choose $i_{ac} = 3{,}8$

The calculated stroke must be within the reach limit (leg extension) of the driver, select $[S_{pe}] = 150\ mm$

Subtitute into the formula above, we can calculate the transmission ratio of the pedal $S_{pe} \in [S_{pe}]$*:*

$$i_{pe} = \frac{[S_{pe}]}{[(\delta_m.Z_{ms} + \delta_{dis}).i_{inter}.i_{of}.i_{ac} + \delta_0.i_{inter}.i_{of} + (\delta_1.\delta_2)]}$$

$$= \frac{150}{[(0,75.2 + 1).1.1,2.3,8 + 2.1.2 + (1,5.0,5)]} = 9,3$$

2.2.7. Determine the force applied to The Clutch pedal

When applying a force to the diaphragm spring, initially, it is necessary to apply a force larger than the force applied to the cylindrical spring for the same profile, then when the deformation increases, the diaphragm spring ensures the adjusting force. Take control of the driver lightly.

Total pressure of the springs on the pressure plate when the clutch is not opened:

$$P_{\Sigma} = \frac{M_c}{\mu.i.R_{avg}} = \frac{630}{0,25.2.0,126} = 10000\ (N)$$

When the clutch is disengaged, the spring tends to be radial rather than axial, leading to an increase in the radial force of the spring by about 20% while the axial force decreases.

So the spring force when the clutch is disengaged is:

$$p' = 1,2.P_{\Sigma} = 1,2.10000 = 12000\ (N)$$

When the clutch is disengaged, the spring bracket moves to the right, at this time the tilt angle is 68°, causing the pressure force to tend to be radial rather than axial, leading to a smaller clutch opening force.

The clutch disengaging force is:

$$p_{cl} = p'_{\Sigma}.\cos(68°) = 12000.\cos(68°) = 4495,28\ (N)$$

The force required by the driver to apply to the pedal to disengage the clutch:

$$Q_{pe} = \frac{p_{cl}}{i_c.\eta_k} \le [Q_{pe}]$$

Here:

- Q_{pe}: The force the driver exerts on the pedals (N)

- η_k: performance of the drive mechanism, $\eta_k = 0{,}83$

- $[Q_{pe}]$: The pedal force required by the driver to open the clutch.

$$Q_{pe} = \frac{p_{cl}}{i_c.\eta_k} = \frac{4495{,}28}{18{,}66.0{,}83} = 290{,}24\ (N)$$

This pedal force is not within the allowable limit of the clutch pedal force of a passenger car with power assist $Q_{pe} < 150N$. Therefore, we need to design and calculate a reinforced vacuum actuator.

Pedal stroke S_{pe} is determined according to the formula: $S_{pe} = (\delta + l).i_c$

Note:

- δ: the gap between the open end of the blow and the balls T (mm)

- δ is within 2 ÷ 4 mm. Take $\delta = 3\ mm$.

- l: working stroke of pressure plate $l = 12\ mm$

So the pedal stroke is:

$$S_{pe} = (\delta + l).i_c = (3 + 12).18{,}66 = 81{,}24\ (mm)$$

With pedal travel allowed $S_{pe} \leq 180\ mm$, the pedal stroke is within allowable limits.

2.3. Calculate details of clutch and clutch drive

2.3.1. Rivets

Rivets for attaching the friction plates to the passive disc. Usually rivets are made of copper and aluminum in diameter 4 ÷ 6 mm.

Take $d = 5\ mm$.

To reduce the size of the clutch, when the clutch works in dry friction conditions, the material has high friction. The passive disc consists of friction plates and disc bone. Disc bone Usually made of medium and high carbon steel (50 and 80 steel). The material of the disc bone is Perado Bronze.

- Normal disc bone thickness: choose 2 mm.

- The thickness of the friction plate is usually chosen to be 4.5 mm.

The thickness of the friction plate and disc bone is selected according to table 1.4 of the book on design and calculation of cars - tractors by Dr. Nguyen Huu Huong, National University Publishing House, Ho Chi Minh City.

We arrange the rivets on the disc in 2 rows, corresponding to the inner radius R_1 and outer radius R_2.

Choose $R_1 = 115\ mm;\ R_2 = 140\ mm.$

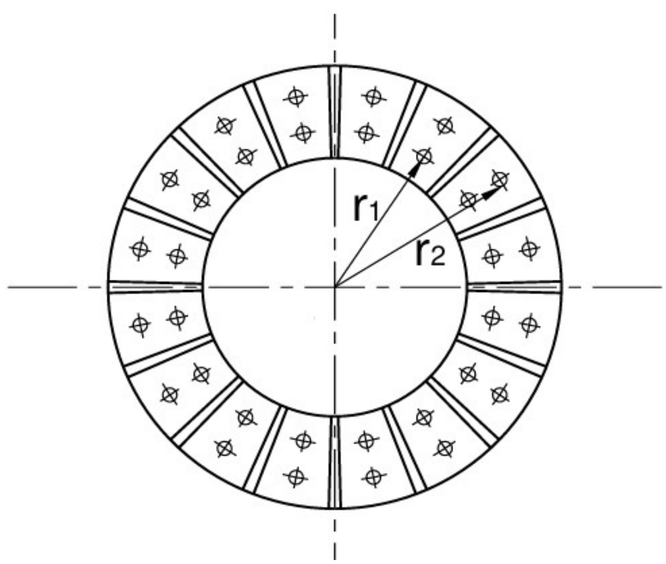


Figure 34: The arrangement of rivets on clutch disc

The force acting on each rivet is determined as follows:

$$F_1 = \frac{M_{emax}.R_1}{2(R_1^2 + R_2^2)} = \frac{420.0,115}{2(0,115^2 + 0,14^2)} = 735,72\ (N)$$

$$F_2 = \frac{M_{emax}.R_2}{2(R_1^2 + R_2^2)} = \frac{420.0{,}14}{2(0{,}115^2 + 0{,}14^2)} = 895{,}66\ (N)$$

Rivets are tested for shear and stamping stresses. When calculating F_1 and F_2 we take load mode isM_{emax} because in reality M_{emax} always smaller M_e. (M_e is the torque calculated from the upward path.)

Shear stress and crushing stress for rivets are calculated according to the formula:

- Shearing and crushing stress for rivets in the inner ring:

$$\tau_{c1} = \frac{F_1}{n_1.\frac{\pi.d^2}{4}} \le [\tau_c]\ (N/m^2); \quad \tau_{cd1} = \frac{F_1}{n_1.l.d} \le [\tau_{cd}]\ (N/m^2)$$

Note:

- τ_c: shear stress of rivets (N/m^2)

- τ_{cd}: rivet insertion stress (N/m^2)

- n_1: Number of rivets arranged on the inner (outer) ring

Take $n_1 = 8$; $n_2 = 10$.

- F: force acting on the inner ring rivet row (N)

- d: rivet diameter (m)

- l: stamping length of the rivet, about 1/2 the thickness of the friction plate, so we have: l = 4/2 = 2 (mm) = 0,002(m)

Check the rivets in the inner ring:

$$\tau_{c1} = \frac{F_1}{n_1.\frac{\pi.d^2}{4}} = \frac{735{,}72}{8.\frac{3{,}14.0{,}005^2}{4}} = 4{,}68.10^6\ (N/m^2) = 4{,}68\ (MN/m^2)$$

$$\tau_{cd1} = \frac{F_1}{n_1.l.d} = \frac{735{,}72}{8.0{,}002.0{,}005} = 9{,}19.10^6\ (N/m^2) = 9{,}19\ (MN/m^2)$$

- Allowable shear and stamping stresses for rivets in the outer ring.

Choosing $n_2 = 10$ similar to above we have:

$$\tau_{c2} = \frac{F_1}{n_2.\frac{\pi.d^2}{4}} = \frac{735{,}72}{10.\frac{3{,}14.0{,}005^2}{4}} = 3{,}75.10^6\ (N/m^2) = 3{,}75\ (MN/m^2)$$

$$\tau_{cd2} = \frac{F_1}{n_1.l.d} = \frac{735{,}72}{10.0{,}002.0{,}005} = 7{,}35.10^6\ (N/m^2) = 7{,}35\ (MN/m^2)$$

The allowable stress of the rivet material is:

$$[\tau_c] = 30\ (M\mathrm{N}/m^2)\ ; [\tau_{cd}] = 80\ (M\mathrm{N}/m^2)$$

So the rivets ensure the allowable durability.

2.3.2. Passive disc hubs

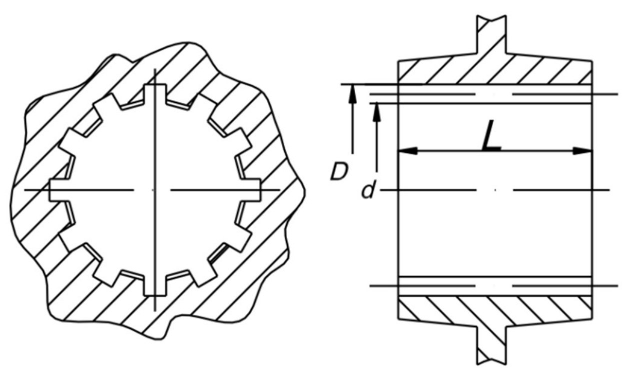


Figure 35: The cross section of passive disc hubs

The length of the passive disc hub is chosen to be relatively large to reduce the runout of the passive disc. The hub is coupled to the passive disc bone with rivets and attached to the clutch shaft with a spline.

The hub length is usually chosen equal to the outside diameter of the clutch shaft sprocket. When working conditions are heavy, choose $L = 1{,}4.D$

When working, the hub's key is subjected to crushing stress and shear stress which is determined by the formula:

$$\tau_c = \frac{4.M_{emax}}{Z_1.Z_2.L.b.(D+d)} \leq [\tau_c]$$

$$\sigma_{cd} = \frac{8.M_{emax}}{Z_1.Z_2.L.(D^2-d^2)} \leq [\sigma_{cd}]$$

Here:

-M_{emax}: maximum torque of the engine, $M_{emax} = 42{,}8\ kg.m$

- Z: Number of splines of a hub, $Z = 10$

- L: Hub length, $L = 3\ cm$

- D: Outside diameter of flower key, $D = 3{,}5\ cm$

- d: Inner diameter of flower key, $d = 2{,}8\ cm$

- b: the width of a flower key, $b = 4\ mm = 0{,}4\ cm$

$[\tau_c]$ and $[\sigma_{cd}]$ are the allowable shear and stamping stresses, respectively.

- $[\tau_c] = 10\ MN/m^2 = 100\ kg/cm^2$

- $[\tau_{cd}] = 20\ MN/m^2 = 200\ kg/cm^2$

$$\tau_c = \frac{4.M_{emax}}{Z^2.L.b.(D+d)} = \frac{4.42{,}8.100}{10^2.3.0{,}4.(3{,}5+2{,}8)} = 96{,}5\ kg/cm^2 < [\tau_c]$$

$$\sigma_{cd} = \frac{8.M_{emax}}{Z^2.L.b.(D^2-d^2)} = \frac{8.42{,}8.100}{10^2.3.0{,}4.(3{,}5^2-2{,}8^2)} = 110{,}35\ kg/cm^2 < [\sigma_{cd}]$$

So the hub ensures the allowable durability.

2.3.3. Clutch disc springs

Diaphragm spring calculation.

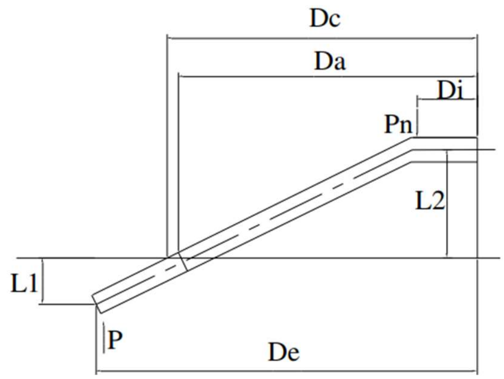


Figure 36: The diagram of clutch disc spring

Pressure needs to be generated to squeeze the pressure plate when closing the clutch:

$$P_{\Sigma} = \frac{M_I}{\mu . R_{avg} . i} = \frac{630}{0{,}25.0{,}126.2} = 10000\ (N)$$

Based on the reference vehicle and the requirements in selecting and designing the diaphragm spring, we choose the following basic dimensions:

- D_e: Diaphragm spring outer diameter $D_e = 210\ mm$

- D_i: Inner diameter $D_i = 70\ mm$

- Diaphragm spring thickness $\delta = 2{,}5\ m$

- The number of bars is evenly distributed on the membrane $Z = 12$

Total force P_{Σ} is expressed through the number of structures as follows:

$$F_{\Sigma} = \frac{2}{3} . \frac{\pi . E}{1 - \mu_p^2} . \frac{\delta . l_1}{D_e^2} . \frac{L_n . \frac{1}{k_1}}{(1 - K_2)^2} . [\delta^2 + (h - l_1 . \frac{1 - K_1}{1 - K_2} . \left(h - \frac{l_1}{2} . \frac{1 - K_1}{1 - K_2}\right))]$$

Note:

$$K_1 = \frac{D_a}{D_e} = \frac{160}{210} = 0{,}76 \qquad (D_a = 160\ mm)$$

$$K_2 = \frac{D_c}{D_e} = \frac{166}{210} = 0{,}76 \qquad (D_c = 166\ mm)$$

- Modulus E = 2.10^5 (N/mm^2)

- Height h = δ.2,2 = 2,5.2,2= 5,5(m)

(The coefficient of 2.2 ensures a wide constant pressure ring and no spring overturning).

- The displacement of the disc at the point where the pressure is applied

$l_1 = 2{,}2\ mm$

- μ_p: coefficient $\mu_p = 0{,}26$

Substitute into the quation above, we have:

$$F_\Sigma = \frac{2}{3}.\frac{\pi.E}{1-\mu_p^2}.\frac{\delta.l_1}{D_e^2}.\frac{L_n.\frac{1}{k_1}}{(1-K_2)^2}.\left[\delta^2 + \left(h - l_1.\frac{1-K_1}{1-K_2}.\left(h - \frac{l_1}{2}.\frac{1-K_1}{1-K_2}\right)\right)\right]$$

$$= \frac{2}{3}.\frac{3{,}14.2.10^5}{1-0{,}26^2}.\frac{2{,}5.2{,}2}{210^2}.\frac{L_n.\frac{1}{0{,}76}}{(1-0{,}79)^2}.\left[2{,}5^2 + \left(5{,}5 - 2{,}2.\frac{1-0{,}76}{1-0{,}79}.\left(5{,}5 - \frac{2{,}2}{2}.\frac{1-0{,}76}{1-0{,}79}\right)\right)\right] = 8802\ (N)$$

By comparing, we see: $F_\Sigma < P_\Sigma = 10000\ (N)$. Greater pressure results in the system β decreases. We recalculate the coefficient β.

We have: $M_l = \beta.M_{emax} = \mu.F_\Sigma.i.R_{avg}$, so:

$$\beta = \frac{\mu.P_\Sigma.\mathrm{i}.R_{tb}}{M_{emax}} = \frac{0{,}25.10000.2.0{,}126}{420} = 1{,}5$$

This result is within the allowable range of β ($\beta = (1{,}3 \div 1{,}75)$

Therefore, the size of the spring meets the standard.

Disc springs are calculated by determining the stress at the most loaded point, which is the center of the elastic element between the open walls and the solid region of the cone.

Stress is calculated:

$$\sigma = \frac{2F_n D_a}{\sigma^2 . (D_i + D_a)} + \frac{E}{2(1 - \mu_p^2)} . \frac{(D - D_a)\alpha^2 + \delta_\alpha^2}{2D_a}$$

With $D = \frac{(D - D_a)}{ln\frac{D_e}{D_a}} = \frac{(210 - 160)}{ln\frac{210}{160}} = 183{,}86\ (mm)$

And $\alpha = \frac{2h}{(D - D_a)} = \frac{2.5{,}5}{(210 - 160)} = 0{,}22$

- δ_a performance of the diaphragm spring, δ_a= 4%.δ = 4%.2,5 = 0,1(mm)

- F_n force needed to disengage the clutch.

$$F_n = \frac{D_e - D_c}{D_c - D_i} . P_\Sigma = \frac{210 - 166}{166 - 70} . 10000 = 4583\ (N)$$

So $\sigma = \frac{2F_n D_a}{\sigma^2 . (D_i + D_a)} + \frac{E}{2(1 - \mu_p^2)} . \frac{(D - D_a)\alpha^2 + \delta_\alpha^2}{2D_a}$

$$= \frac{2.4583.0{,}16}{2{,}5^2 . (0{,}070 + 0{,}160)} + \frac{2.10^5}{2(1 - 0{,}26^2)} . \frac{(0{,}184 - 0{,}16). 0.22^2 + 0{,}1^2}{2.0{,}16}$$

$$= 7{,}4.10^8 (N/m^2)$$

The material used to make the diaphragm spring is limited stress 60T steel

$$[\sigma] = 14.10^8 (N/m^2)$$

So the diaphragm spring is durable enough.

2.3.4. Damper spring

The damper spring is placed on the passive disc to avoid high-frequency resonance of torsional vibrations due to changes in engine and transmission system torque, ensuring smooth transmission of torque from the passive disc to the clutch shaft hub.

The maximum torque of pressing the damper spring is determined by the formula:

$$M_{max} = \frac{G_b.\varphi.r_w}{i_0.i_{h1}.i_f}$$

Here:

- G_b: the car's grip weight on the active bridge.

$$G_b = 2735.9{,}81 = 26830{,}35\ (N)$$

- φ: adhesion coefficient of the road surface. Take φ = 0,8

- r_w: working radius of the wheel

- i_0: transmission ratio of the main transmission, $i_0 = 4{,}53$

- i_{h1}: transmission ratio of the transmission in the gear lever 1, $i_{h1} = 4{,}12$

- i_f: transmission ratio of the auxiliary gearbox, $i_f = 1$

Substitute into the equation above, we have:

$$M_{max} = \frac{G_b.\varphi.r_w}{i_0.i_{h1}.i_f} = \frac{26830{,}35.0{,}8.0{,}416}{4{,}53.4{,}12.1} = 478{,}42\ (kg.m)$$

The torque transmitted through the shock absorber is calculated as the sum of the torque of the damper spring forces and the friction moment:

$$M_g = M_{max} = M_{lx} + M_{ms} = P_{lxg}.R_{xl}.Z_{lx} + P_{ms}.R_{ms}.Z_{ms}$$

Note:

- M_{sp}: Torque is generated due to the force of the springs.

- M_f: Friction moment.

- P_{dp}: Pressure of a damper spring.

- R_{dp}: Damper spring placement radius.

Take R_{dp} = 50 mm = 0,05 m.

- Z_{dp}: number of damper springs.

Take Z_{dp} = 6

- P_f: force acting on the friction ring.

- R_f: The average radius sets the friction rings.

Choose $R_f = 30\ mm = 0{,}03\ m$

Z_f: number of friction rings. Select $Z_f = 2$

When no torque is transmitted, the support bar connecting the discs will have a gap λ_1, λ_2 to the sidewalls of the hub. According to the spring and damper diagram we have:

- λ_1: characteristic gap for limited spring deformation when transmitting torque from the motor.

- λ_2: the gap represents the limited deformation of the spring when transmitting traction torque from the wheel. Minimum stiffness of the damping spring (also known as the torque applied to the passive disc to rotate the disc 1° relative to the hub):

$$S = 17{,}4.R_{dp}^2.K.Z_{dp} = 17{,}4.0{,}05^2.1300.6 = 339{,}3\ (mm)$$

The spring placement windows of the hub have a length dimension of A which must be a little smaller than the free length of the spring. The spring is always in the initial state of tension with: A = (25 ÷ 27) mm. We choose A = 25 mm.

When transferring torque from the engine and from the wheel through the same damper springs, the windows in the hub and the passive disc have the same length. Damper springs with different hardnesses, the length of the hub window must be one segment smaller than the disc window: $a = A_1 - A$.

Usually, $a = (1{,}4 \div 1{,}6)\ mm$. Choose $a = 1{,}5\ mm$.

The side of the window is tilted at an angle (1 ÷ 1,5º). We choose 1,5º.

Diameter of the support bar, choose $d = (10 \div 12)\ mm$ and place it in hole size B.

We choose $d = 12\ mm$.

The size of hole B is determined according to the gap $\boldsymbol{\lambda}_1$, $\boldsymbol{\lambda}_2$. The values $\boldsymbol{\lambda}_1$, $\boldsymbol{\lambda}_2$ are selected in the range from (2.5 ÷ 4) mm. We choose: $\lambda_1 = \lambda_2 = 3{,}5\ mm$.

So the size of the hole for the title bar is:

$$B = d + \lambda_1 + \lambda_2 = 12 + 3{,}5 + 3{,}5 = 19\ (mm)$$

According to experiment, it is often taken that:

$$M_f = 25\%. M_{max} = 25\%. 478{,}42 = 119{,}605\ (N.m)$$

And $M_{sp} + M_f = M_{max}$, therefore we have:

$$M_{sp} = M_{max} - M_f = 478{,}42 - 119.605 = 358{,}815\ (N.m)$$

The material used to make the damper spring is 65Ѓ steel, with an allowable torsional stress of [τ] = (6500 ÷ 8000) kG/cm^2

→ Choose [τ] = 6500 kG/cm^2

We have the pressure acting on a damper spring as:

$$P_{max} = P_1 \leq \frac{[\tau]}{8.\pi.d'^3.D'.K} = \frac{6500}{8.3,14.0,4^3.1,6.1,3} = 1944\ (kg)$$

The number of working turns of damper spring:

$$n_0 = \frac{\lambda.G.D^4}{1,6.P_{lx}.D^3}$$

Note:

- G: Elastic modulus displacement G = 8.105 (Kg/cm^2)

- $\boldsymbol{\lambda}$: Deformation of the damper spring from the non-working position to the working position, $\lambda = 2 \div 4$ mm. Take $\lambda = 3$ mm $= 0,003$ m

- d: Spring wire diameter, select $d = 4\ mm = 0,004\ m$

- D: Average diameter of the spring ring, choose $D = 16\ mm = 0,016\ m$

Substitute into the equation above:

$$n_0 = \frac{\lambda.G.D^4}{1,6.P_{dp}.D^3} = \frac{3.8.10^5.0,4^4}{1,6.1944.1,6^3} = 4,8$$

Take $n_0 = 5$

The working length of the spring is calculated according to the formula:

$$l_1 = n_0.d = 5.4 = 20\ (mm)$$

The length of the spring in the free state:

$$l_2 = l_1 + \lambda + 0,5.d = 20 + 3 + 0,5.4 = 25\ (mm)$$

Springs are tested for torsional stress:

$$\tau = \frac{8.P_{dp}.D'.k}{\pi.d^3}$$

Note:

- P_{dp}: maximum force acting on a damper spring

- D': average diameter of spring ring, $D' = 0{,}016\ m$

- d': diameter of spring wire, $d' = 4\ mm$

- K: Stress concentration coefficient:

$$K = \frac{4C-1}{4C-4} + \frac{0{,}615}{C} = \frac{4.4-1}{4.4-4} + \frac{0{,}615}{4} = 1{,}40375$$

Substituting these parameters into the formula for calculating τ, we have:

$$\tau = \frac{8.P_{dp}.D'.k}{\pi.d^3} = \frac{8.1196{,}05.0{,}016.1{,}40375}{3{,}14.0{,}004^3} = 10{,}7.10^8\ \left(\frac{N}{m^2}\right)$$

$$\leq [\tau] = 14.10^8\ (N/m^2)$$

Conclusion: So the spring is durable enough.

The material used to make the damper spring is 65T steel with an allowable torsional stress of [τ] = [6500÷ 8000] kg.cm^2, take [τ] =6500 kg.cm^2

Therefore,

$$P_{max} = P_{dp} \leq \frac{[\tau]}{8.\pi.d'^3.D'.K} = \frac{6500}{8.3{,}14.0{,}4^3.1{,}6.1{,}3} = 1944\ (kg)$$

2.3.5. Clutch axle

Calculate the gear ratio in the first gear:

$$i_{h1} = 4{,}12$$

The transmission ratio of the main transmission is calculated

$$i_0 = 4{,}53$$

Calculate the distance A-axis of the gearbox shaft according to the following formula:

$$A = a.\sqrt[3]{M_{emax}}$$

Here:

- K_a: experience coefficient

- For passenger car gearbox: $a = 14{,}5 - 16$

- For truck transmissions using gasoline engines: $a = 17 - 19{,}5$

- For truck transmissions using diesel engines: $a = 20{,}5 - 21{,}5$

- For auxiliary gearboxes and distribution gearboxes: $a = 17 - 21{,}5$

Choose: $a = 15$

$$A = a.\sqrt[3]{M_{emax}} = 15.\sqrt[3]{420} = 112{,}3\ (mm)$$

Choosing the normal module for gears that always mesh:

Choosing the tooth module is also consistent for gears in the same gearbox to simplify manufacturing and repair technology. To reduce the weight of the gearbox with the same shaft spacing, the module should be increased and the ring width should be reduced.

The normal modulus of the tooth ring can be selected according to the standard or according to the table.

Maximum torque of the engine M_{emax}, N.m	Normal modulus of gearbox gears m_n, mm
50 – 100	2,25 - 2,50
100 - 200	2,75 - 3,00
200 - 400	3,00 - 3,75
400 - 800	3,75 - 4,50
800 – 1000	4,50 - 6,00

Table 3: The normal module m_n of the automobile transmission gears follows the engine module.

Based on the table above, we can roughly select the gear module based on the maximum torque of the motor here is $M_{emax} = 420\ N.m$ then take $m_n = 3{,}75$.

On the other hand, we can choose m_n based on this formula:

$$m_n = \frac{2.A.cos\beta}{Z_a.(i+1)}$$

Here:

- Z_a: the number of teeth of the active gear

- i: transmission ratio of the calculated gear pair

- Β: angle of inclination of teeth

In this case, the number of pinion gears Z_a in the always meshing gear pair is selected according to the condition of no peak cutting with the number of teeth being 17 or more, for non-aligned teeth it is 14 teeth or more for the teeth with repair.

$$m_n = \frac{2.A.cos\beta}{Z_a.(i+1)} \rightarrow Z_a = \frac{2.A.cos\beta}{m_n.(i_a+1)} = \frac{2.99{,}52.cos\beta}{3{,}75.(i_a+1)}$$

After knowing A, m_n and Z_a, we can determine the gear ratio of the gear pair that is always in mesh according to the formula:

$$i_a = \frac{2.A.cos\beta}{m_n Z_a} - 1$$

Note:

- i_a: The transmission ratio of the gear pair is always consistent, which is often valuable for today's automobile transmissions $i_a = 1{,}6 - 2{,}5$

- β_a – The tooth inclination angle of the gear pair always meshes; the angle is called the average angle:

+ For passenger cars: $\beta_a = 30 - 45°$

+ For transport cars: $\beta_a = 20 - 30°$

The number of teeth z'_a driven gear of the always meshing gear pair is determined according to:

$$z'_a = z_a.i_a$$

Choose:

$$\beta = 30°;\ Z_a = 24;\ m_n = 3{,}75$$

We have:

$$i_a = \frac{2.A.cos\beta}{m_n Z_a} - 1 = \frac{2.99{,}52.\cos(30)}{3{,}75.24} - 1 = 2{,}39$$

Which satisfies the condition of $i_a = 1{,}6 \div 2{,}5$

The number of teeth $Z'_a = Z_a.i_a = 24.2{,}39 = 57{,}36$. Choose $Z'_a = 57$

Accurate calculation of A-axis distance:

$$A = \frac{m_n.(Z_a + Z'_a)}{2.\cos(\beta)} = \frac{3{,}75.(24 + 57)}{2.\cos(30)} = 175{,}37\ (mm)$$

The clutch axle is also the cartridge number primary shaft. The front end of the cylinder rests on the ball in the flywheel, the rear end attaches the ball bearing to the gearbox housing. The flat end has a straight helical gear that always meshes with the intermediate gear of the transmission.

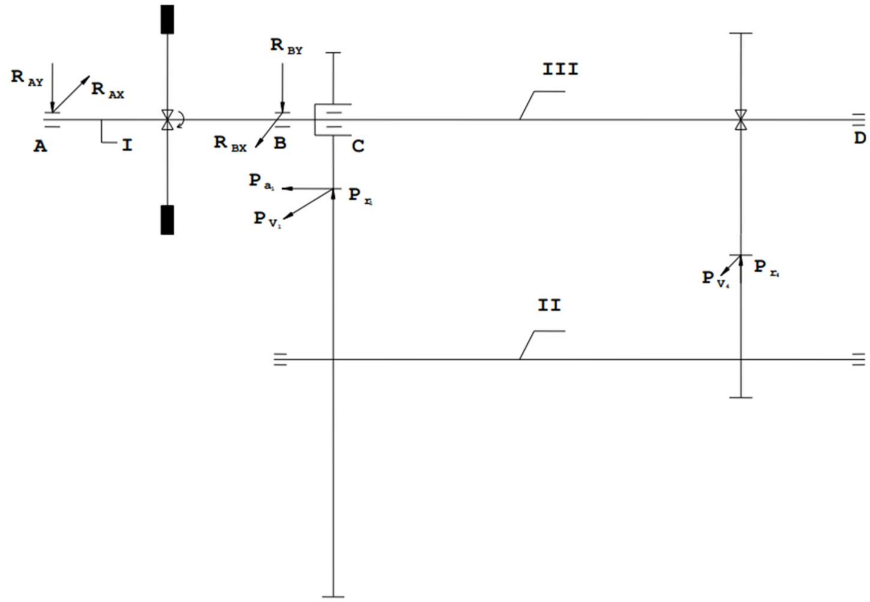


Figure 37: The diagram of forces acting on the clutch axle

Where:

Shaft I: Is the clutch shaft, and is also the primary shaft of the gearbox. The end of the shaft has a tilt gear assembly and has a wide hole in the center to install it in the needle ball supporting the head of shaft III.

Shaft II: Is the intermediate shaft of the gearbox, the two ends of the shaft are supported by two cylindrical ball bearings installed on the gearbox housing.

Shaft III: Is the secondary shaft of the gearbox, one end is resting on a needle ball bearing in the level comparison shaft, the other end is resting on a ball bearing installed on the gearbox body.

To test the shaft, we choose the maximum torque mode and put the gearbox in 1st gear.

Shaft I: $M_I = M_{emax}$ = 420 N.m

Shaft II: $M_{II} = M_I . i_a$ = 420.2,39 = 1003,8 N.m

Shaft III: $M_{III} = M_{II} . i_{h1}$= 1003,8.4,12= 4135,656 N.m

Calculation steps:

- Step 1: Calculate the forces on gear shaft I and shaft III

- Step 2: Determine the reaction force on the fish shaft supporting shaft I and shaft III.

- Step 3: Check the durability of gear shaft I (Clutch shaft).

Step 1: Calculate the forces on gear shaft I and shaft III

No.	Name	Symbol	Spur Gear	Helical gear
1	Circular force	P_v	$P_v = \frac{2.M_t}{Z.m}$	$P_v = \frac{2.M_t}{Z.m_s}$
2	Centripetal force	P_r	$P_r = P_v.tg\alpha$	$P_R = P_v.\frac{tg\alpha}{cos\beta}$
3	Axial force	P_a	$P_a = 0$	$P_a = P_v.tg\beta$

Table 4: Formulas to calculate force acting on gear pairs

Note:

- Z: Number of teeth of the gear being calculated

- M_t: Calculated torque on gearbox shafts (N.m)

- m_s: Face module (geometric parameter table of gear)

- α: Original profile angle (geometric parameter table of gear)

- β: Angle of tooth inclination (geometric parameter table of gear)

Shaft I

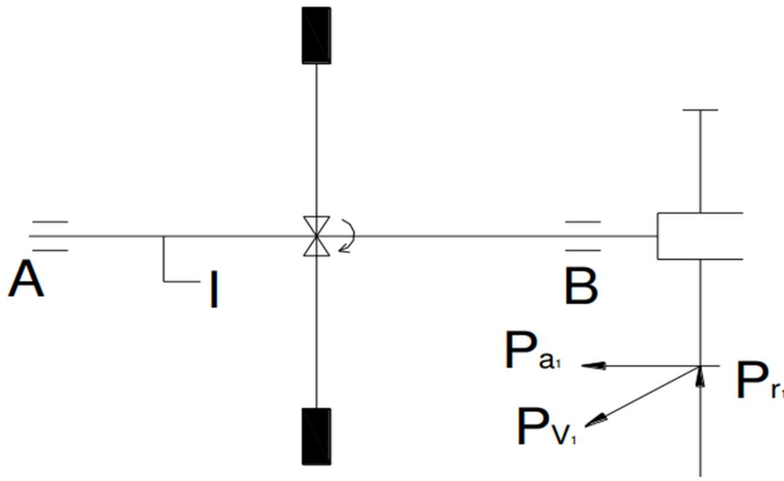


Figure 39: The diagram of forces acting on shaft I

The gear on shaft I is a helical gear with:

Front module

$$m_s = \frac{m_n}{cos\beta} = \frac{3{,}75}{\cos{(30)}} = 4{,}33\ (mm)$$

Diameter of dividing ring

$$d_1 = m_s.Z_a = 4{,}33.24 = 103{,}92\ (mm) = 0{,}10392\ (m)$$

$$d'_1 = m_s.Z'_a = 4{,}33.57 = 246{,}81\ (mm) = 0{,}24681\ (m)$$

- Angle of inclination of teeth: $\beta = 30°$

- Matching angle: $\alpha = 20°$

- Normal module: $m_n = 3{,}75$ mm

- Number of teeth: Z_a =24

Then,

Circular force:

$$P_{v1} = \frac{2M_1}{d_1} = \frac{2.420}{0,10392} = 8083,14\ (N)$$

Centripetal force:

$$P_{r1} = \frac{P_{V1}.tg\alpha}{cos\beta} = \frac{8083,14.tg(20)}{\cos(30)} = 3397,15\ (N)$$

Axial force:

$$P_{a1} = P_{v1}.tg\beta = 8083,14.tg(30) = 4666,8\ (N)$$

Shaft III:

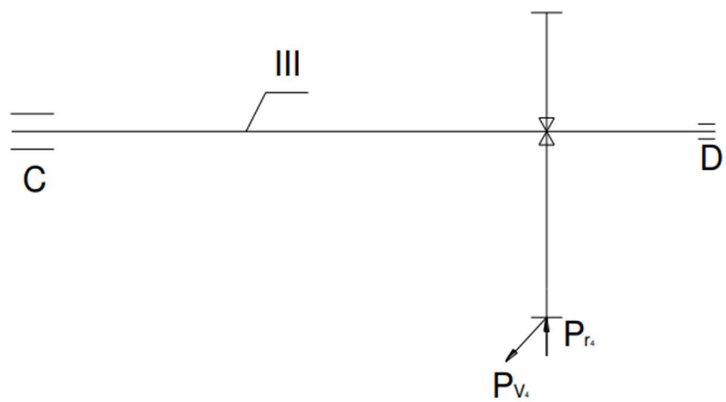


Figure 40: The diagram of forces acting on shaft III

The gear on shaft III is a straight gear with a rolling circle diameter:

$$d_4 = m.Z_4 = 3,75.53 = 198,75\ (mm) \approx 0,2\ (m)$$

- Matching angle: $\alpha = 20°$

- With: $Z_4 = 53$

- $m = 3,75$: is the normal module (mm)

Then,

Circular force:

$$P_{v4} = \frac{2.M_{III}}{d_4} = \frac{2.4135,656}{0,2} = 41356,56\ (N)$$

Centripetal force:

$$P_{r4} = P_{v4}.tg\alpha = 41356,56.tg(20°) = 15052,5\ (N)$$

Step 2: Determine the reaction force on the shaft of the bearings supporting shaft I and shaft III

Shaft III:

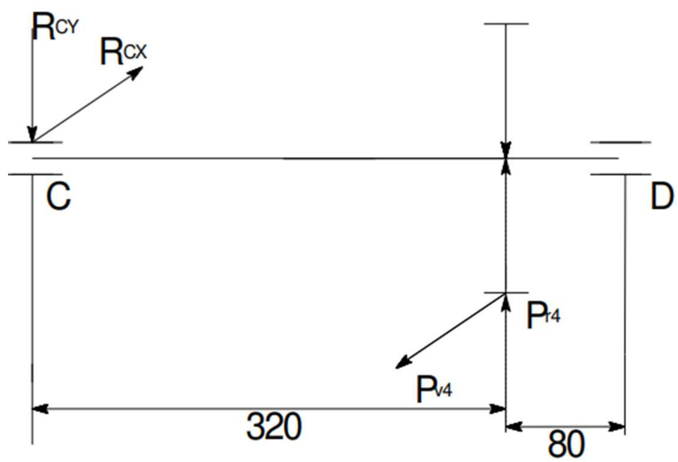


Figure 40: The diagram of reaction forces on the shaft of bearing supporting shaft III

We take:

$$\Sigma M_{Dx} = 0\ <=>\ R_{Cx}.(320 + 80) = P_{v4}.80$$

$$=> R_{Cx} = \frac{P_{v4}.80}{(320 + 80)} = \frac{41356,56.80}{320\ +\ 80} = 8271,312\ (N)$$

We take:

$$\Sigma M_{Dy} = 0 <=> R_{Cy} = \frac{P_{r4}.80}{320 + 80}$$

$$=> R_{Cy} = \frac{P_{r4}.80}{320 + 80} = \frac{15052,5\ .80}{320\ +\ 80} = 3010,5\ (N)$$

Shaft I:

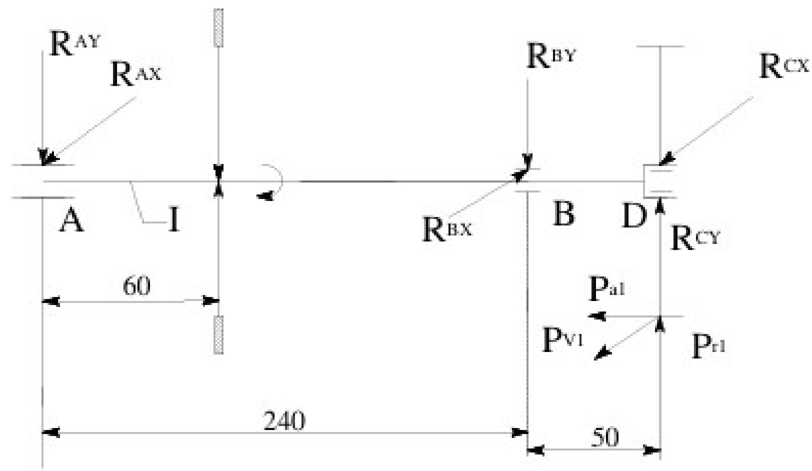


Figure 41: The diagram of reaction forces on the shaft of bearing supporting shaft I

We take:

$$\Sigma M_{Bx} = 0 \iff R_{Ax}.240 = (R_{Cx} + P_{V1}).50$$

$$\Rightarrow R_{Ax} = \frac{(R_{Cx} + P_{V1}).50}{240} = \frac{(8271{,}312 + 8083{,}14).50}{240} = 3407{,}17\ (N)$$

$$\Sigma M_{By} = 0 \iff R_{Ay}.240 - P_{a1}.\frac{128}{2} + (R_{Cy} + P_{R1}).50 = 0$$

$$\Rightarrow R_{Ay}.240 = P_{a1}.64 - (R_{Cy} + P_{R1}).50$$

$$\Rightarrow R_{Ay} = \frac{P_{a1}.64 - (R_{Cy} + P_{r1}).50}{240} = \frac{4666{,}8\ .64 - (3010{,}5 + 3397{,}15).50}{240} = 90{,}44\ (N)$$

So R_{ay} has the direction as in the diagram.

$$\Sigma R_y = 0 \iff P_{r1} + R_{Cy} - R_{Ay} = R_{By}$$

$$\Rightarrow R_{By} = P_{r1} + R_{Cy} - R_{Ay} = 3397{,}15 + 3010{,}5 - 90{,}44 = 6317{,}21\ (N)$$

$$\Sigma R_X = 0 \iff R_{Ax} + R_{Cx} - P_{v1} = R_{Bx}$$

$$\Rightarrow R_{Bx} = R_{Ax} + R_{Cx} - P_{v1} = 3407{,}17 + 8271{,}312 - 8083{,}14$$
$$= 3595{,}342\ (N)$$

Summarize the forces acting on the Shaft I:

$P_{v1} = 8083{,}14\ (N)$

$P_{r1} = 3397{,}15(N)$

$P_{a1} = 4666{,}8\ .64\ (N)$

$R_{Cx} = 8271{,}312\ \ (N)$

$R_{Cy} = 3010{,}5\ (N)$

$R_{Ax} = 3407{,}17\ (N)$

$R_{Ay} = 90{,}44\ \ (N)$

$R_{Bx} = 3595{,}342\ (N)$

$R_{By} = 6317{,}21\ (N)$

Step 3: Check the durability of gear shaft I (clutch shaft)

Determine bending moment M_{ux}

- At B: Consider from left to right:

$$M_B^x = 0{,}24.R_{Ay} = 0{,}24.90{,}44 = 21{,}7\ (N.m)$$

- At C: Consider from right to left:

$$M_c^x = -P_{a1}.\frac{d}{2} = -4666{,}8\ .64.\frac{0{,}128}{2} = 298{,}8\ (N.m)$$

Determine the bending moment M_{uy}

- Consider from left to right:

$$M_B^y = -0{,}24.R_{Ax} = -0{,}24.3407{,}17\ = 871{,}72\ (N.m)$$

- Consider from left to right: M_X = M_{emax} maximum torque of the engine

$$M_X = M_{emax} = 420\ N.m$$

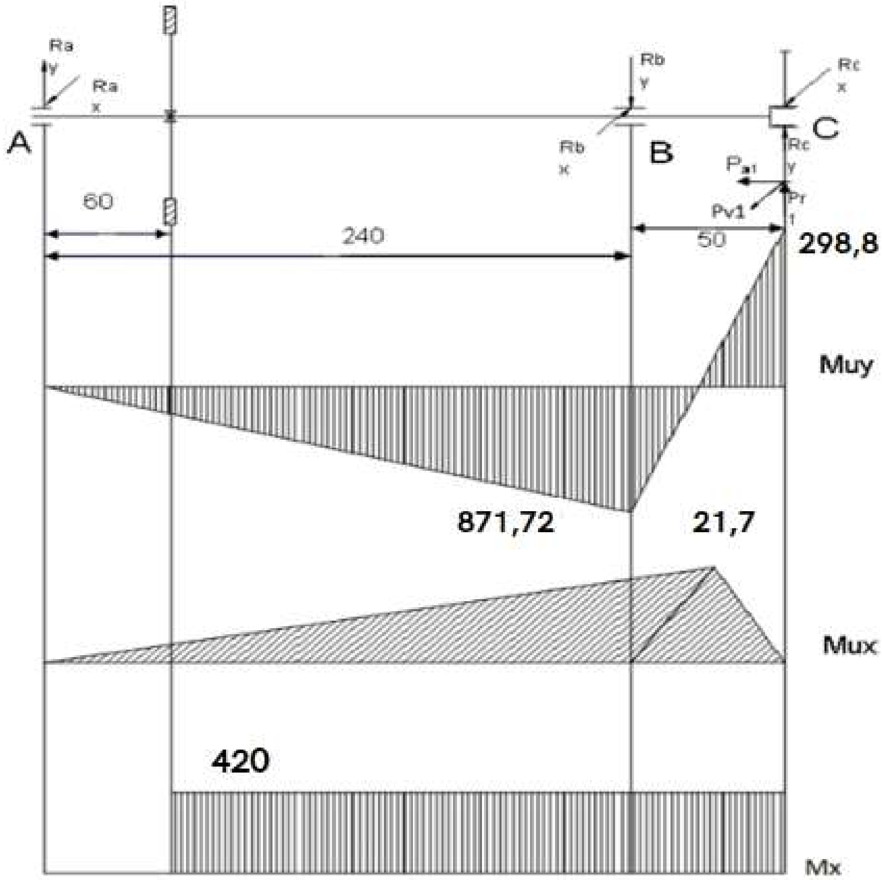


Figure 43: The diagram of torque acting on the clutch shaft

From the above moment diagram, we can determine a satisfactory uniform stability axis.

$$D \geq \sqrt[3]{\frac{M_{uxi}^2 + M_{uyi}^2 + M_{xi}^2}{0{,}1[\sigma_{th}]}}$$

(According to the theory of maximum tangential stress stability)

Here we already have an axis with cross-sections that are said to vary reasonably (uniformly). So we tested the durability at the most dangerous cross-section.

At B,

$|M_{ux}|$= 21,7 N.m

$|M_{uy}|$= 871,72 N.m

$|M_x|$= 420 N.m

The allowable stress value [σ] of the shaft manufacturing steel is selected according to table (7-2) of the book Machine Design - Nguyen Trong Hiep - Nguyen Van Lam.

With 40X steel we choose [σ] = 70 N/mm^2

$$M_u = \sqrt{M_{ux}^2 + M_{uy}^2} = \sqrt{21{,}7^2 + 871{,}72^2} = 872\ (N.m)$$

$$M_{td} = \sqrt{M_u^2 + 0.75.M_x^2} = \sqrt{872^2 + 0{,}75.420^2} = 944{,}8\ (N.m)$$
$$= 944800\ (N.mm)$$

$$D \geq \sqrt[3]{\frac{M_{td}}{0{,}1[\sigma]}} = \sqrt[3]{\frac{944800}{0{,}1.70}} = 51{,}3\ (mm)$$

Choose $D = 52\ mm = 0{,}052\ m$

Step 3: Test shaft durability

The clutch shaft is made of 50X steel, with allowable stresses:

$[\sigma_{th}]$ = 1200KG/cm^2

Tested according to bending stress:

At dangerous sections, bending stress is determined by the following formula:

$$\sigma_u = \frac{M_u}{W_u} \leq [\sigma_u]$$

Note:

- W_u: Bending moment, due to a solid shaft, we have: $W_u = 0{,}1.d^3$

- M_u: The composite bending moment at the critical section of the shaft, M_u is determined:

$$M_u = \sqrt{M_{ux}^2 + M_{uy}^2} = \sqrt{21{,}7^2 + 871{,}72^2} = 872\ (N.m)$$

$$= 872000\ (N.mm)$$

$$\sigma_u = \frac{M_u}{W_u} = \frac{872000}{0{,}1.102^3} = 8{,}21\ (N/mm^2)$$

Torsional stress testing:

$$\tau_x = \frac{M_x}{W_x} < [\tau_x]$$

Where:

- M_x: torque at X, $M_x = 420\ N.m$

- W_x: torsional moment at X, with the solid shaft, we have $W_x = 0{,}2.d^3$

Subtitute into the equation, we have:

$$W_x = 0{,}2.d^3 = 0{,}2.(102)^3 = 212241{,}6\ (mm^3)$$

Therefore, the torsional stress is:

$$\tau_x = \frac{M_x}{W_x} = \frac{420}{212241{,}6} = 0{,}00198\ (N/mm^2)$$

So the composite bending and torsional stress is calculated using the formula:

$$\sigma_{th} = \sqrt{\sigma_u^2 + 4.\tau_x^2} \le [\sigma_{th}] = \sqrt{8{,}21^2 + 4.0{,}0018^2} = 8{,}21\ (N/mm^2)$$

$$\sigma_{th} = 8{,}21.10^6 (N/m^2) \le [\sigma_u]$$

In conclusion, the shaft has enough allowable durability.

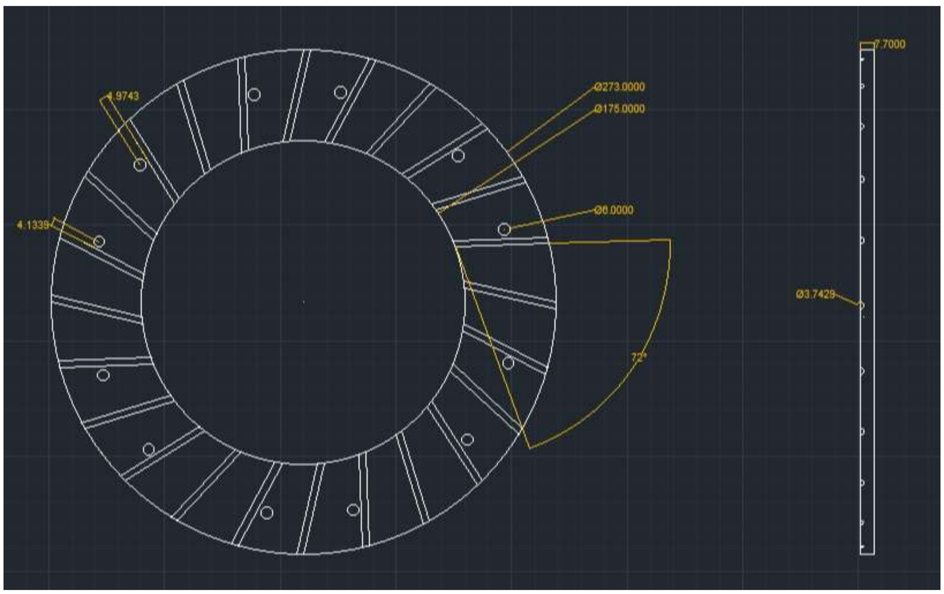
C. DESIGN THE CLUTCH ASSEMBLY BY USING AUTOCAD

The link drive below includes AutoCAD files (dwg, pdf), Project Report (PDF, Word, PPT), Presentation video.

<u>AUTOCAD - DESIGN AND CALCULATION FOR</u>

<u>THE CLUTCH OF TOYOTA FORTUNER 2021</u>

1. Friction facing



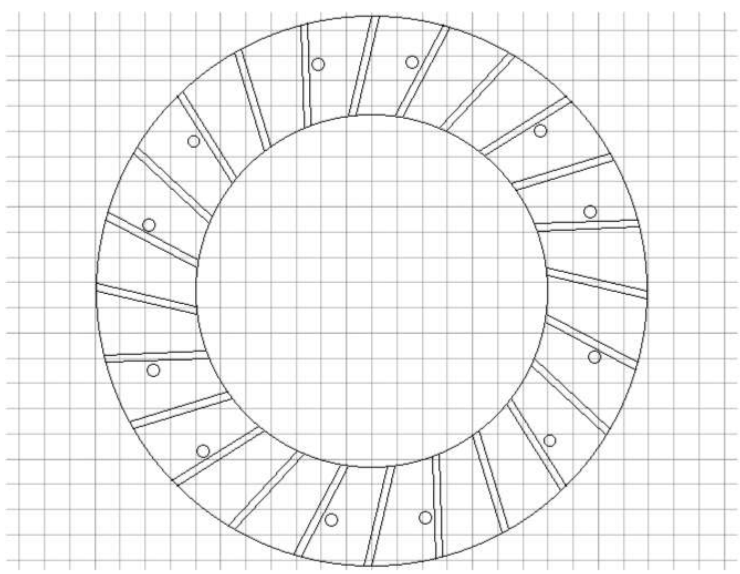


Figure 44,45: Friction facing in AutoCAD

2. Disc hub

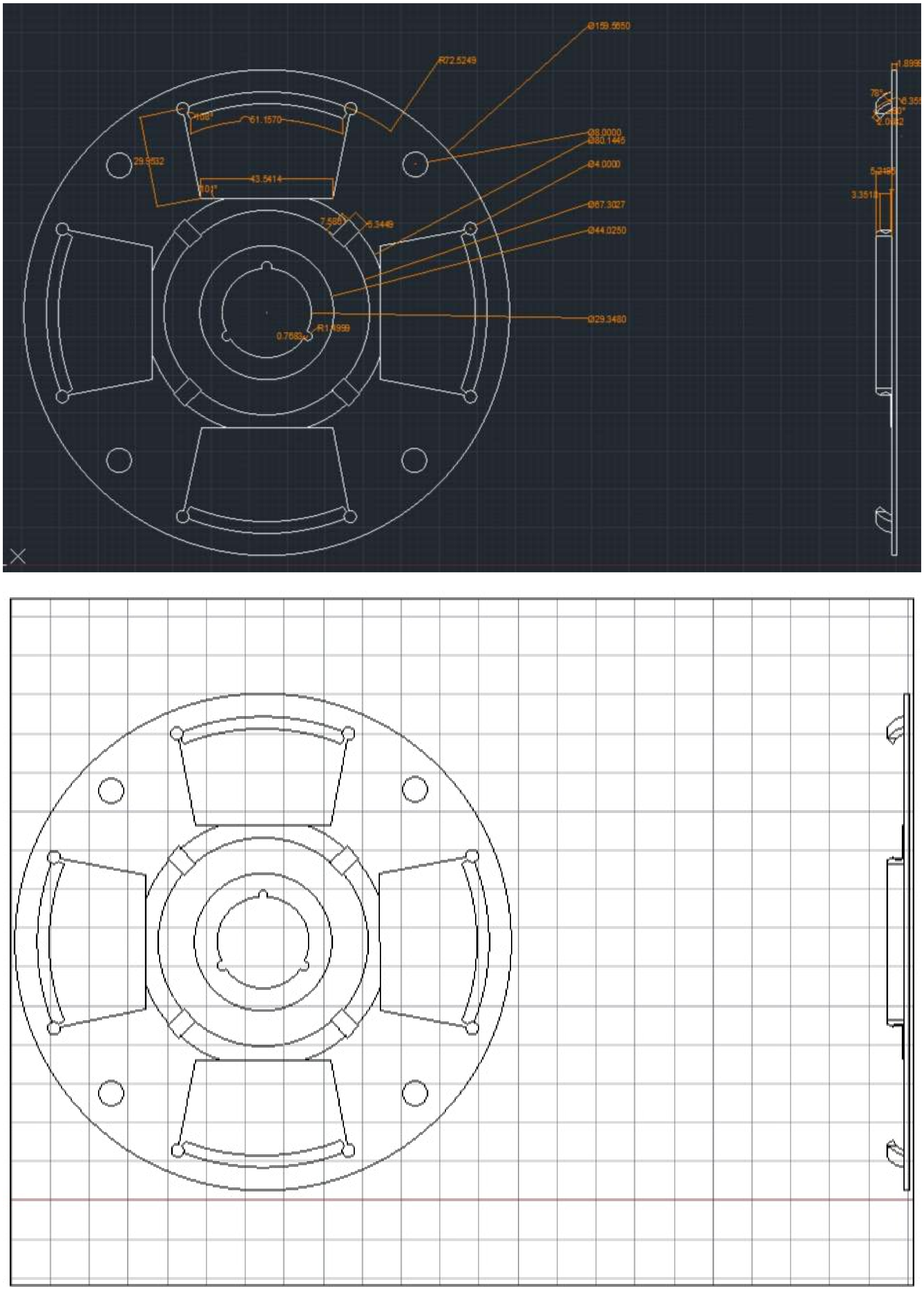
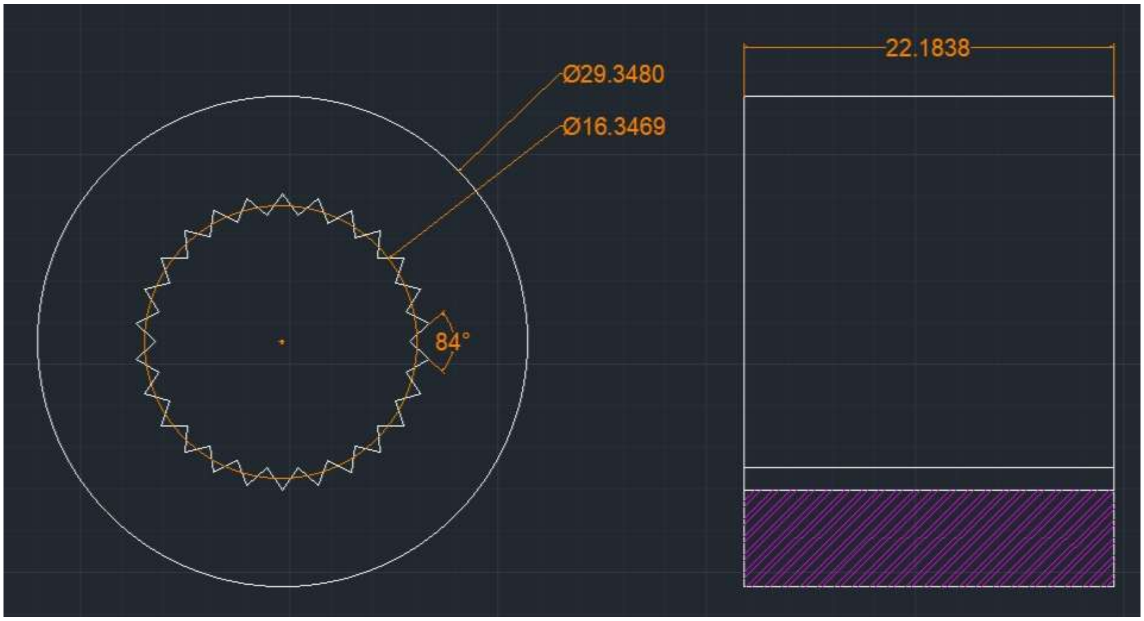


Figure 46, 47: Disc hub in Auto CAD

3. Splined hub



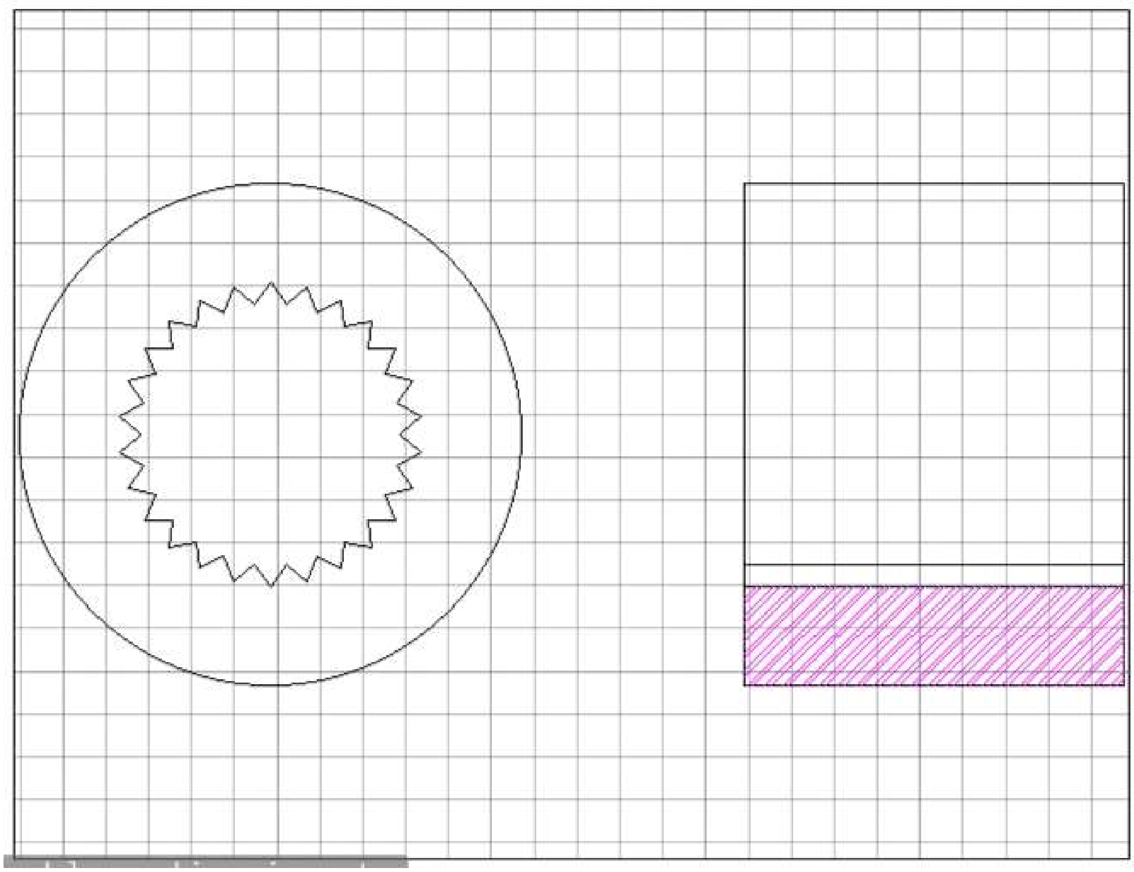
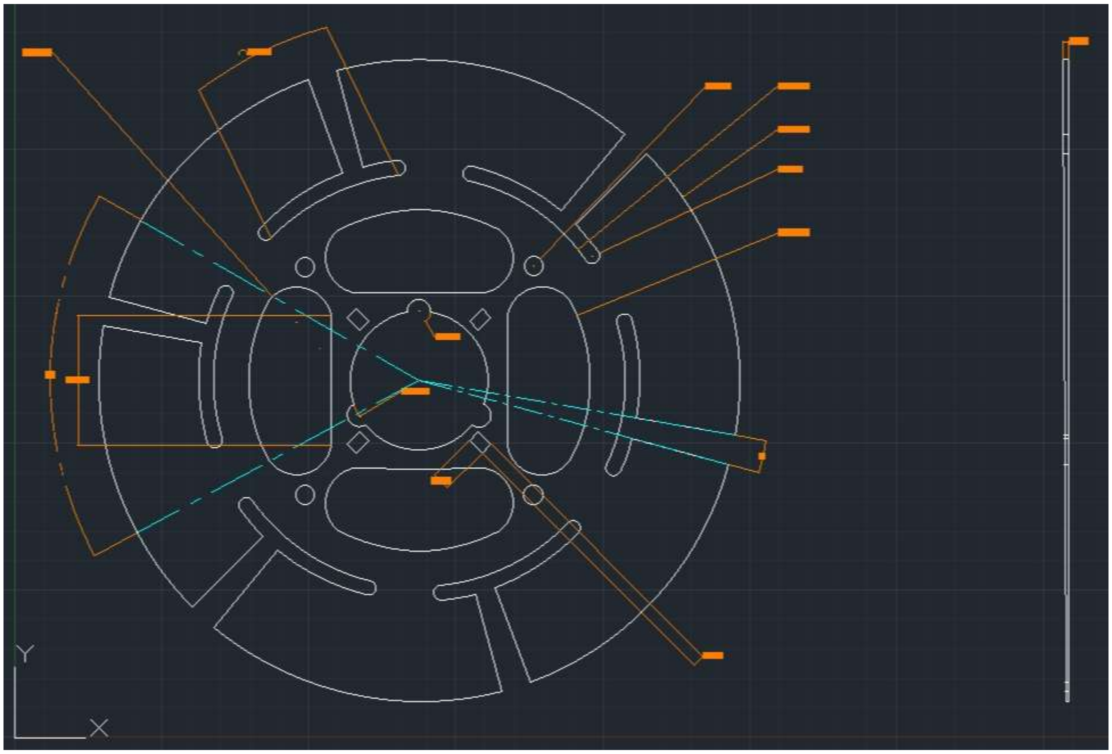


Figure 48, 49: Splined hub in Auto CAD

4. Driven plate



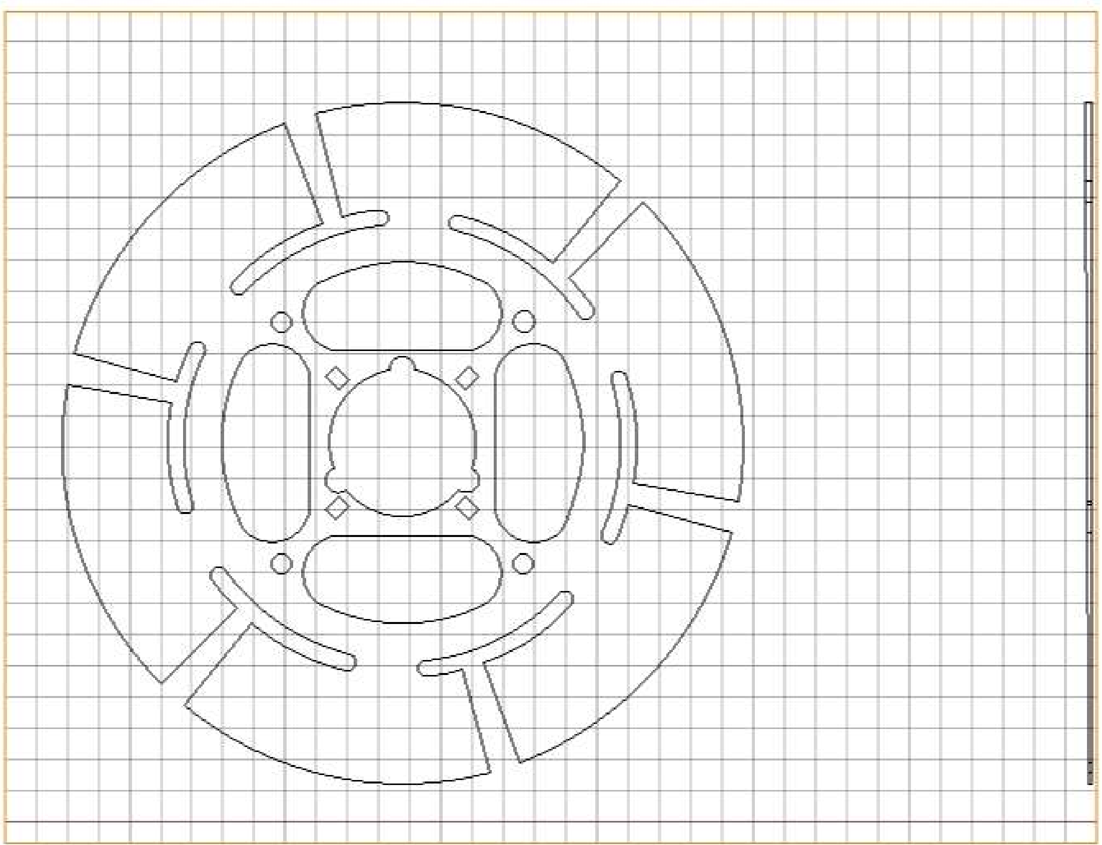
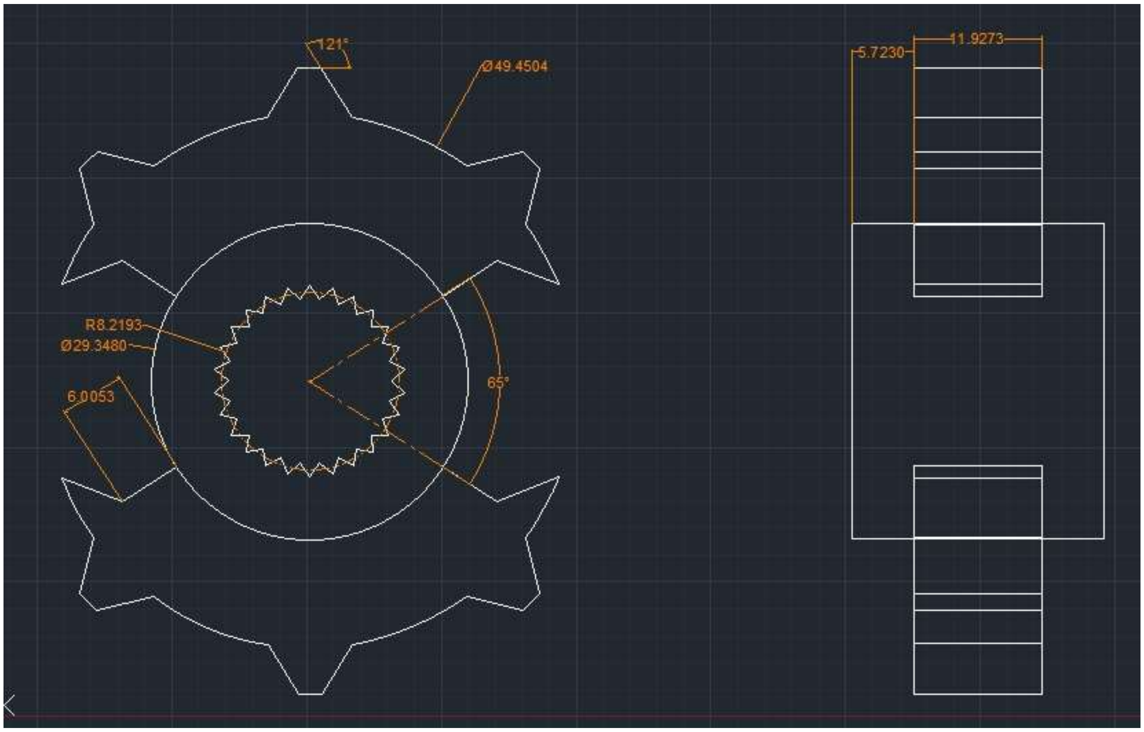


Figure 50, 51: Driven plate in AutoCAD

5. Hub flange



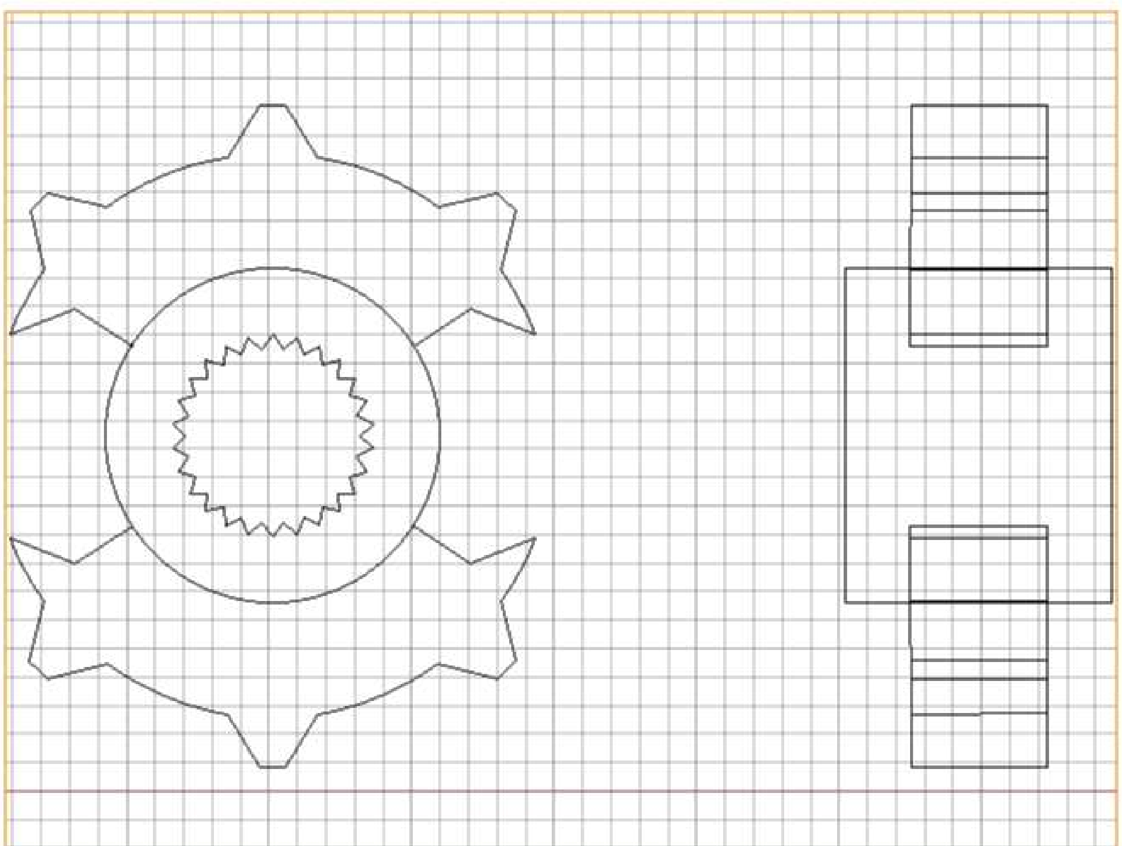
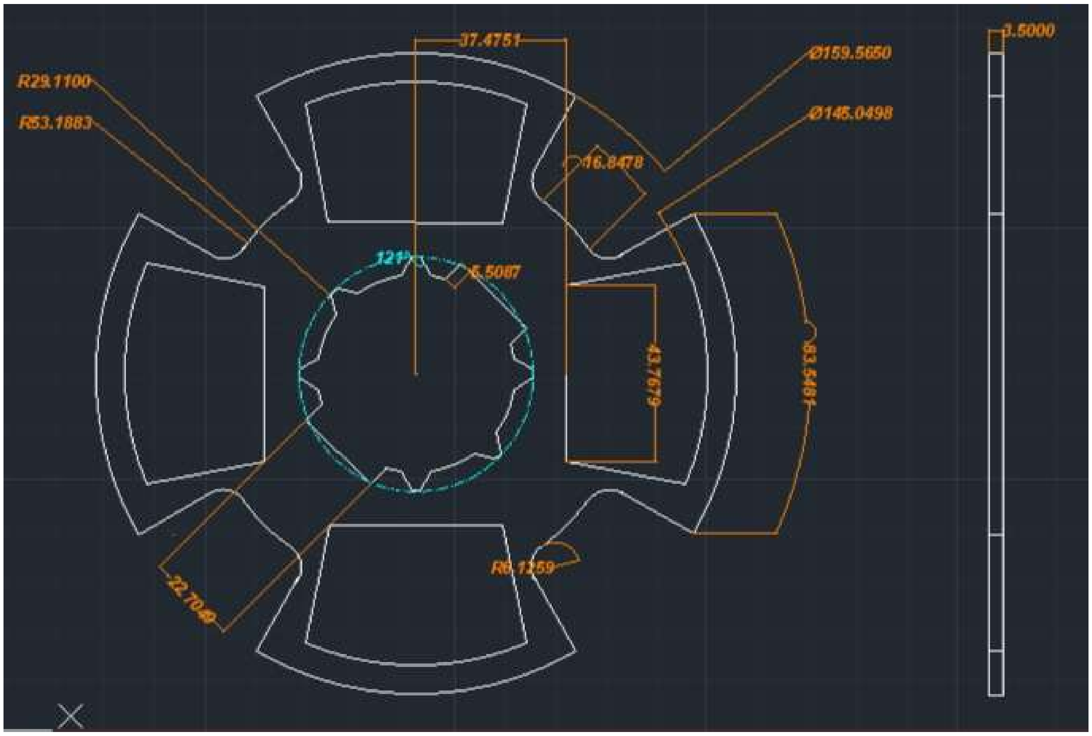


Figure 52, 53: Hub flange

6. Main plate



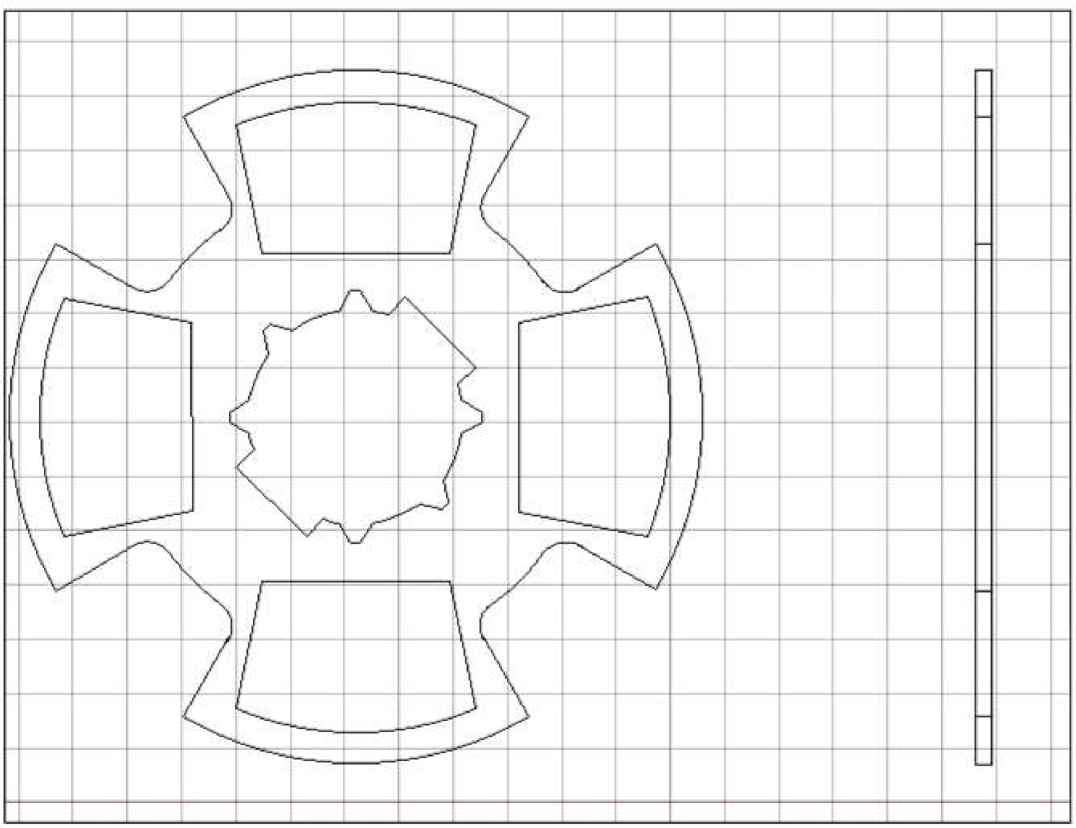
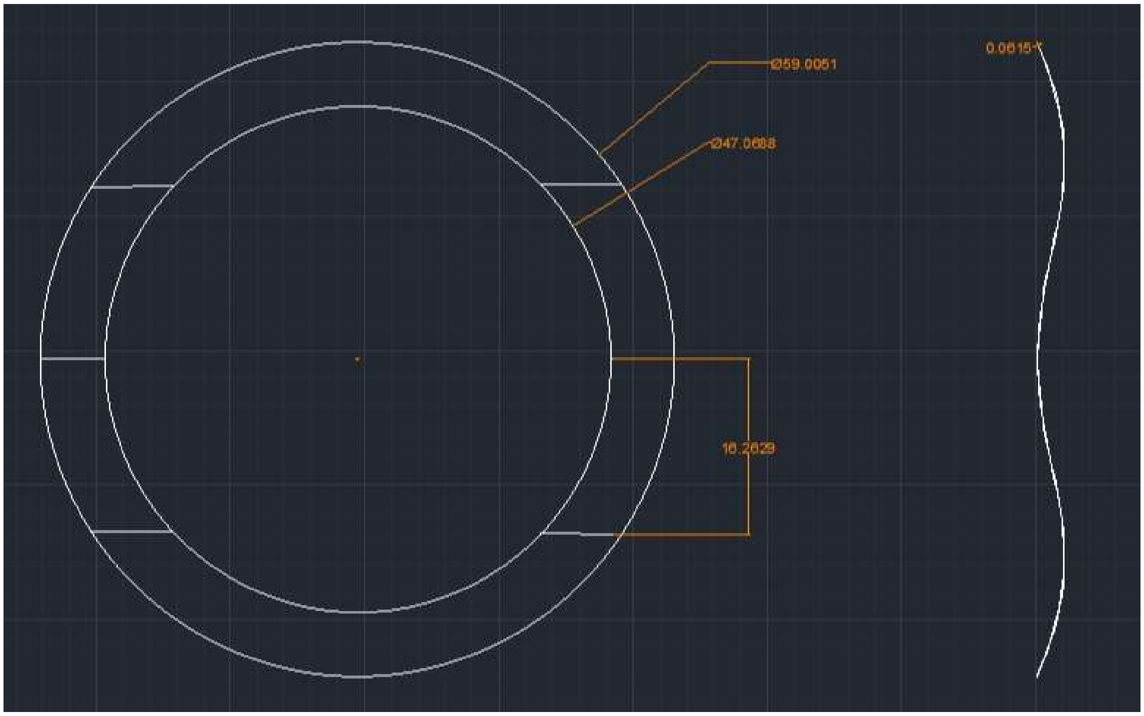


Figure 54, 55: Main plate in AutoCAD

7. Clutch disc component 1, 2

Clutch disc component 1



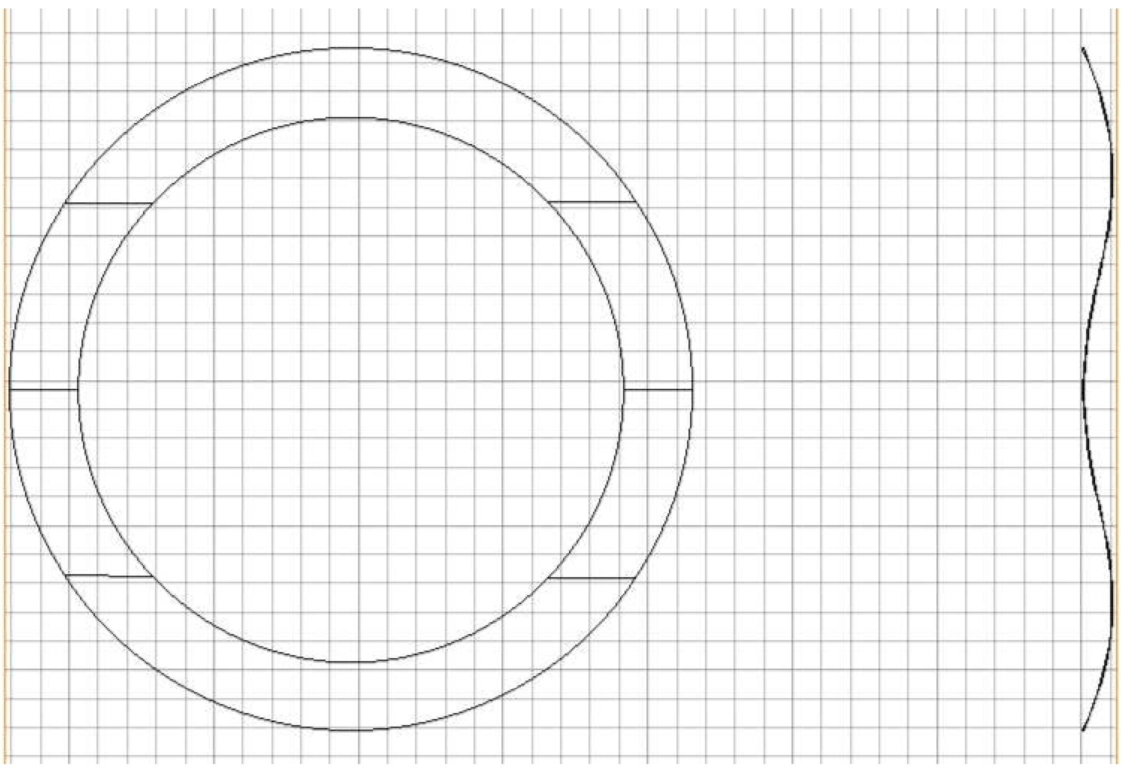
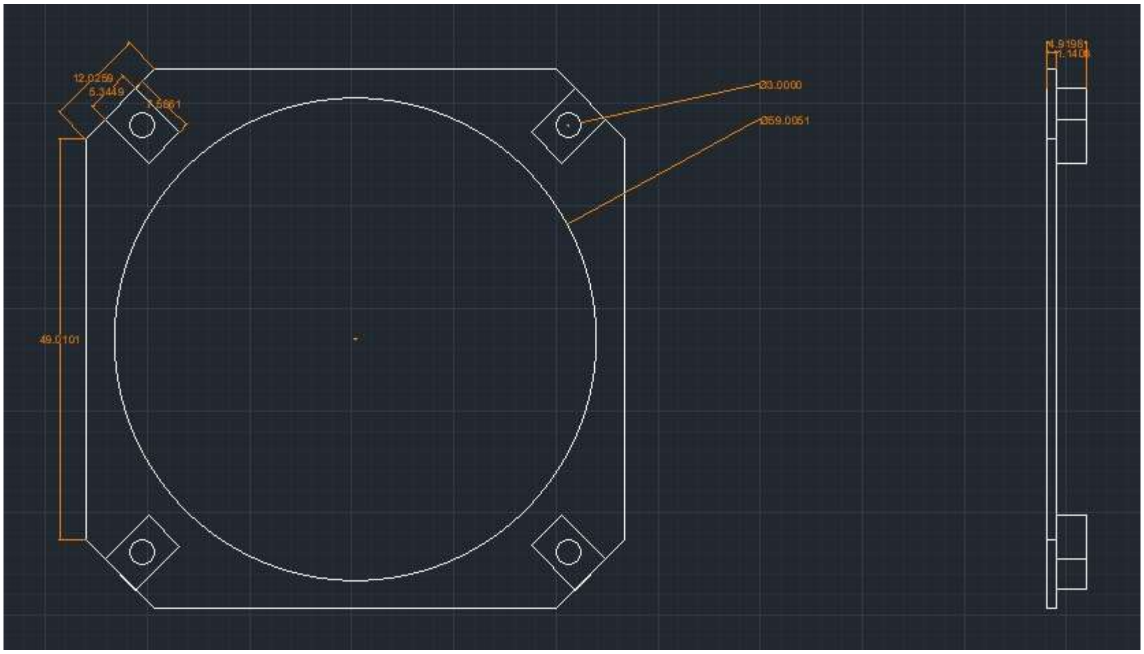


Figure 56, 57: Clutch disc component 1 in AutoCAD

Clutch disc component 2



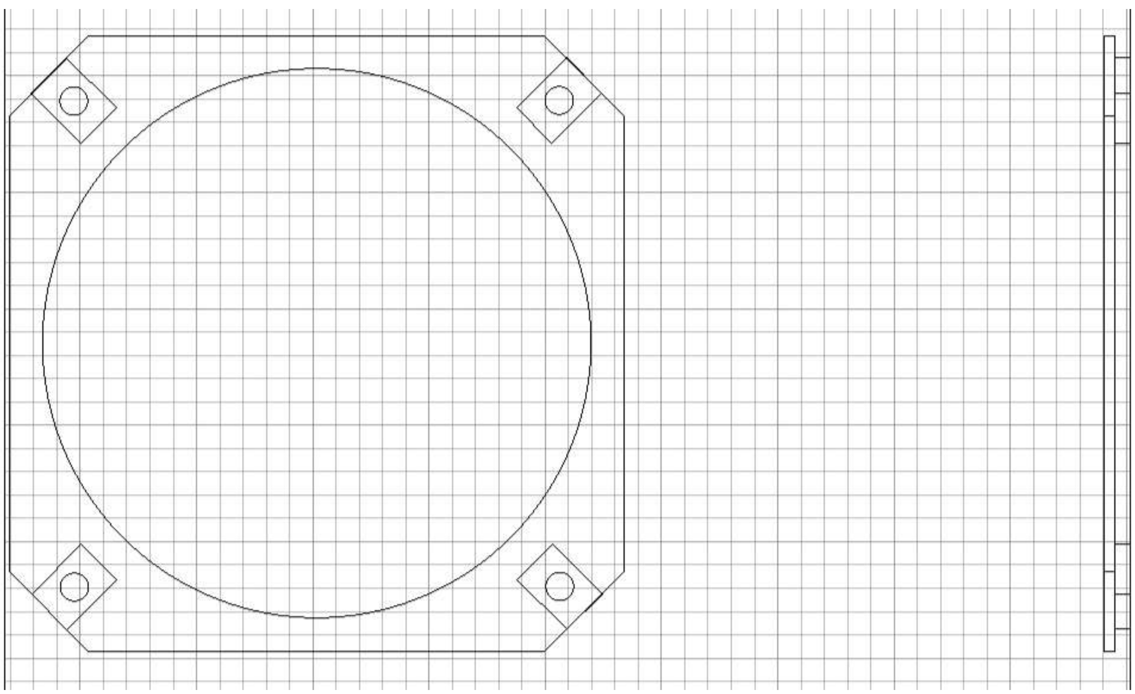
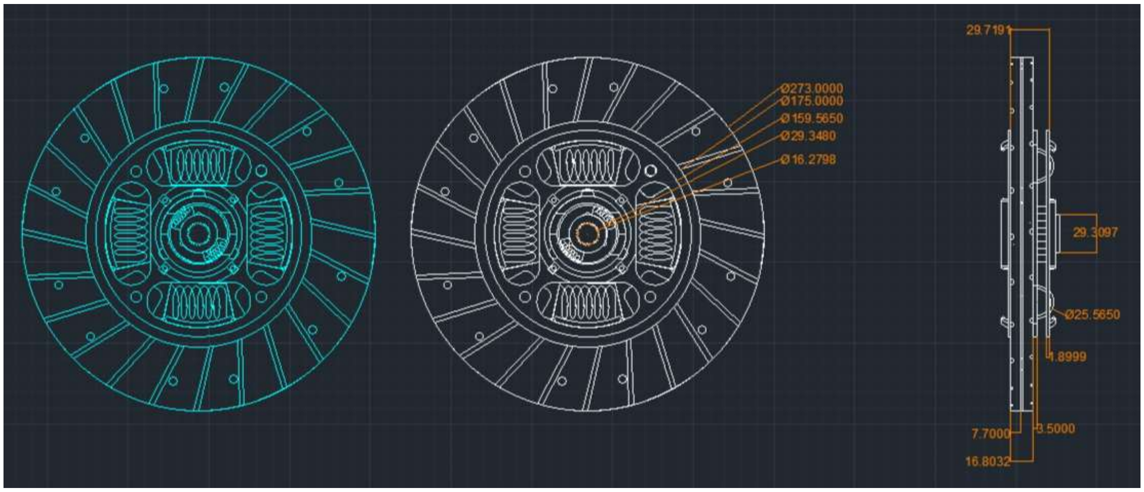


Figure 58, 59: Clutch disc component 2

8. Clutch disc assembly



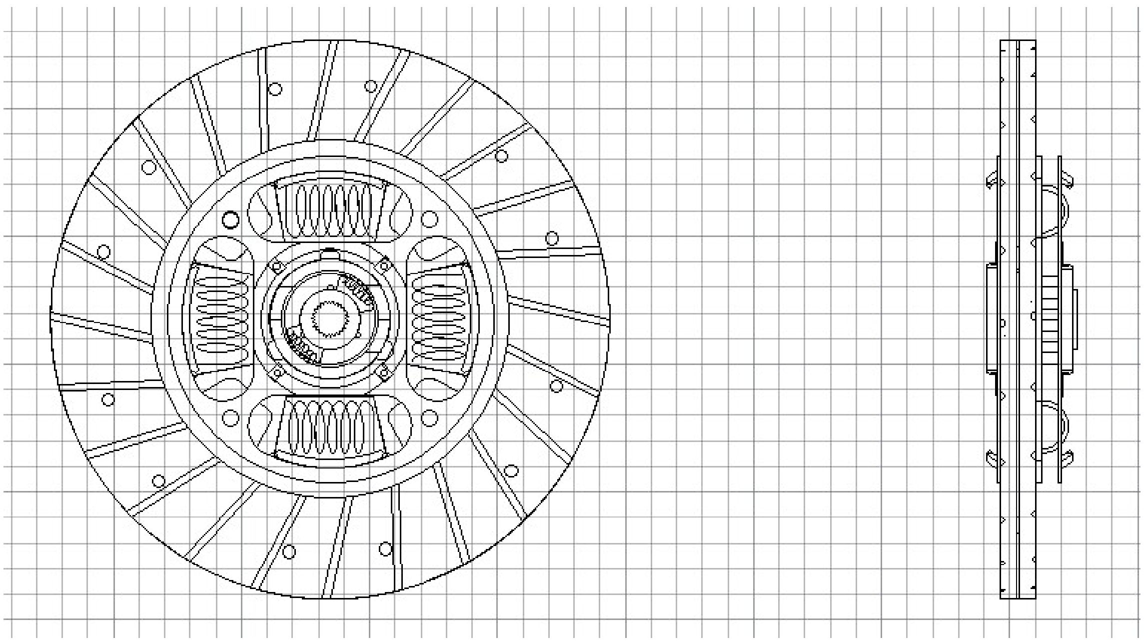


Figure 60, 61: Clutch disc assembly in AutoCAD

9. Flywheel

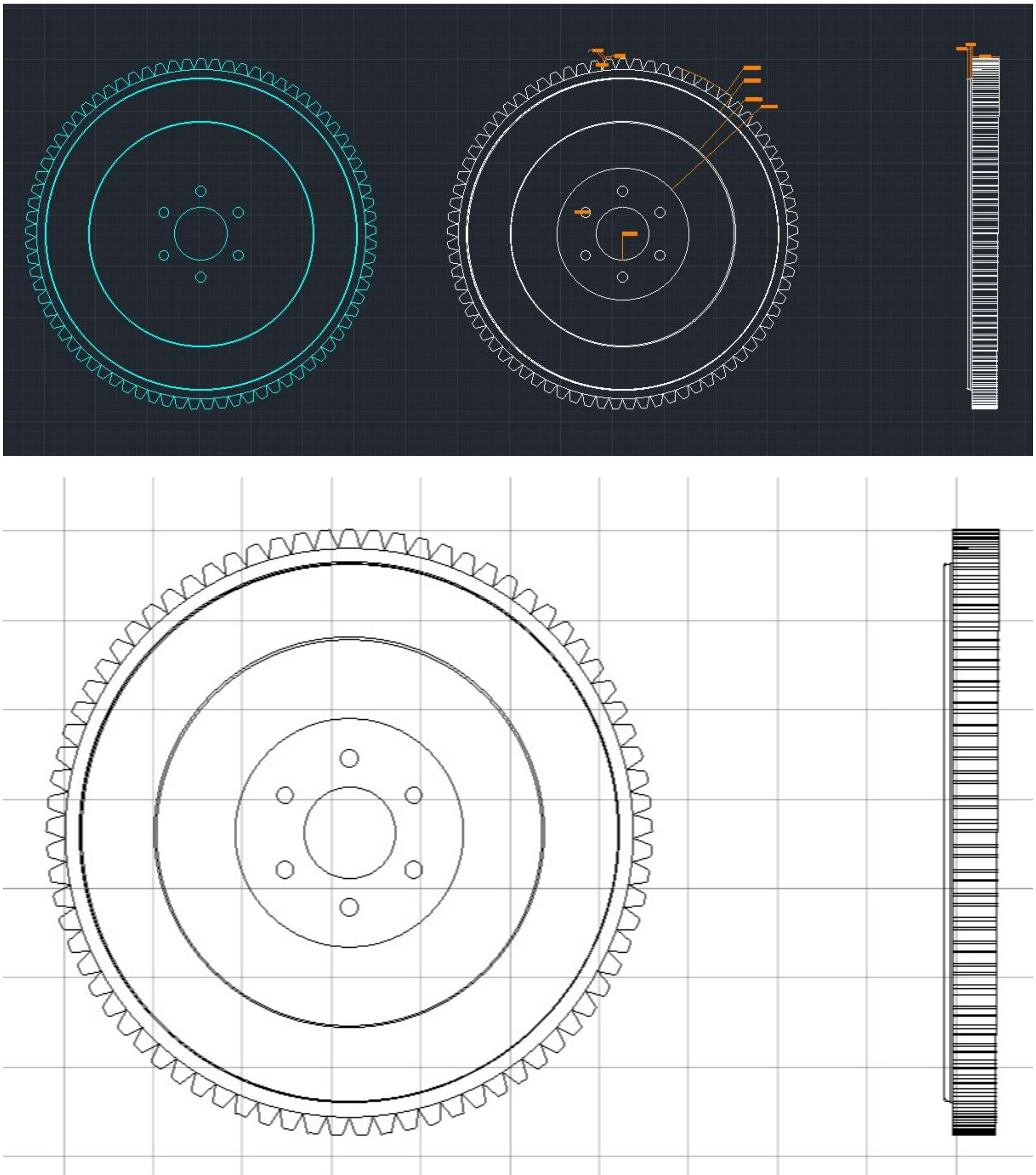
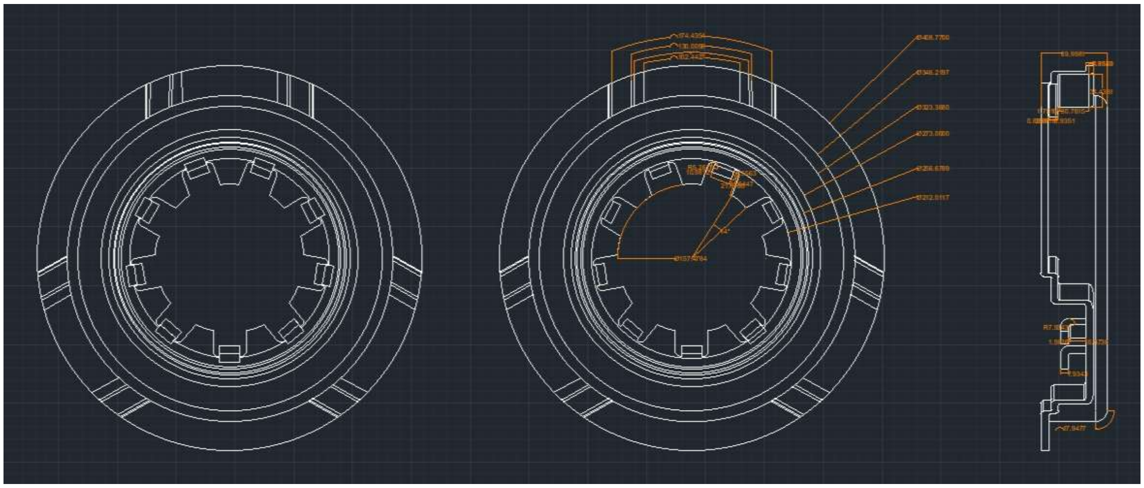


Figure 62, 63: Flywheel in AutoCAD

10. Clutch cover 1, 2

Clutch cover 1



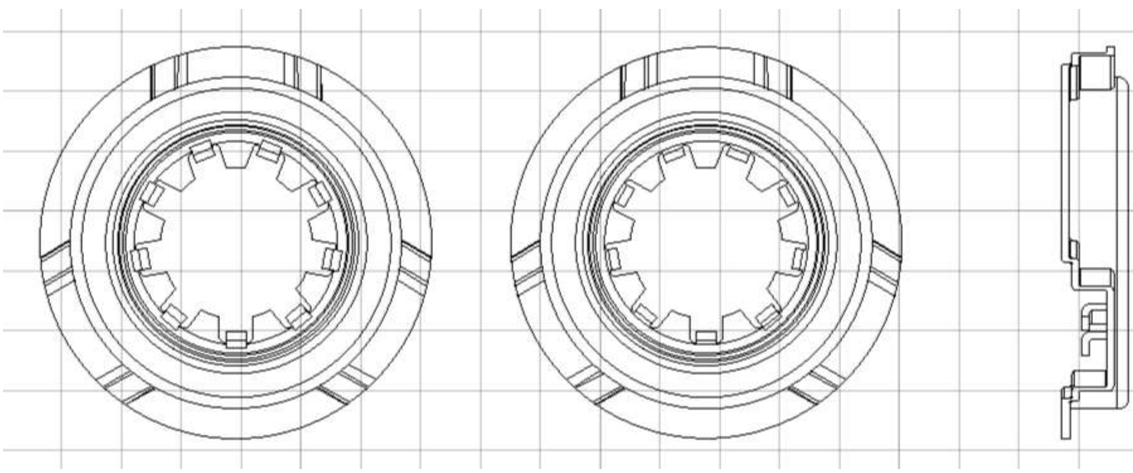


Figure 64, 65: Clutch cover 1 in AutoCAD

Clutch cover 2

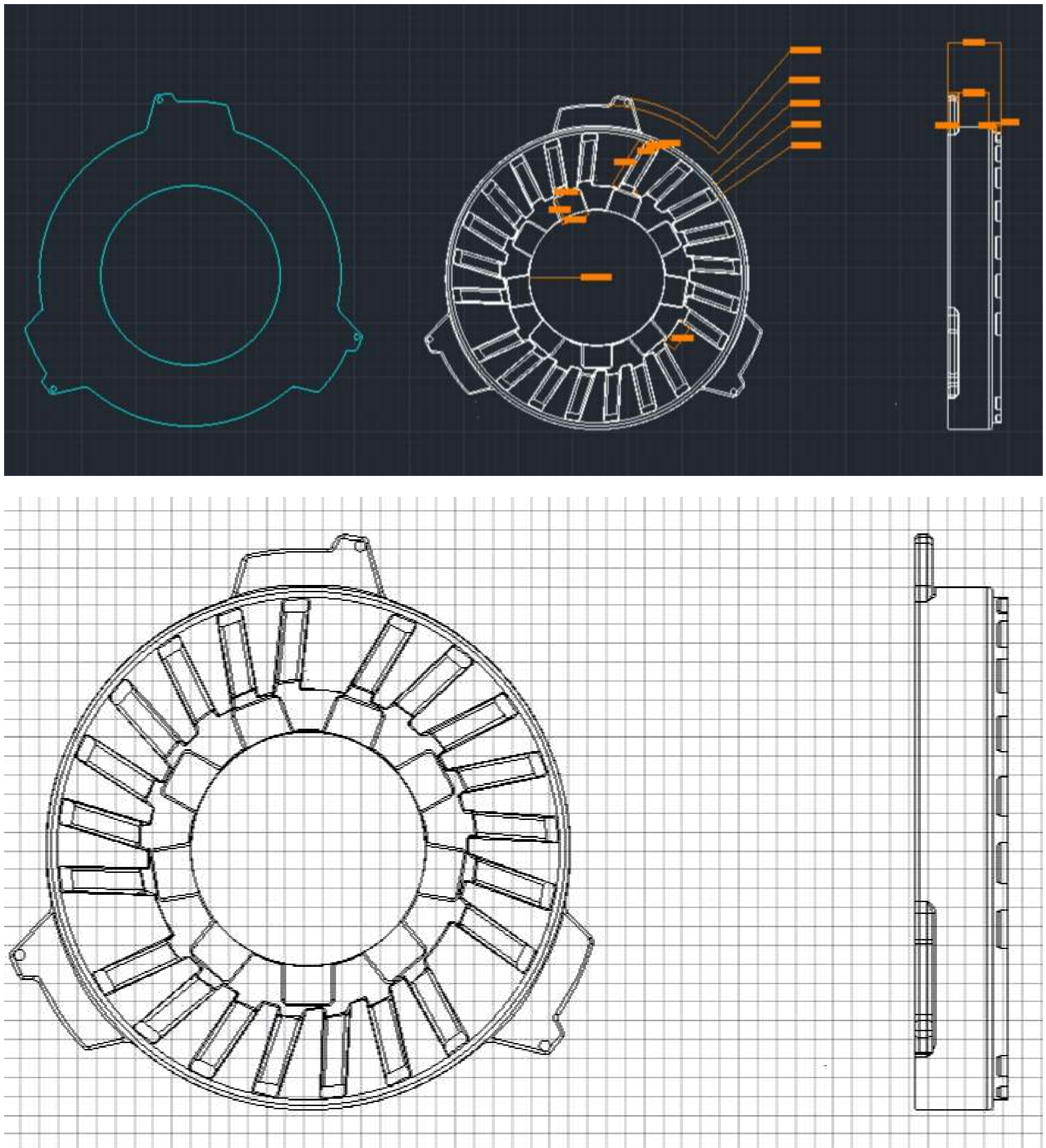


Figure 66, 67: Clutch cover 2 in AutoCAD

11. Diaphragm spring

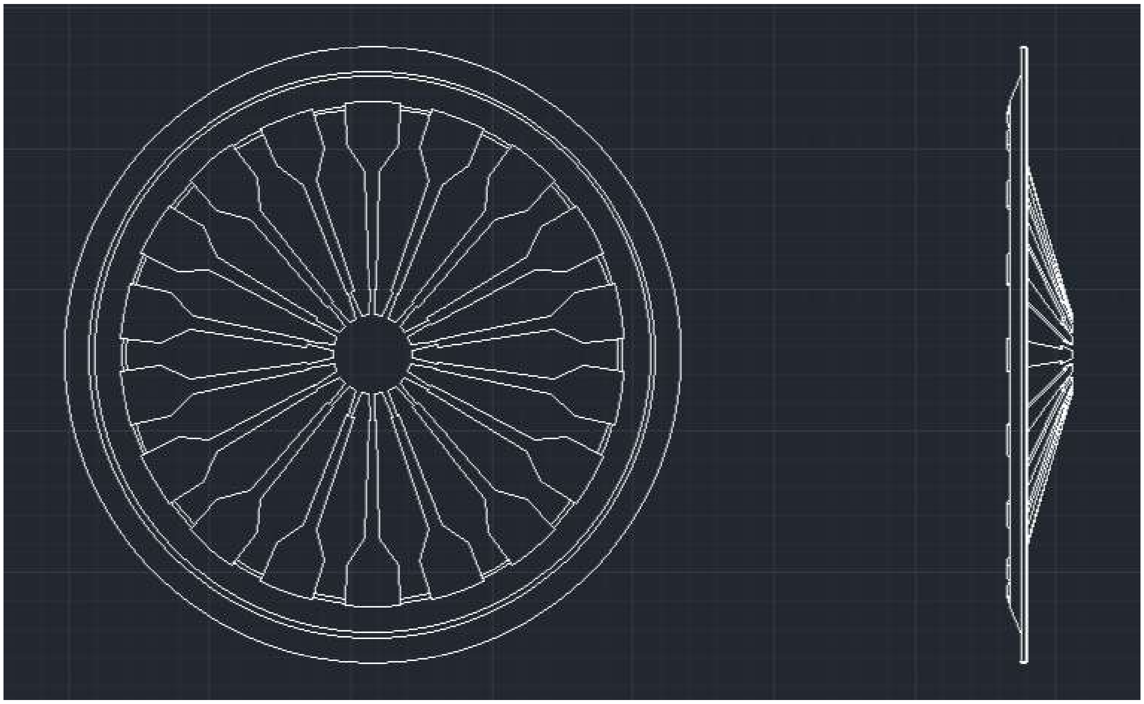
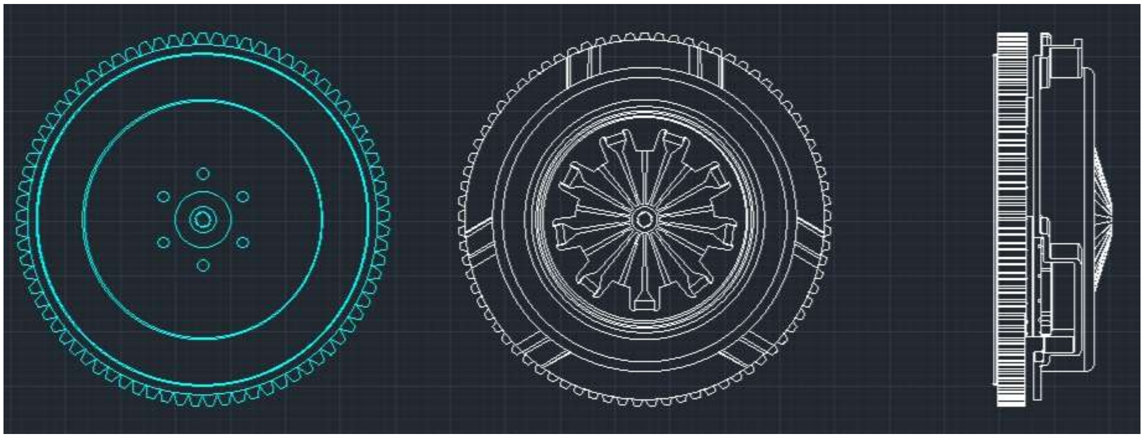


Figure 68: Diaphragm spring in AutoCAD

12. Clutch assembly



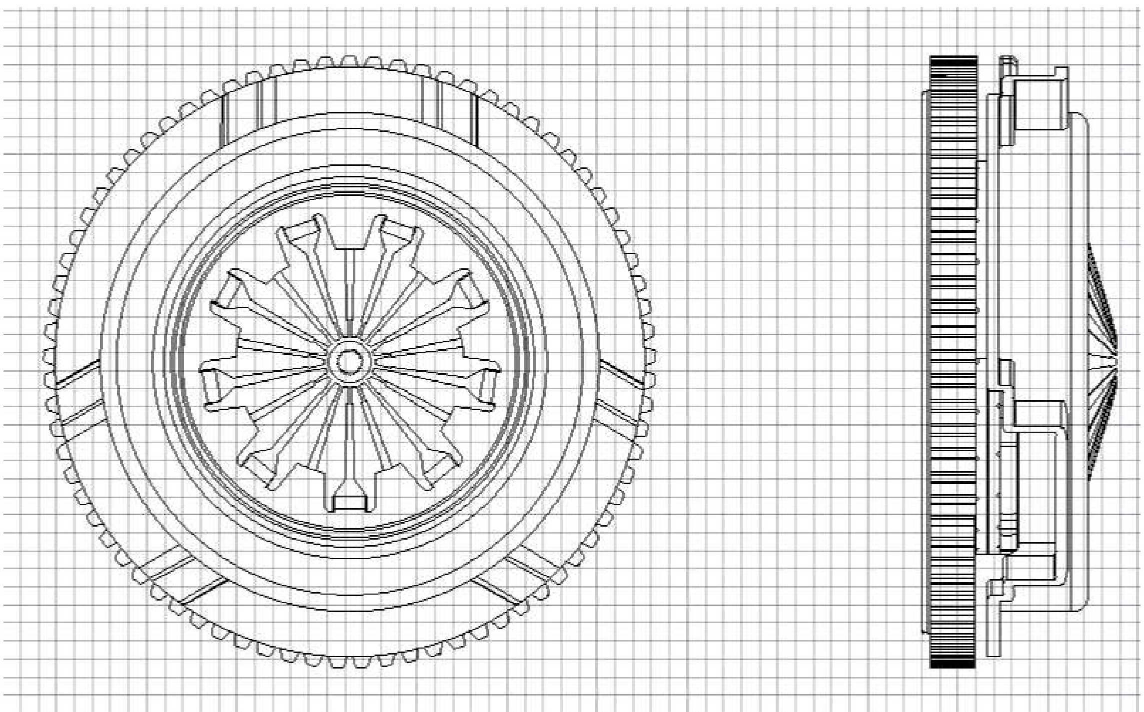
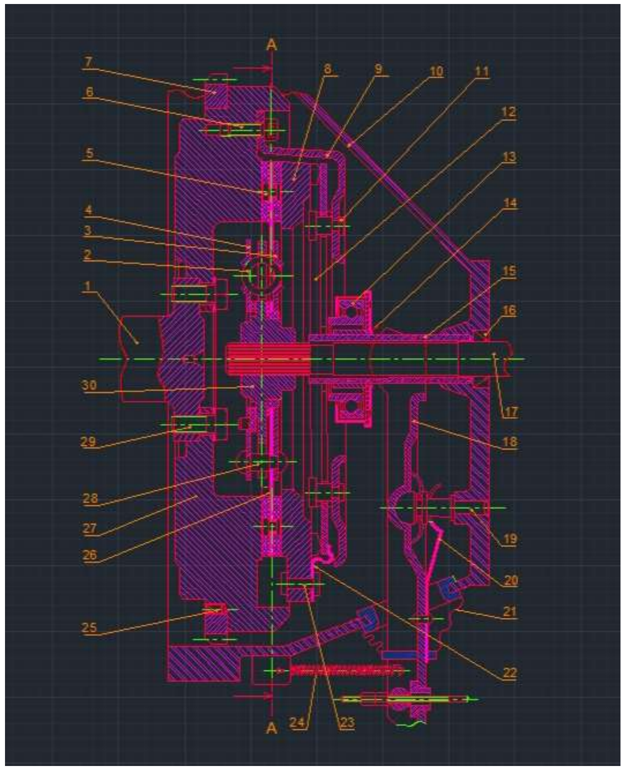


Figure 69, 70: Clutch assembly in AutoCAD

13. Cross section of the clutch



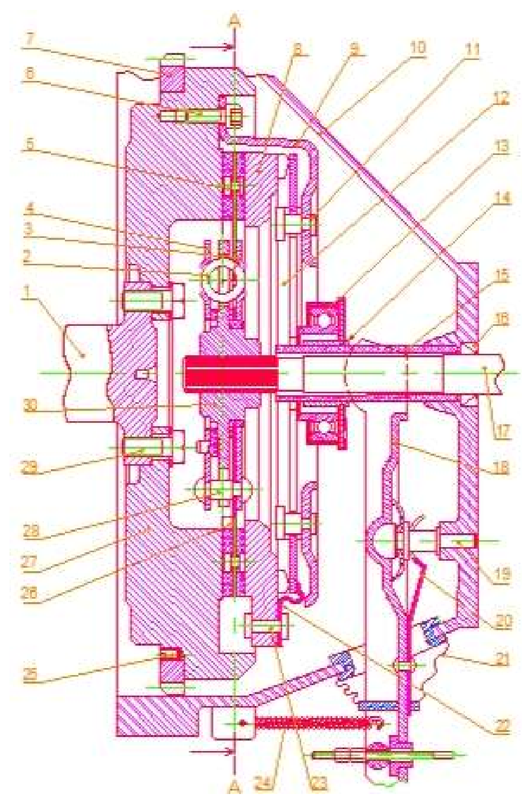


Figure 71, 72: Cross section of the Clutch in AutoCAD

14. Clutch control system

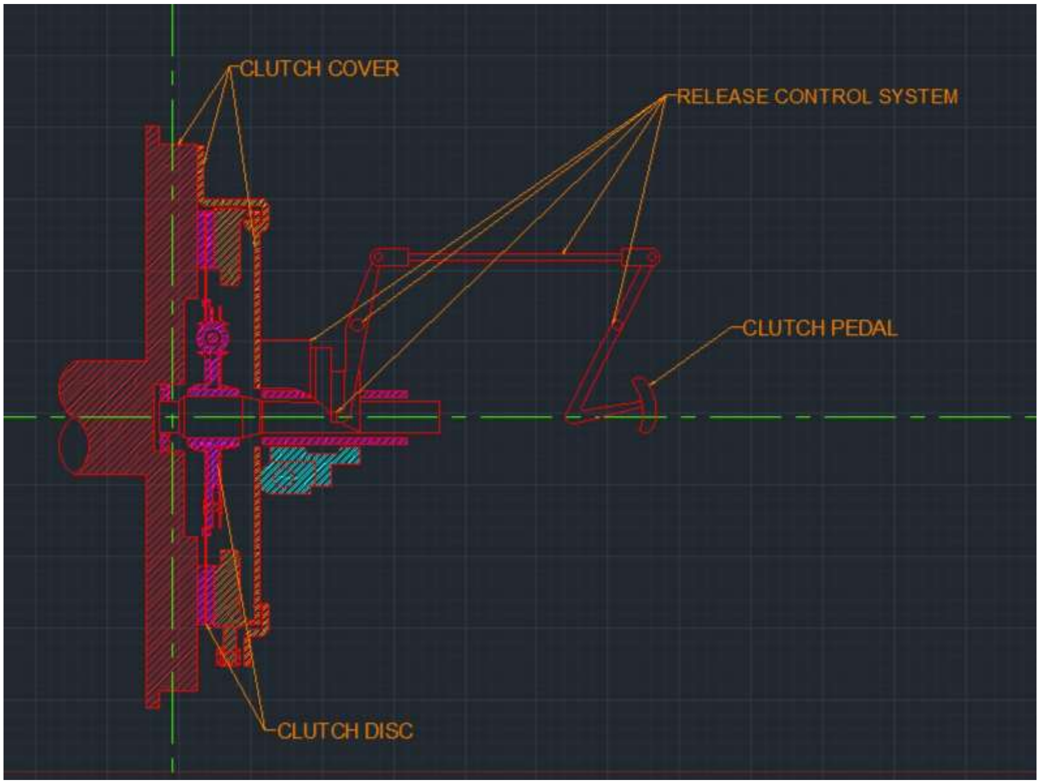


Figure 73: Clutch control system

D. CONCLUSION

In conclusion, the project on the design and calculation of the clutch system for the Toyota Fortuner 2021 model has successfully achieved its objectives. The research focused on developing an optimized clutch design that meets the performance requirements and durability standards set by Toyota.

Through a comprehensive analysis of the clutch system, various factors such as torque capacity, frictional characteristics, engagement and disengagement speed, and overall system integration were taken into account. Design parameters including clutch disc materials, pressure plate specifications, and actuation work were carefully considered to ensure optimal performance and longevity.

The research findings provide valuable insights into the design and calculation aspects of the clutch system for the Toyota Fortuner 2021 model. The optimized clutch design not only ensures efficient power transmission but also enhances the driving experience by enabling smooth gear shifting and reducing wear on transmission components. This contributes to improved performance, durability, and customer satisfaction in automotive engineering.

The project utilized a systematic approach to design and calculation, considering fundamental parameters of the Toyota Fortuner 2021 model and carefully evaluating the details of the clutch assembly. The use of AutoCAD further enhanced the design process, allowing for precise and accurate representation of the clutch assembly.

Overall, the project's outcomes contribute to the advancement of clutch systems, specifically in the context of the Toyota Fortuner 2021 model. The research serves as a valuable reference for future clutch designs, promoting enhanced performance, durability, and customer satisfaction in the automotive industry.

E. REFERENCE

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