CZECH TECHNICAL UNIVERSITY IN PRAGUE

FACULTY OF MECHANICAL ENGINEERING

DEPARTMENT OF AUTOMOTIVE, COMBUSTION ENGINE AND RAILWAY ENGINEERING

MASTER THESIS

Pneumatic System for Gear and Clutch Control

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Program: Master of Automotive Engineering

Specialization: Advanced Powertrain

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MASTER'S THESIS ASSIGNMENT

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II. Master's thesis details

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Pneumatic System for Gear and Clutch Control

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Pneumatický systém pro ovládání spojky a řazení

Guidelines:

Design pneumatic system for the clutch and gearbox actuation of a hybrid sports car. The task is to design the following:
-pneumatic piston to operate the clutch master cylinder
-test bench design to verify the clutch actuation system
-clutch control simulation in Simulink for the WLTP cycle and various drive modes (ICE only, hybrid, EV only)
-gearchange simulation in Simulink for WLTP cycle

Bibliography / sources:

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Date of master's thesis assignment: **19.10.2022** Deadline for master's thesis submission: **11.01.2023**

Assignment valid until: ______________

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

III. Assignment receipt

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

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Declaration

I, Sai Kalyan Achanta, declare that this thesis and the work presented in it, titled “Pneumatic System for Gear and Clutch Control”, are my own and have been generated by me as the result of my original research. I confirm that this work was done wholly or mainly while in candidature for a master’s degree in Automotive Engineering at Czech Technical University. Where any part of this thesis has previously been submitted for a degree or any other qualification at this University or any other institution, this has been clearly stated. Where I have consulted the published work of others, this is always clearly attributed. Where I have quoted from the work of others, the source is always given. With the exception of such quotations, this thesis is entirely my work. I have acknowledged all main sources of help. None of this work has been published before submission.

Date: Wednesday, January 11, 2023 Signature:

Acknowledgements

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Last but not least, my heartfelt gratitude goes to my family for their never-ending encouragement, unwavering support, and boundless love.

Abstract

This thesis study deals with the use of pneumatic pistons to operate the clutch system of a car to automate the operation of the clutch actuation, the proposal of a testing system for the designed system, and the simulation of the proposed logic on SIMULINK. The pneumatic pistons and the corresponding electrically operated pneumatic solenoid valve were decided based on the requirements to operate the clutch designed for the high-performance engine. The testing system was proposed to test the designed pneumatic system to check the working of the clutch actuation according to various types of actuation cases which deal with both low-speed and high-speed gear changes involving faster clutch functionality and modulation to allow the vehicle to start moving from a standing start. The logic for the clutch actuation for these conditions was verified using MATLAB SIMULINK for a WLTP cycle which simulated the city driving conditions.

Keywords: Clutch, Pneumatics, Solenoids, SIMULINK.

Abstrakt

Tato diplomová studie se zabývá použitím pneumatických pístů k ovládání spojkového systému automobilu k automatizaci provozu ovládání spojky, návrh zkušebního systému pro navržený systém, a simulace navrhované logiky na SIMULINK. Na základě požadavků na provoz spojky určené pro vysoce výkonný motor, bylo rozhodnuto o pneumatických pístech a odpovídajícím elektricky ovládaném pneumatickém solenoidovém ventilu. Testovací systém byl navržen tak, aby testoval navržený pneumatický systém pro kontrolu fungování aktivace spojky podle různých typů ovládacích případů, které se zabývají jak nízkými rychlostmi, tak vysokorychlostními změnami rychlostních stupňů, které zahrnují rychlejší funkčnost spojky a modulaci, aby vozidlo mohlo začít pohyb od stálého startu. Logika pro ovládání spojky pro tyto podmínky byla ověřena pomocí MATLAB SIMULINK pro cyklus WLTP, který simuloval jízdní podmínky města.

Klíčová slova: Spojka, Pneumatika, Solenoidy, SIMULINK.

Nomenclature

A_m	Area of the master cylinder piston
A_{rb}	Area of the slave cylinder piston
R_s	Averaged frictional clutch radius
Φ	Bore diameter of the pneumatic piston
F_A	Clamp Force on clutch plates
C	Clutch Torque capacity
C_{geom}	Coefficient of the geometry of diaphragm spring
δ	Deflection of spring
A	Distance from Cover fulcrum to Pressure fulcrum
B	Distance from Cover fulcrum to spring finger bearing contact point
D	External Diameter of Diaphragm Spring
R_e	External Radius of the Clutch
R	External Radius of the slave cylinder piston
R_b	Force on the release bearing
μ	Friction coefficient
h	Height of spring
p_{hyd}	Hydraulic Line pressure
p_{hyd}	Hydraulic Line pressure
D	Internal Diameter
R_i	Internal Radius of the Clutch
r	Internal Radius of the slave cylinder piston
C_{eng}	Maximum engine torque
E	Modulus of Elasticity
n	Number of Frictional Surface
P_n	Operating pneumatic pressure
F_p	Pedal force
ν	Poisson coefficient
p	Pressure
r_p	Radius of the master cylinder piston
β	Safety coefficient

S	Surface area
F_t	Tangential force
e	Thickness of spring

Abbreviations

AC	Alternating Current
DC	Direct Current
DAC	Double Acting Cylinder
HIL	Hardware-in-loop
NC	Normally Closed
NO	Normally Open
PDM	Pneumatic Double Acting Magnetic
PWM	Pulse Width Modulation
SAC	Single Acting Cylinder
WLTP	Worldwide Harmonized Light Vehicles Test Procedure

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1 Introduction

1.1 Project Background

The company, Corbellati Automobili, is developing a high-performance vehicle with a V8 engine mated to a gearbox developed in-house with 4 speeds and an EV unit with 2 gears of its own. The Clutch system chosen for the 1800 HP engines is designed to transmit about 2400 Nm of torque produced by the engine. The vehicle uses air suspension on all 4 wheels and for the rear wing actuation system. Thus, using a pneumatic system for the clutch actuation makes it easier and does not require introducing more components into the vehicle for just the clutch actuation, and research and choice of components for the systems were required which forms the basis for this thesis.

1.2 Project Goals

The objective of this project is to develop a system for the actuation of a clutch with the help of pneumatic components and propose an idea to test the proposed clutch actuation system and check the functionality of the clutch according to the desired operations. The testing process would be under various working scenarios and simulation of the logic behind the clutch system and corresponding gear shifting mechanism on MATLAB SIMULINK for the WLTP cycle for ICE, Hybrid and EV modes of operation. The simulation would help to understand the implementation of the said logic in a control unit for the gearbox that would control both the clutch actuation mechanism and the gearbox actuation mechanism.

2 Literature Review

2.1 Clutch

2.1.1 Working Principle of Clutch

When the two rotating friction surfaces come into contact and are pressed, they come together and begin to rotate at the same speed due to the action of friction. This is the working principle of a clutch. The area of the contact surface, friction material and the application pressure define the friction between the two surfaces. The flywheel (Figure 1 - 1) is mounted on the engine crankshaft (Figure 1 - 4), and the driven part is friction plates or clutch plates (Figure 1 - 3), are held between the pressure plate (Figure 1 - 9) and the Flywheel. The clutch plates are mounted on the drive shaft that facilitates the transmission of torque when no force is applied by the pressure plate to disengage the clutch plates from being in contact with the flywheel. The clutch is always engaged and transmitting torque and to facilitate the shifting of gears with the help of a release bearing and lever mechanism the clutch plates are moved to remove connection with the flywheel thus not transmitting any torque to the primary shaft of the gearbox.[3]

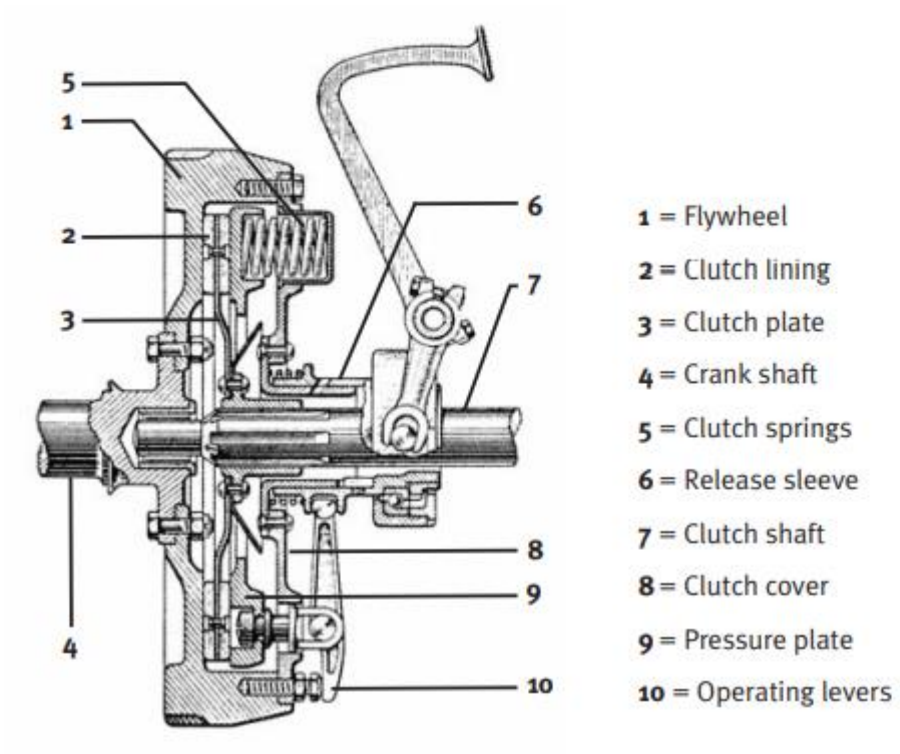


Figure 1 Clutch System

2.1.2 Diaphragm Clutch

The type of clutch which uses a diaphragm spring in place of a coil or helical spring is called the Diaphragm Clutch. The Diaphragm type springs do not have constant rate characteristics as coil springs and the pressure on the diaphragm spring keeps increasing until it is in the flat position and then decreases after passing this position. This reduces the pedal force that the driver must exert to hold the clutch in a disengaged position compared to the coil spring type.

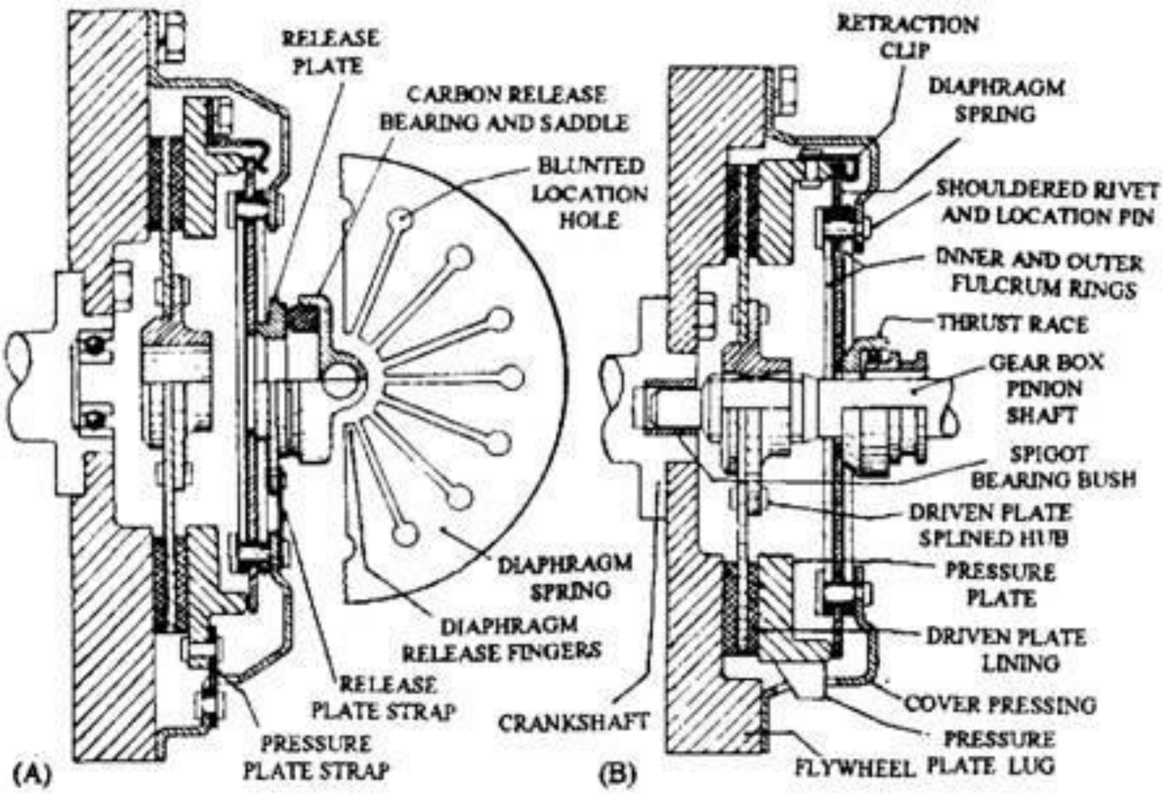


Figure 2 Diaphragm Clutch

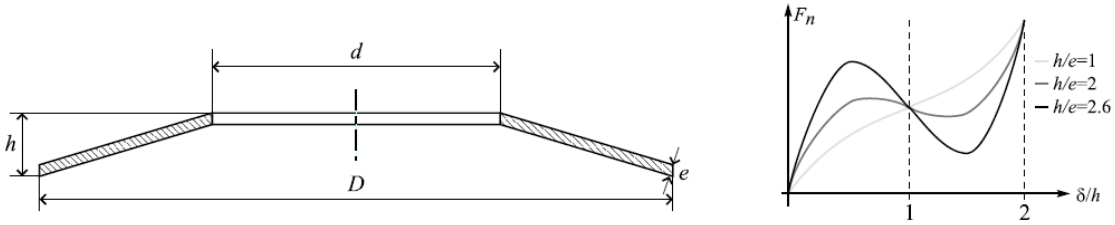


Figure 3 Geometry of a diaphragm spring and Stiffness curve

The characteristics of the diaphragm spring dictate the force required to operate the clutch and this can be seen from the spring stiffness curve in Figure 3 where h/e is the height-to-thickness ratio of the diaphragm. This h/e ratio is modified to achieve the desired pedal force. To operate the diaphragm, clutch a throw-out bearing or a central release bearing is used.

2.1.3 Working of Diaphragm Clutch

When the clutch pedal is depressed, the linkage with the clutch release fork moves the throw-out bearing towards the flywheel. As the bearing comes in contact with the diaphragm spring, it moves the diaphragm spring to move forward causing the rim to move backwards as the diaphragm pivots on the pivot ring. By reducing the pressure on the pressure plate completely the clutch plates are released from contact with the flywheel.[11]

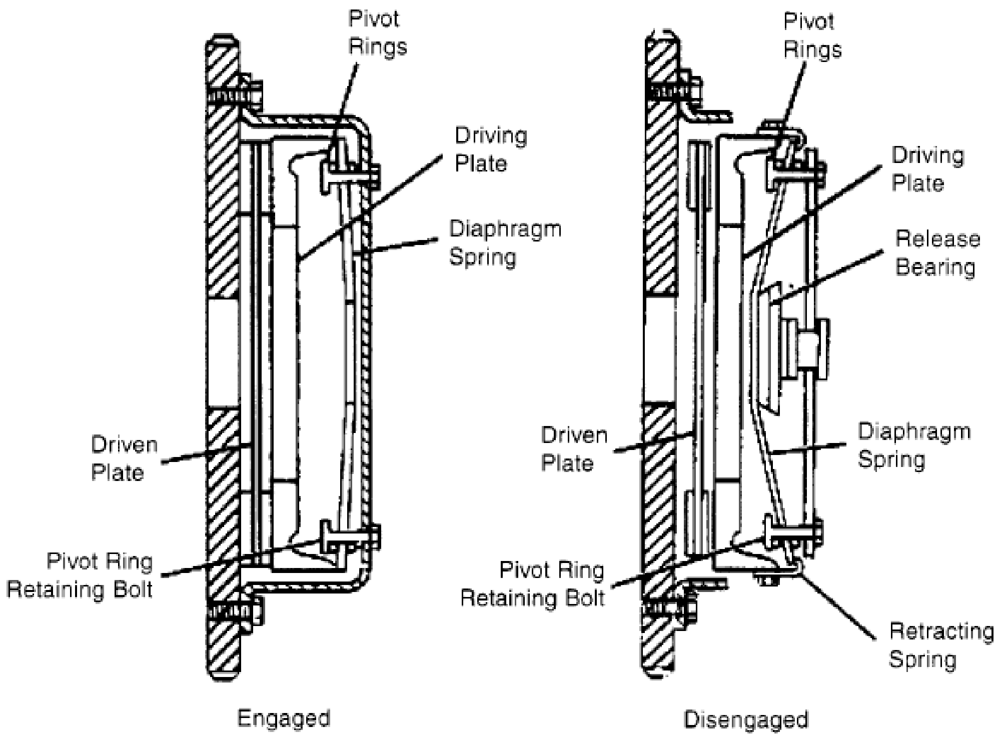


Figure 4 Engaged and Disengaged States of a Diaphragm Clutch

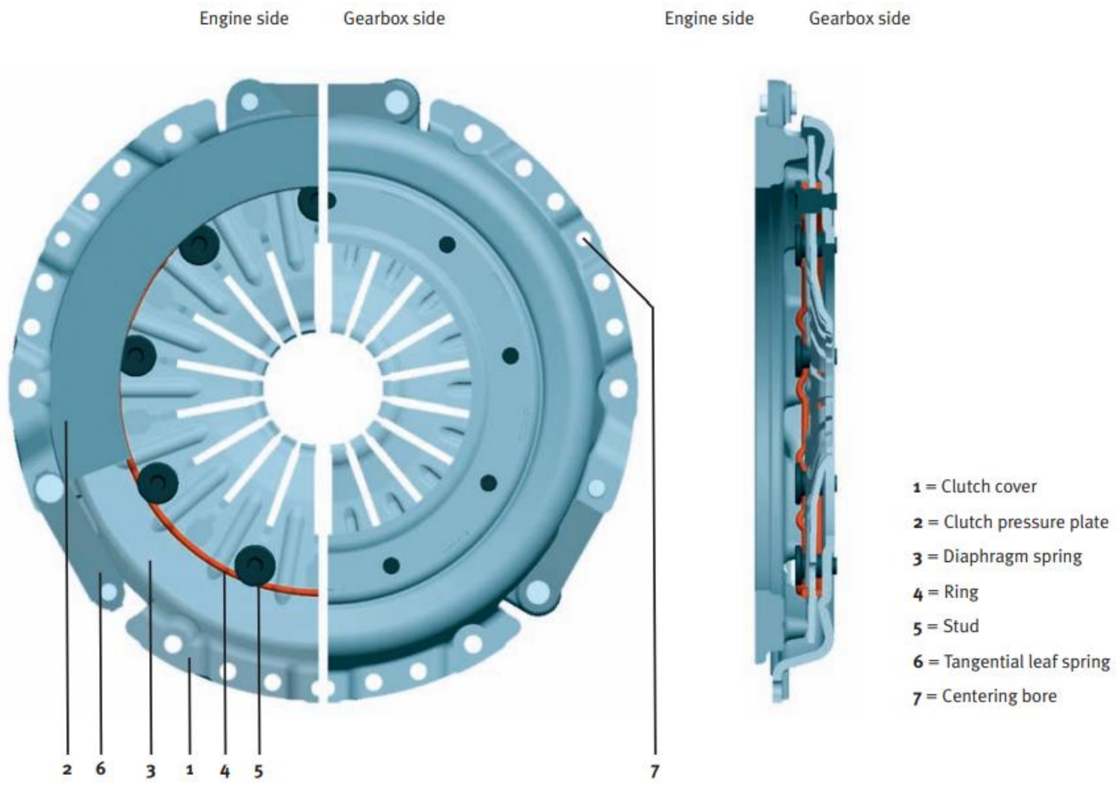


Figure 5 Diaphragm Clutch Cut-section

2.1.4 Multiplate Clutch

Several annular discs are spline-coupled to the inner and outer housing in a multi-plate clutch system. Every alternative disc has friction material on one or both sides, and these discs are referred to as friction plates. Figure 6 depicts the clutch plates in their clutch housing layout. In a wet clutch, contact always occurs between the friction material and the plates' direct metal contact. An axial force is applied via the discs to engage the pack, bringing the speed differential to zero. Multi-plate clutches are used to disengage the clutch and facilitate gear changes in gearboxes, and it is desirable to have fast gear changes to aid in better acceleration. So, the time of a clutch pack engagement is short, usually under one second, during which time a large amount of torque is transferred until both surfaces are rotating at the same speed. A disc pack containing several discs is used because this gives a large surface area and can therefore transmit a greater torque.

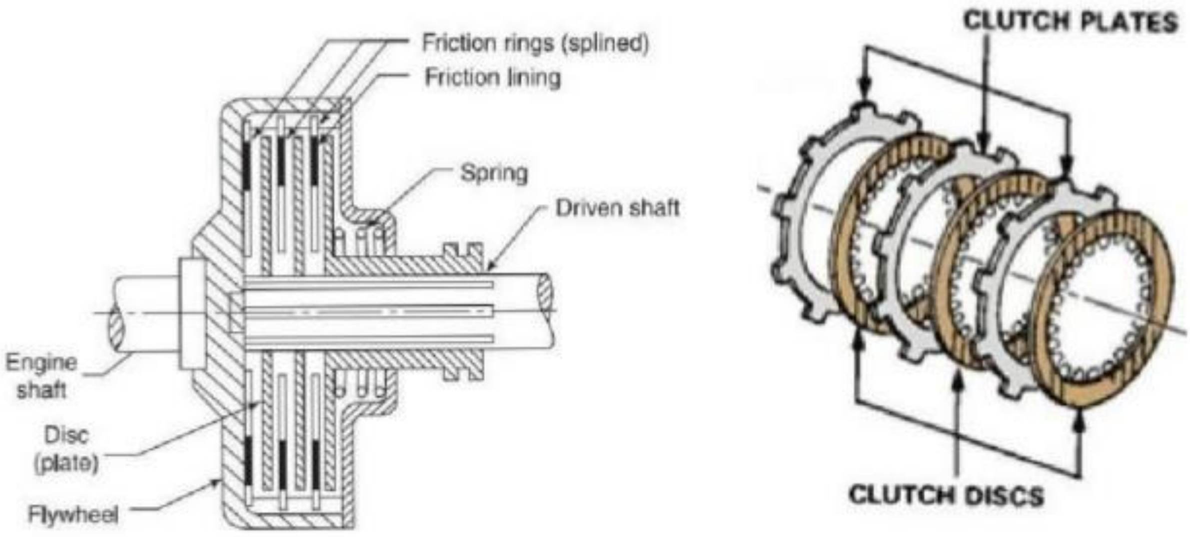


Figure 6 Multiplate Clutch

2.1.5 Working of a Multiplate Clutch

This type of clutch has multiple clutch plates that are used to transmit power from the engine to the transmission shaft of the vehicle. In a car, disengagement occurs between the engine and gearbox by applying force over the clutch due to the force applied on the pressure plate by the clutch peddle. This disengages the pressure plate from the flywheel and releasing the clutch pedal will re-engage the clutch. This being a multiplate system, it uses several clutch plates to transfer the torque instead of the usual single plate present in a single-plate clutch. The number of clutch plates defines the torque capacity of the multi-plate clutch.

2.1.6 Components of a Clutch System

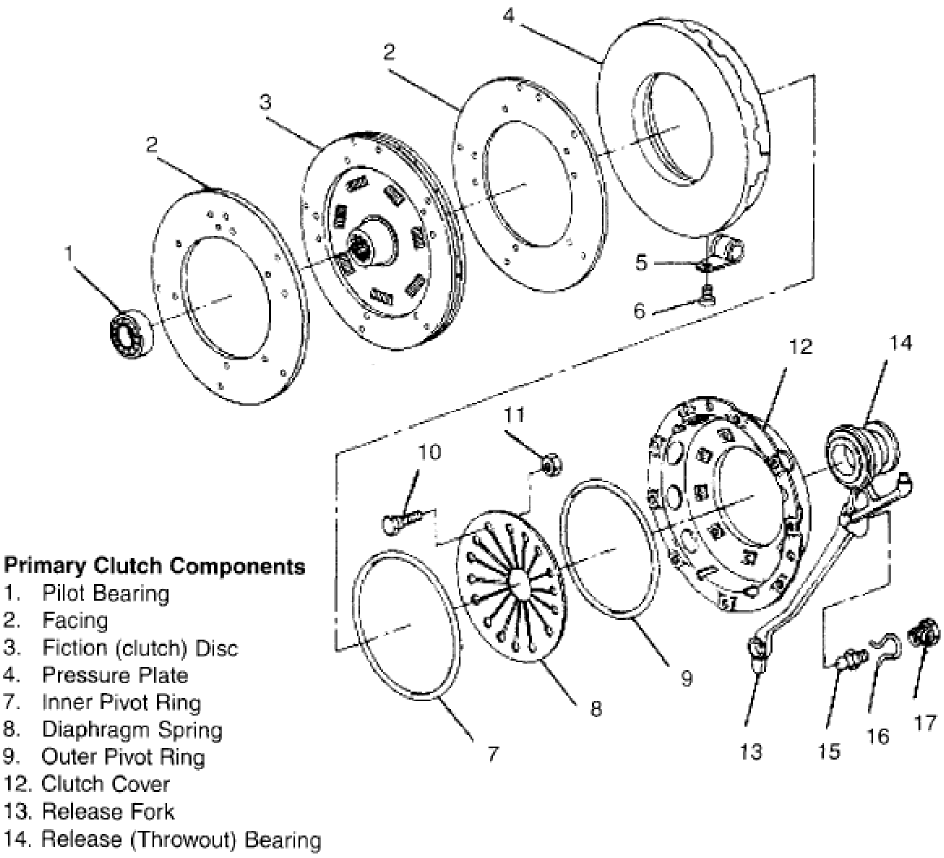


Figure 7 Exploded View of a Clutch System

Pilot Bearing

The clutch pilot bearing is used to support and centre the clutch disc and the transmission shaft, and it allows the transmission shaft to spin at a different speed to the engine speed when the clutch is disengaged from the flywheel. The pilot bearing can be seen in Figure 7 part number 1.

Flywheel

The function of the flywheel is to store and pass on the kinetic energy provided by the internal combustion engine. The flywheel is utilised primarily to smooth out temporary deviations in load and power, to achieve higher power peaks and to bridge power interruptions. This is necessary because of the working principle of the piston engine, which delivers uneven torque and speed [9]. Flywheel is the contact linkage connecting the engine crankshaft to the transmission shaft through the clutch pressure plates. The engage and disengage function of the pressure plate dictates the transmission of torque from the engine to the gearbox. The friction plates in the clutch are splined and connect the flywheel to the transmission shaft through the contact region.

Figure 8 Flywheel

Pressure Plate

A plate connected to the splined sleeves is further connected to the fulcrum of the pedal. In this manner, when the clutch pedal is depressed, the sleeves attached to the pedal fulcrum move outward thus driving the pressure plate connected to the splined sleeve. This is a very important component to apply clamping force onto the clutch plates which make contact with the flywheel. Figure 7

Clutch Plate

The metallic plates with frictional surfaces on them are called Clutch Plates. It uses frictional surfaces to make contact with the flywheel to transmit torque between the engine and the transmission. Figure 7

Thrust Springs

These are springs used behind the pressure plate, and the pressure plate uses their stiffness to maintain frictional contact with clutch plates, which supports the clutch engagement. An illustration of the thrust spring is shown in Figure 9.

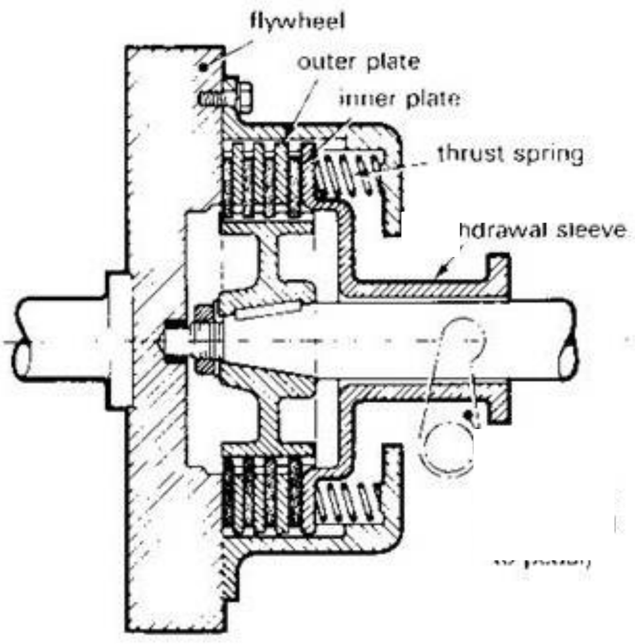


Figure 9 Clutch Thrust Springs Illustration

Diaphragm Spring

The diaphragm is one of the types of spring which is circular and has fingers by which it applies pressure on the pressure plate. Figure 7

Release Bearing

The release bearing is the connecting element between the rotating clutch pressure plate on the engine side and the rigid release mechanism on the transmission side. It is mounted on clutch shaft. When the driver presses the clutch pedal, through a linkage, the release bearing applies pressure on the diaphragm to disengage the clutch.

Figure 10 Clutch Release Bearing

Clutch Release Fork

It is a mechanical linkage used to apply force on the release bearing against the diaphragm spring to disengage the clutch. It has a retraction or a return spring to bring it back to its original position when the clutch pedal is released, and the clutch is re-engaged.

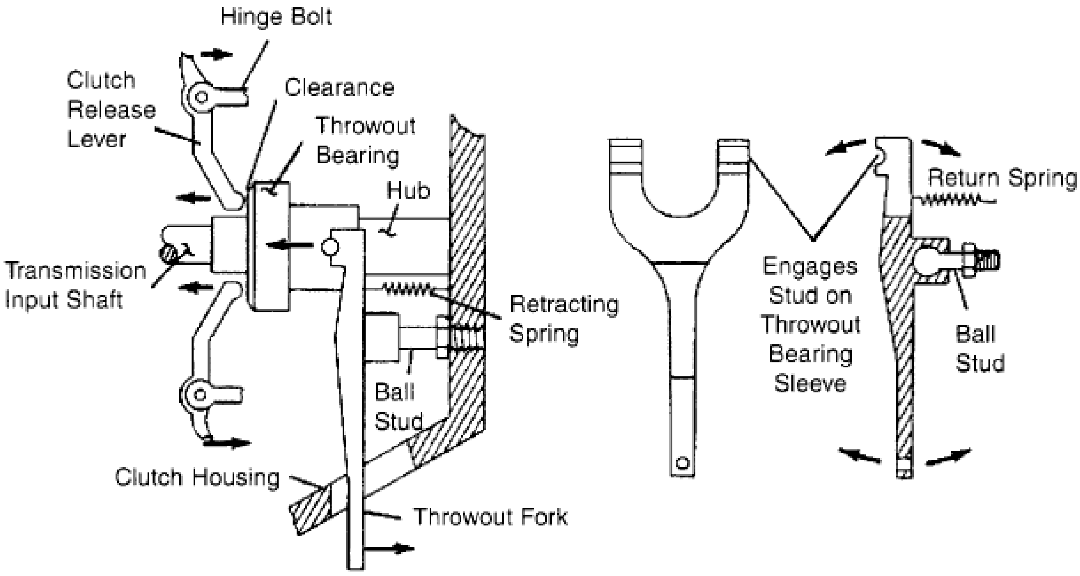


Figure 11 Clutch Release Fork

Clutch Pedal

A clutch pedal is operated by the driver of the vehicle to control the engagement and disengagement of the clutch. This is done by a mechanical linkage between the pedal and the clutch release fork or has hydraulic linkage if a hydraulic central release bearing is being used.

Splined Shaft & Inner Splined Sleeves

All the components of the clutch system are connected to the transmission shaft via the splines present over the shaft and an internal spline present on the hub of the clutch components. The splines allow the transfer of torque from the flywheel through the clutch to the transmission shaft. These splines are illustrated in Figure 12.

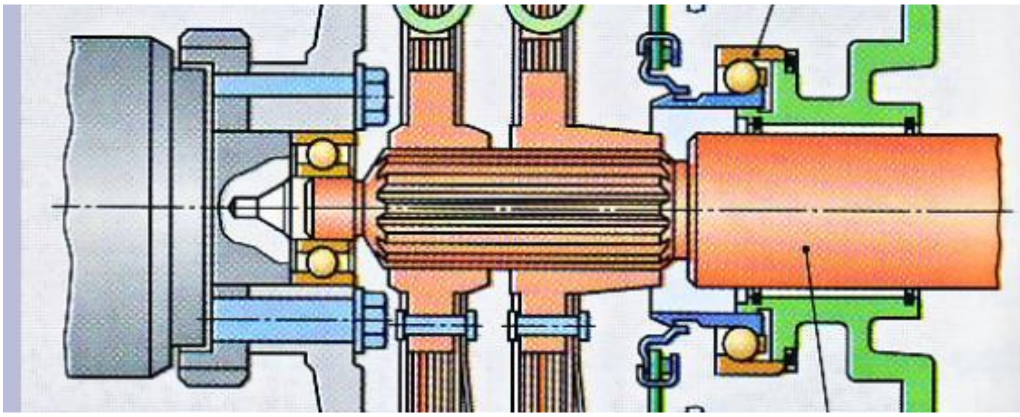


Figure 12 Transmission Shaft inside the Clutch Assembly with splines

2.1.7 Hydraulic Clutch Actuation

The Hydraulic clutch actuation system was developed as a result of increased difficulty in the assembly of cable-actuated (mechanical linkage) clutch systems due to the reduced space and increased complexity of the engine bay. There are two types of hydraulic actuation mechanisms where one replaces the mechanical cable linkage from the pedal and the clutch fork with a master cylinder operated hydraulic lines while the other type is using a central slave cylinder or a central release bearing that operates hydraulically and connects to the master cylinder via hydraulic lines. The hydraulically operated clutch release bearing is a system that improves the clutch actuation mechanism by reducing the number of components involved in sending the clutch pedal input to the release bearing to engage or disengage the clutch. The hydraulic actuation system with mechanical linkage is represented in Figure 13 and the system with a hydraulic central slave cylinder is shown in Figure 14.

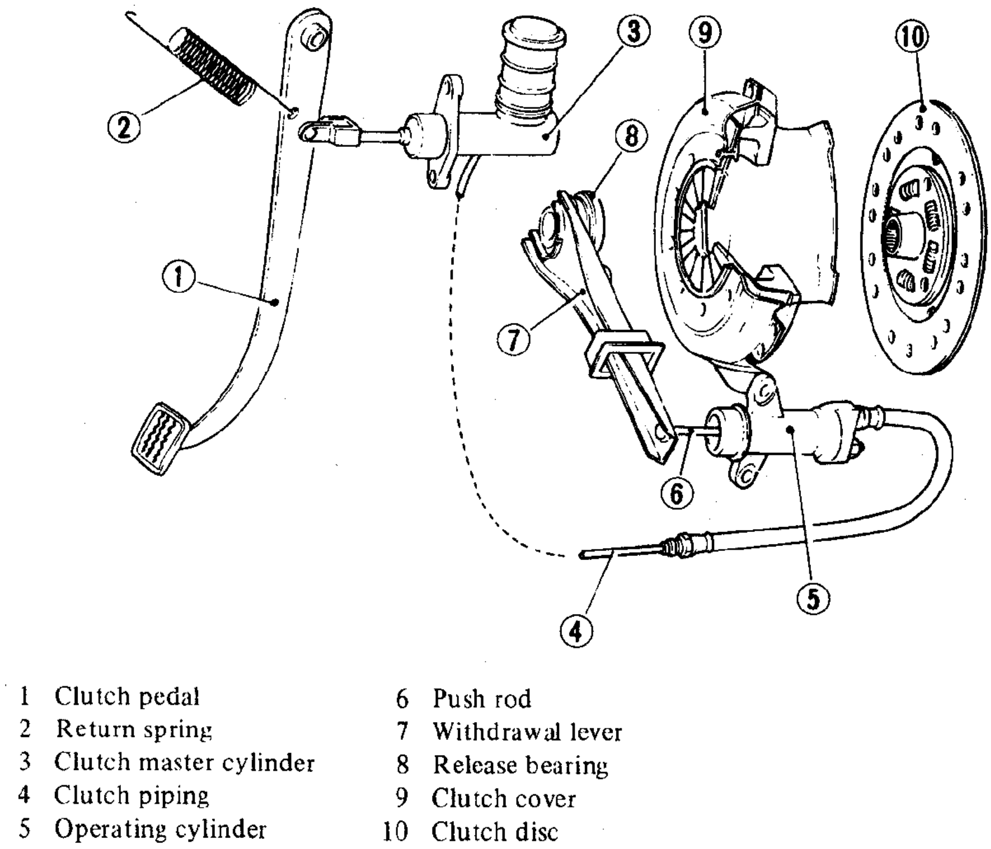


Figure 13 Hydraulic Clutch Actuation Mechanism with Mechanical Linkages

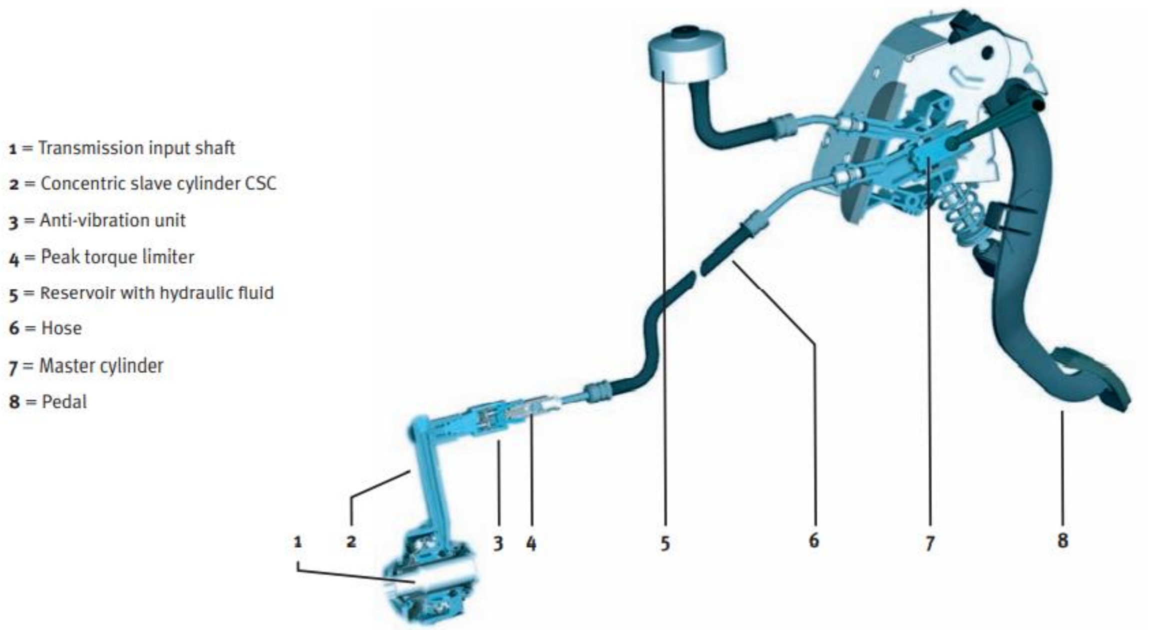


Figure 14 Hydraulic Actuation with Hydraulic Central Slave Cylinder

Hydraulic clutch activation systems consist of a master and a slave cylinder, reservoir and hydraulic lines. When the clutch pedal is depressed, the pushrod contacts the plunger and pushes it up the bore of the master cylinder. During the first few millimetres of movement, the centre valve seal closes the port to the fluid reservoir tank and as the plunger continues to move up the cylinder, the fluid is forced through the hydraulic line to the slave cylinder mounted on the clutch housing. As fluid is pushed from the master cylinder, this, in turn, applies force on the inner side of the piston in the slave cylinder thus moving it outward. The slave cylinder has a piston that rests on the fingers of the diaphragm spring which applies pressure on the diaphragm spring to disengage the pressure plate from the clutch disc. When the pedal is released, the hydraulic pressure to the slave cylinder is reduced and as the force applied by the diaphragm spring on the slave cylinder is higher the piston retracts to the rest state.

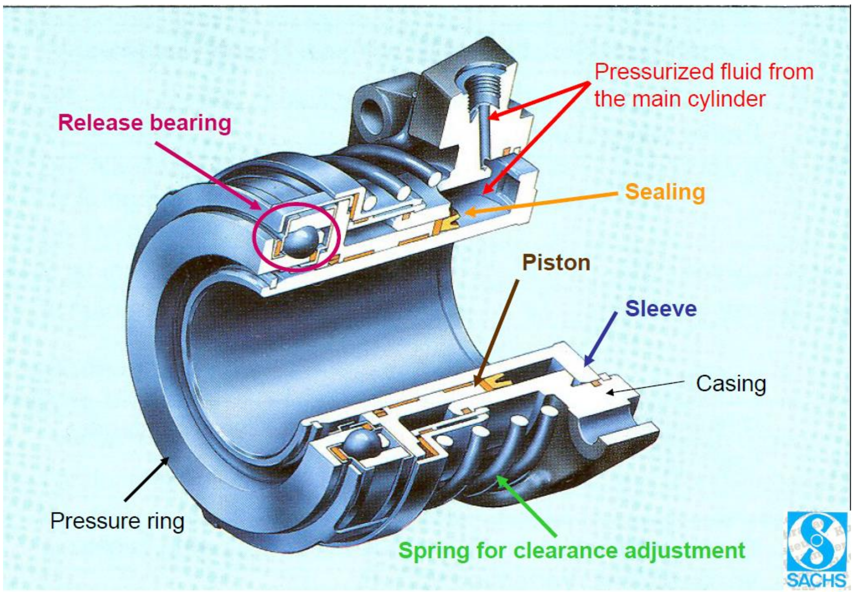


Figure 15 Cut section of Central Release Bearing

2.1.8 Clutch Modelling

Clamp force calculation [2] on a Clutch plate can be calculated as

$$C = R_s \,.\, \mu \,.\, F_A \,.\, n \qquad \text{Eq. 1}$$

$$p = \frac{F_A}{\pi \,.\, (R_e^2 - R_i^2)} \qquad \text{Eq. 2}$$

$$C = C_{eng} \,.\, \beta \qquad \text{Eq. 3}$$

Where,

C = Clutch Torque capacity

C_{eng} = Maximum Engine Torque

β = Safety coefficient

R_s = Averaged Frictional Clutch Radius

F_A = Clamp Force on clutch plates

μ = Friction Coefficient

F_t = Tangential Force

n = Number of Frictional Surface

p = Pressure

S = Surface Area

R_e = External Radius of the Clutch

R_i = Internal Radius of the Clutch

The force on the diaphragm spring can be calculated as

$$F_{clamp} = \frac{4 \,.\, E.\, C_{geom}}{1 - \,v^2} \,.\, \frac{e^3 \,.\, h}{D^2} \qquad \textit{Eq. 4}$$

Where,

D = External Diameter of Diaphragm Spring

E = Modulus of Elasticity

D = Internal Diameter

e = thickness of spring

h = height of spring

δ = deflection of spring

v = Poisson coefficient

C_{geom} = Coefficient of the geometry of diaphragm spring

The Hydraulic Line pressure in the lines near the slave cylinder can be calculated to be

$$A_{rb} = \pi \,.\, (R^2 - r^2) \qquad \textit{Eq. 5}$$

Where,

A_{rb} = Area of the slave cylinder piston

R = External Radius of the slave cylinder piston

r = Internal Radius of the slave cylinder piston

$$R_b = (F_{clamp})\ \frac{A}{B} \qquad \textit{Eq. 6}$$

$$p_{hyd} = \frac{R_b}{A_{rb}} \qquad \textit{Eq. 7}$$

Where,

R_b = Force on the release bearing

A = Distance from Cover fulcrum to Pressure fulcrum

B = Distance from Cover fulcrum to spring finger bearing contact point

p_{hyd} = Hydraulic Line pressure

Using the hydraulic lines pressure the pedal force can be calculated with

$$A_m = \pi . \mathrm{r}_p^2 \qquad \textit{Eq. 8}$$

Where,

A_m = Area of the piston

r_p = Radius of the master cylinder piston

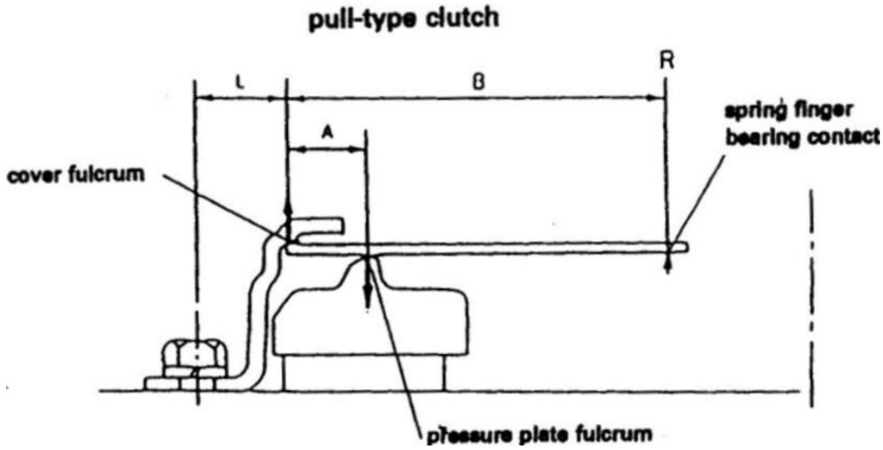


Figure 16 Force Equilibrium for Pull-type Clutch

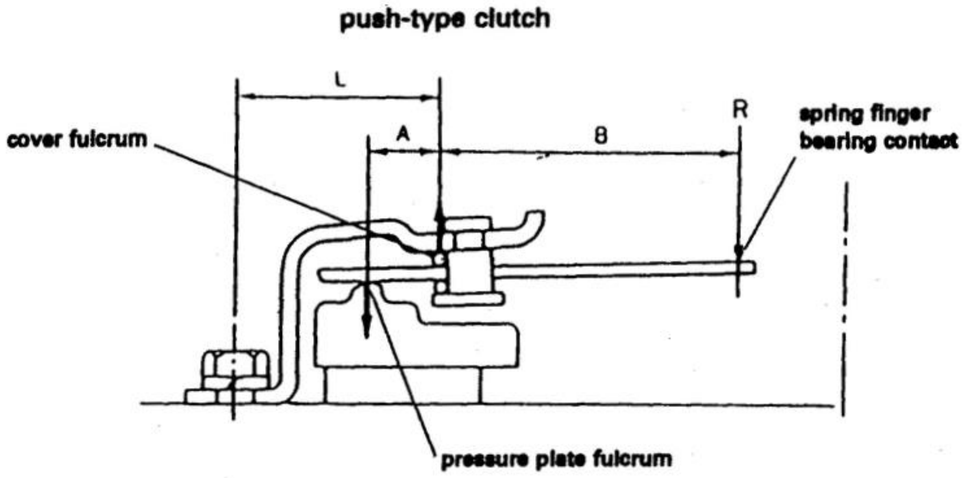


Figure 17 Force Equilibrium for Push-type Clutch

The pedal force can be calculated with

$$F_p = A_m \cdot p_{hyd} \qquad \textit{Eq. 9}$$

Where,

A_m = area of the master cylinder piston

p_{hyd} = Hydraulic Line pressure

F_p = Pedal Force

The pneumatic piston bore calculation can be done with

$$\emptyset = \sqrt{{4.F_p}/{\pi \,.\, P_n}} \qquad \textit{Eq. 10}$$

Where,

Φ = Bore diameter of the pneumatic piston
F_p = Pedal Force
P_n = Operating pneumatic pressure

2.2 Hybrid Vehicles

A Hybrid Vehicle is defined as a vehicle that is equipped with at least two energy storage units (batteries, fuel tank, etc.,) and with the corresponding power units (electric motors, internal combustion engine, etc.,) that these energies transform into traction work.

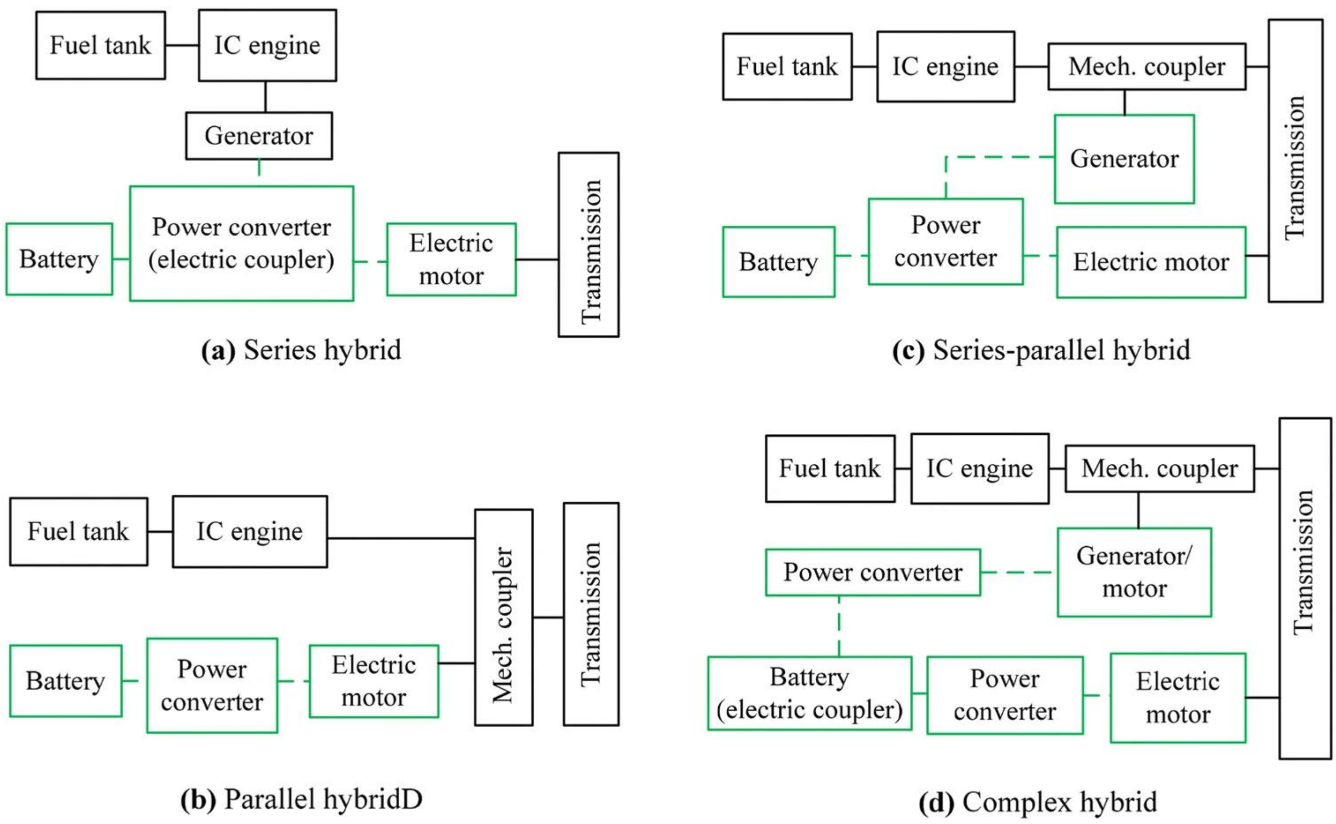


Figure 18 Architecture of Hybrid Electric Vehicles [22]

Due to the rise in global oil depletion and an increase in emissions across the globe, newer legislations are being put into place for the use of a vehicle that can run on alternative sources of power instead of conventional IC engines and arrest the extent of emissions from the vehicles on road. This led the automotive manufacturers to come up with Hybrid electric vehicles in various configurations based on the applications and size of the vehicle. With that, the Series, Series-parallel and the Parallel Hybrid concepts have been well developed and are being used widely in the current market to develop newer vehicles to comply with the norms while offering high mileage compared to conventional IC engine vehicles.

2.2.1 Series Hybrid

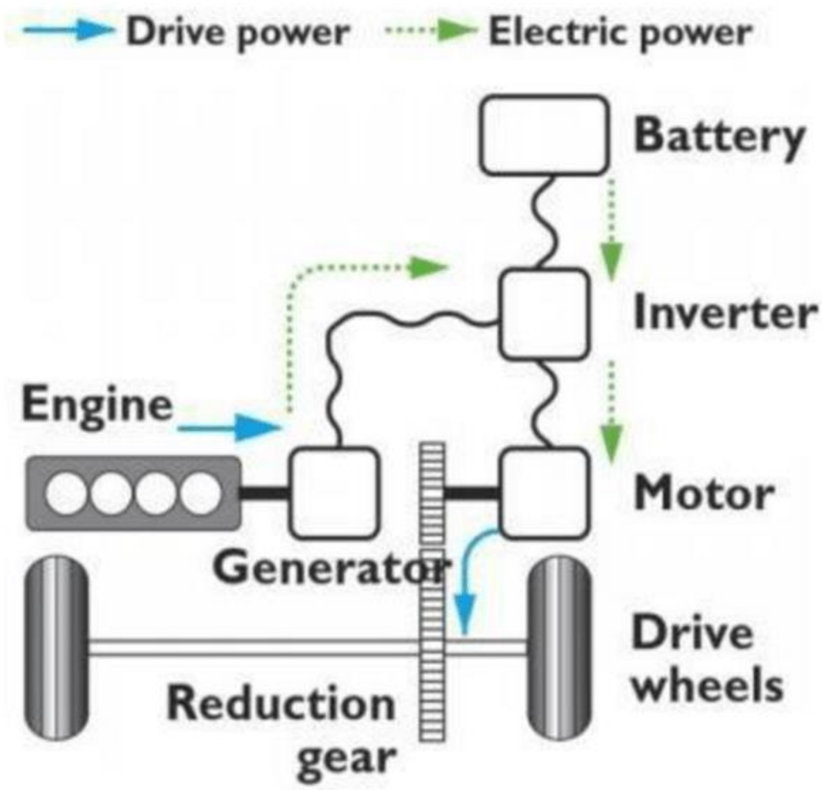


Figure 19 Series Hybrid Electric Vehicle Configuration [6]

A Series Hybrid Electric Vehicle consists of the ICE, generator, power converter, Electric motor, and battery, as represented in Figure 19. There is no mechanical connection between the ICE and the transmission, and the ICE's function is to run at a designed optimum speed to generate power that is stored in the batteries which are then consumed by the electric motor to drive the vehicle. The engine's operational speed is independent of the vehicle's operational speed. The engine speed is controlled by a controller based on the State of Charge of the batteries and the requirements of the Electric Motor or the Traction motor. If there is a very high or sudden requirement of charge from the electric motor, then the generated power is bypassed from the generator to the motor through the inverter. During the braking conditions or moving downhill the electric motor can act as a generator to generate electricity which goes through the inverter and is then stored in the batteries. This type of configuration needs to have a well-equipped electric motor as it is the only mechanical linkage to drive the vehicle and needs to perform under all road conditions and gradient changes.

2.2.2 Series-Parallel Hybrid

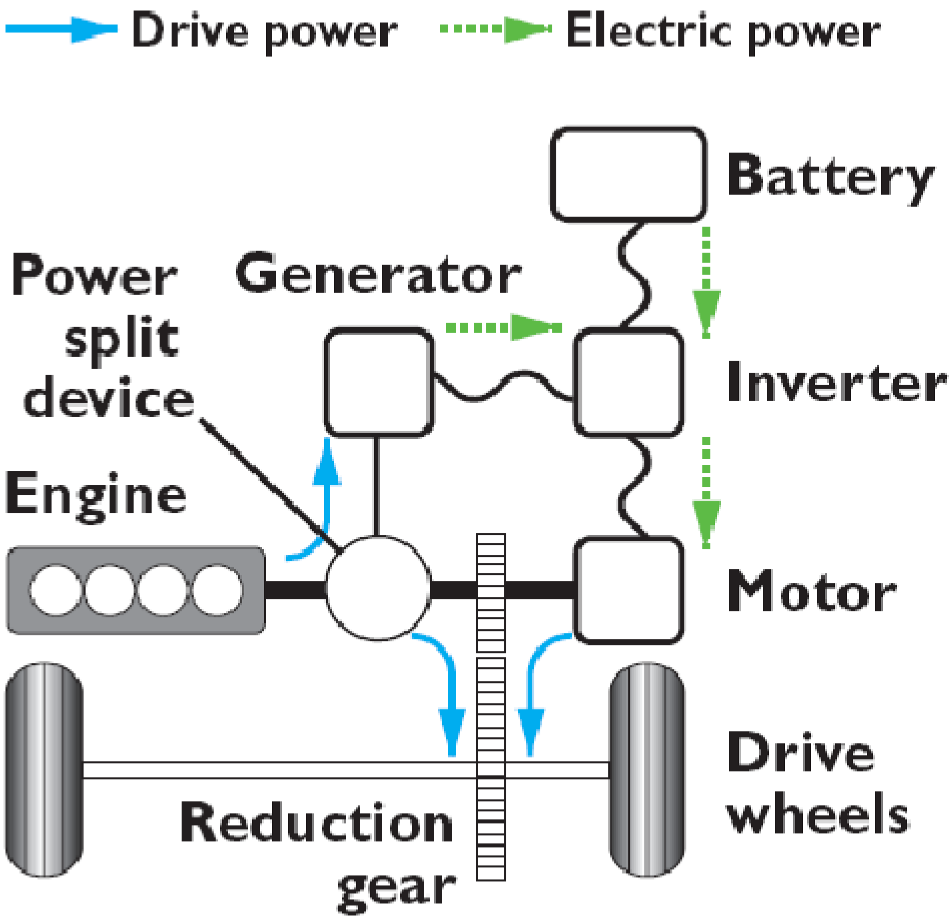


Figure 20 Series-Parallel Hybrid Electric Vehicle Configuration [6]

A Series-Parallel Hybrid Vehicle is a hybrid vehicle configuration, Figure 20, in which the power from the ICE and motor is transmitted to the wheels through a common mechanical linkage with the ICE and motor output in series. The ICE is also connected to a generator unit to generate electricity which is stored in the batteries for use with the Electric motor. In this HEV configuration, the driver can choose the drive mode to be ICE or EV. There is an intelligent controller that decides the power split between the drive and the electricity generation to aid drivability and have the batteries charged at the same time. The electric motor is used to perform regeneration functionality while braking. This type of a system is more complex compared to a series hybrid system mainly due to the additional mechanical linkages and the control strategies required to handle the power output and keep a steady power generation to keep the batteries charged.

2.2.3 Parallel Hybrid

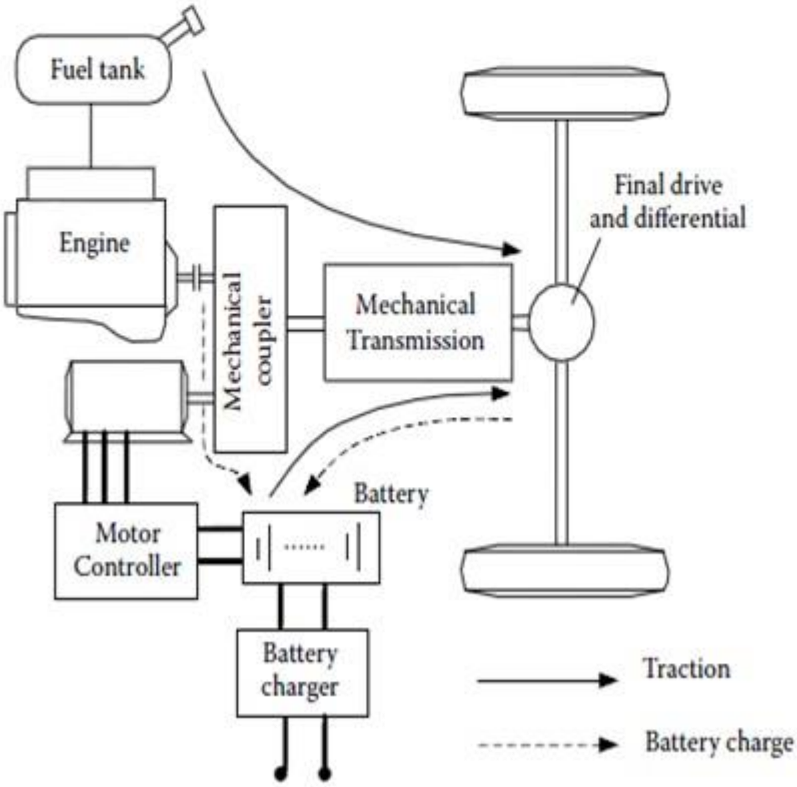


Figure 21 Parallel Hybrid Electric Vehicle Configuration 1

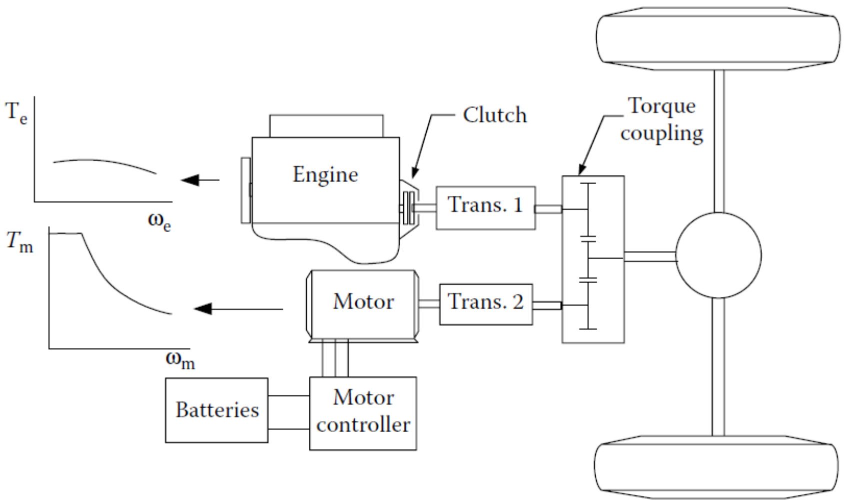


Figure 22 Parallel Hybrid Electric Vehicle Configuration 2

A Parallel Hybrid Vehicle is an HEV configuration in which the ICE and the Electric Motor have a mechanical coupler between them and the gearbox, as shown in Figure 21, or a secondary configuration in which the Motor and ICE have a separate set of gearboxes for each of them and their outputs are coupled and then sent as an output to the wheels, as shown in

Figure 22. In this HEV configuration, the ICE and the Electric motor can both deliver power parallelly which is then transmitted to the wheels through a mechanical gearbox connecting the two power sources. In some configurations, the ICE has several sets of gears while the electric motor has only one reduction and in other scenarios, as represented in Figure 22 they have a few gears, and this would be able to provide a wider number of gear pairs to have a varying range of torque outputs. This parallel hybrid configuration can also be implemented to individual axles where the front axle can be driven by an ICE while the rear is driven by the Electric Motor and both of them do not have a direct mechanical linkage between the two and only have a controller that decides the power output from each power source. The other main difference between series hybrid and series-parallel configurations is that the ICE cannot be used as a power generation source as there is no generator on board and the way to charge the batteries is through a charging outlet or electric motor regeneration on the go.

2.3 Pneumatics

2.3.1 Pneumatic Cylinders

Pneumatic Cylinders are mechanical cylinders that use a compressed gas supply to transmit force in a linear and reciprocating motion. [20]

The main components of a pneumatic cylinder constitute a Cap-end port (A), Tie rod (B), Rod-end port (C), Piston (D), Barrel (E), and Piston rod (F). Figure 10.

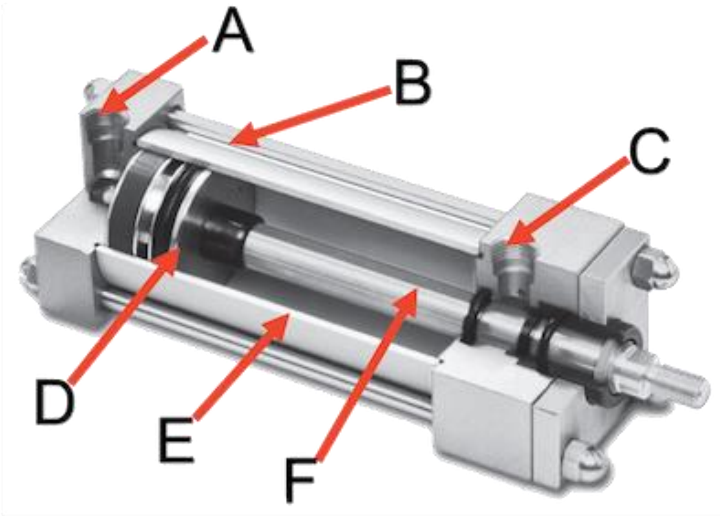


Figure 23 Pneumatic Cylinder Design

The cylinder barrel is sealed on both ends with a head cap and an end cap. Inside the cylinder, a piston linearly drives the rod. When compressed air enters through the cap-end port, the piston way starts to move towards the rod end port while pushing the rod out. This movement is called the extension and the chamber associated with this movement is called the plus chamber. When the compressed air enters from the rod end port and pushes the rod back and this movement is called retraction.

Figure 24 Pneumatic Piston Motion

The cylinder diameter is the diameter of the piston. The stroke length identifies how far the piston can travel. The diameter and stroke of a pneumatic cylinder are two essential attributes by which it is identified and chosen for various applications.

Pneumatic cylinders can be classified into single-acting and double-acting pneumatic cylinders.

2.3.1.1 Single-acting

In a single-acting cylinder (SAC), the air can only be supplied from one side of the piston which causes the piston to move. The movement of the piston in the opposite direction is a result of the decompression of a mechanical spring. A single-acting cylinder is available in a configuration with the out-stroke or in-stroke by varying the position of the spring to either in front or behind the piston. In case of pressure or power loss to the controlling valve, a single-acting cylinder returns the piston to a base position either extended or retracted based on the cylinder configuration.

A disadvantage of a single-acting cylinder is the inconsistent output force due to the spring creating a force which opposes the piston and thus needs more input pressure to do the same amount of work in a cylinder without the spring. The construction length of a single-acting cylinder is longer than the actual stroke.

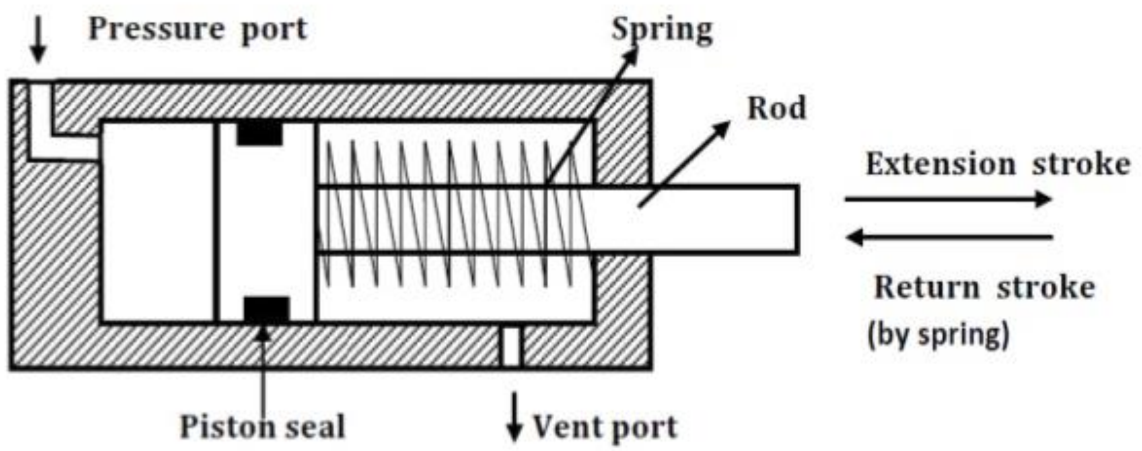


Figure 25 Single-acting Cylinder

2.3.1.2 Double-acting

In a double-acting cylinder (DAC), the air is supplied to chambers on both sides of the piston. Higher air pressure on one side of the piston drives it to the other side. Double-acting cylinders are the most common type, as they give complete control of the stroke time, perform functions like actuation modulation, and enable very precise control over the actuation functionality. The disadvantage of double-acting cylinders is their need for compressed air for movement in both directions and a lack of return to base position functionality like the single-acting magnetic cylinders in case of a power or pressure failure.

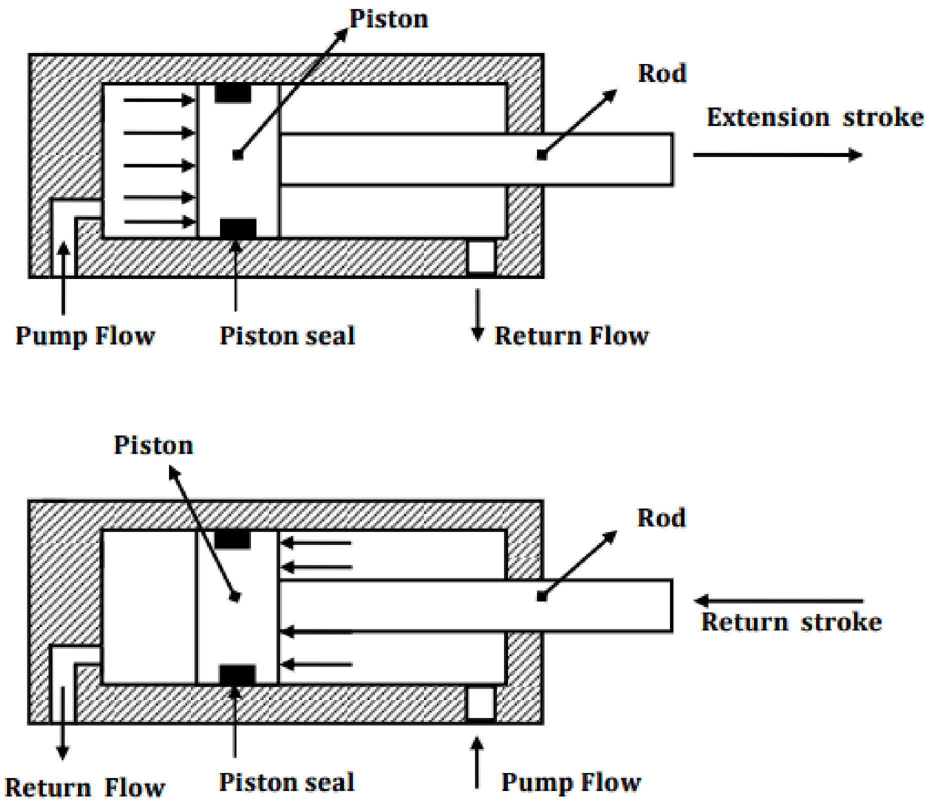


Figure 26 Double-acting Cylinder

2.3.1.3 Position Sensing for Pneumatic Cylinders

The pneumatic cylinders have a magnetic base attached to the piston which is used to get feedback for the position sensing of the piston which can thereby be used for precise control of actuation. These types of cylinders usually are marked with the nomenclature “magnetic” at the end of the name. For the sensing of the position, Magnetic Position Sensors are used which sense the magnetic field of the magnet on the cylinder piston to give an analogue output signal either in the form of 0 – 10 V DC or 4 -20 mA or a digital signal with a standardized communication protocol used mainly in industrial PLC’s. Reed switches and hall effect sensors are the most used sensor types.

Figure 27 Pneumatic piston with a Magnetic Position Sensor

The reed switch is a magnetically controlled switch with two magnetizable tongues or reeds sealed in an inert gas housing. When a magnet is brought closer to this sensor the magnetic field causes the two tongues of the sensor to get closer and closer to complete the circuit and allow the flow of current and this current is taken as an analogue input to read the position of the magnet in space. There would be variation in the current output based on the gap between the two tongues of the sensor and this dictates the positioning of a magnet in the measurable range of the sensor. The effect of the magnetic field of an external source which is used by the reed switch can be seen in Figure 28.

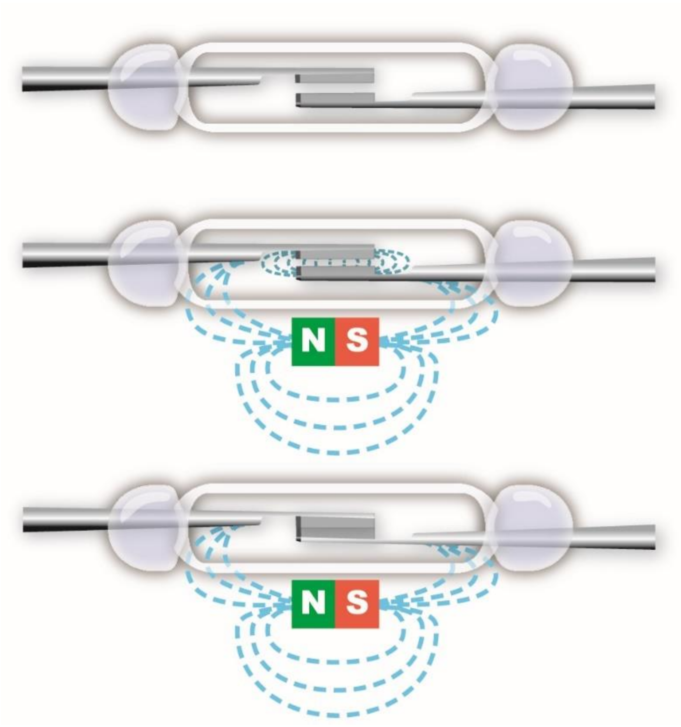


Figure 28 Reed Switch Working Stages

2.3.1.4 Reading the Pneumatic Cylinders

The standards for the representation of pneumatic cylinders have been set by ISO and the representation is independent of the diameter, stroke, and subset ISO standards followed by various manufacturers. [25]

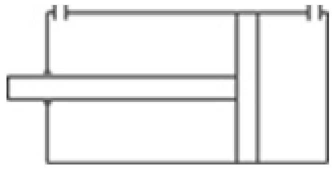
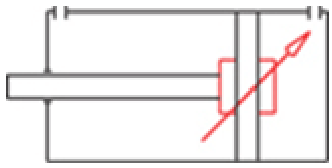
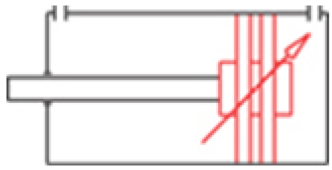
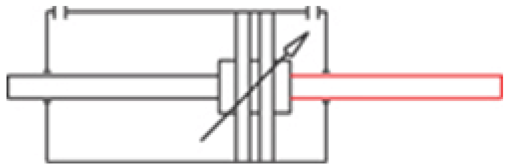
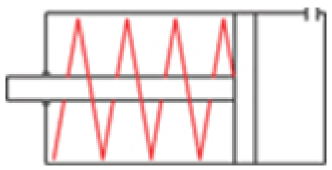
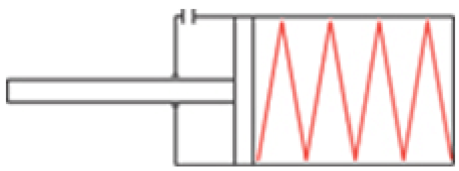
Type of Cylinder	*Representation*	*Explanation*
Double-acting cylinder		*Standard design*
Double-acting cylinder with magnetic piston		*the piston is different from figure 1, indicating the magnetic piston*
Double-acting cylinder with adjustable cushioning		*Cushioning is symbolized by two rectangular objects: adjustable symbolized by an arrow*
Double-acting cylinder with adjustable cushioning and magnetic piston		*Combination of Figures 2 and 3*
Double-acting cylinder with through piston rod, adjustable cushioning, and magnetic piston		*The through piston rod is added*
Single-acting cylinder (minus)		*Single-acting cylinder with spring in minus chamber*
Single-acting cylinder (plus)		*Single-acting cylinder with spring in plus chamber*

Table 1 ISO standardized representation of Pneumatic Cylinders

2.3.2 Pneumatics and Solenoid Valves

A pneumatic solenoid valve also called the Directional Control Valve, is an electrically operated valve that uses electromagnetic force to operate a valve core or a plunger made of ferrous material. When an electrical current is passed through the solenoid coil, a magnetic field is generated, and it causes an internal metal rod to move. This is the working principle by which the valve opens, and it either works with or without pneumatic assistance depending on the construction of the solenoid plunger and the minimum required differential pressure along with the energized coil to move the plunger along its path to open or close the pneumatic circuit. This principle of working can be seen in Figure 29. The usual representation of the Inlet port is P or 1, the Outlet Port (source to a system) is A or 2 and the Exhaust Port is T/R or 3. In the case of more than 1 exhaust and outlet port, other alphabets and numbers are used to represent the ports. The symbols on the solenoid valves represent the type of solenoid valve, the actuation mechanism, and the flow direction with the help of arrows on the body of the solenoid valve. [26]

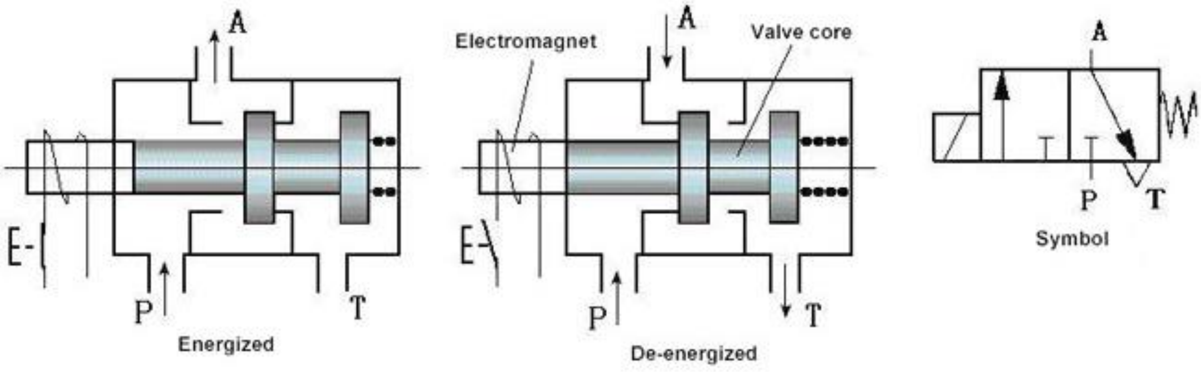


Figure 29 Working states of Pneumatic Solenoid Valve

Figure 30 Directional Control Valve Cut-Section

The pneumatic solenoid valves work under a wide range of input voltages but the most common is the 230 V AC and 24 V DC solenoid coil types. For smaller load applications the 12 V DC versions are also available. The solenoids have a switching frequency between 5 ms to 50 ms which indicated the minimum required time to start the actuation once the coil is energized.

2.3.2.1 Classification of Pneumatic Solenoid Valves

The Solenoid Valves are usually classified into two types depending on the resting state of the plunger that directs the airflow in the solenoid from one side to the other. These types are namely Normally Open (NO) and Normally Closed (NC). [26]

In a **Normally Open (NO)** type solenoid valve, the valve remains open when the solenoid is not energized which means that the air can flow from the input port to the output port without any restriction and this is usually used in cases where the pneumatic actuation is required only to stop the flow of air to execute an actuation and where the system always need airflow to continue operating as intended. As represented in Figure 31, the connection is always made between Port 1 and 2 causing the air to continually flow from the inlet to the outport.

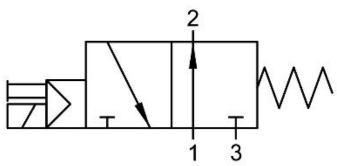


Figure 31 Normally Open Solenoid Valve

In a **Normally Closed (NC)** type solenoid valve, the valve remains closed when the solenoid is not energized which means that the air only enters the system when the connection is made between the input and output port, and in the rest case the solenoid plunger closes the path of airflow from the source to the system. This type of solenoid is usually used when the actuation mechanism is required to operate occasionally, and the system doesn’t require a continuous input of air for its regular operation. As represented in Figure 32, the path is opened between the system and the exhaust port 3 indicating an outflow of air from the system and as the system usually doesn’t have a secondary source of air to operate on this allows any air to flow out of the system.

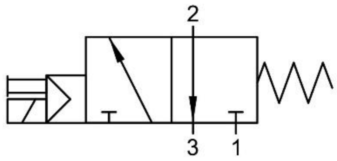


Figure 32 Normally Closed Solenoid Valve

The Solenoid Valves are usually classified into two types depending on the number of coils that actuate the solenoid plunger. They are Mono-stable and Bi-stable solenoid valves.

In a **Mono-stable** type solenoid valve, the coil that operates the plunger is placed on one side of the solenoid valve and there is a spring on the other end of the plunger, as represented in Figure 33, so when the coil is energized it moves the plunger and the spring compresses and when the coil is de-energized the spring decompresses to move the plunger back to its initial position. The main advantage of using a mono-stable type solenoid valve is that in case of a power loss to the solenoid valve, the spring helps to bring the plunger back to its initial state thereby preventing damage to the system by allowing a continuous flow of air into the system.

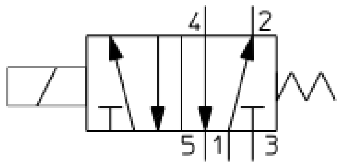


Figure 33 Mono-stable Pneumatic Solenoid Valve

In a **Bi-stable** type solenoid valve, the coil that operates the plunger is placed on both sides of the solenoid valve, as represented in Figure 34, and to move the plunger to open or close the circuit either of the coils needs to be energized. The advantage of this type of solenoid is to achieve precise control of the plunger to vary the operating frequency and to streamline the airflow into the system by varying the actuation speed of the plunger with the help of supply current to either of the solenoids to move and oppose the plunger motion. The disadvantage of this type of solenoid is that in case of a power loss, the plunger will not be able to return to its initial state as there is no spring to move the plunger.

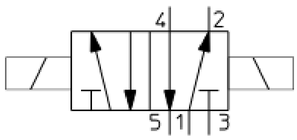


Figure 34 Bi-stable Pneumatic Solenoid Valve

The Pneumatic Solenoid Valves are usually classified into three types depending on the number of ports that are present in the valve. They are 2-way, 3-way, and 5-way pneumatic solenoids.

2-way Pneumatic Solenoid Valve

The 2-way solenoid valves are the most basic types of solenoid valves that have one inlet port and one outlet port. The 2-way nomenclature indicates that the air can flow in either direction depending on the type of actuation, intake, or exhaust. These are usually used in controlling the pneumatic lines to either turn them on or turn them off. Figure 35 shows a representation of 2/2-way solenoid valves with 2 states for the flow of air to or from the system. These solenoids have the option to have a normally closed and normally open circuit. The drawback with this type of solenoid is that one solenoid can only do one action, either exhausting or allowing the inflow of air.

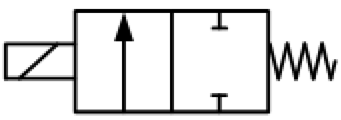


Figure 35 2/2-way Solenoid Valve

3-way Pneumatic Solenoid Valve

The 3-way solenoid valves are a type of solenoid valve that have 3 ports that help to direct the flow of air across the solenoid to and from the system. With a 3-way solenoid valve, it is possible to have both inlet and exhaust functionalities in one solenoid, and thus it is represented by 3/2-way where the 2 indicates the number of possible states. Figure 36 represents the 3/2-way mono-stable type solenoid valve. There is a possibility to have the 3/2-

way solenoid with a bi-stable type coil. These types are solenoids are usually used to control pneumatic cylinders which need to have an intake and exhaust to control the motion of the piston, and these are mainly used for single-acting cylinders as they only have one port for both intake and exhaust.

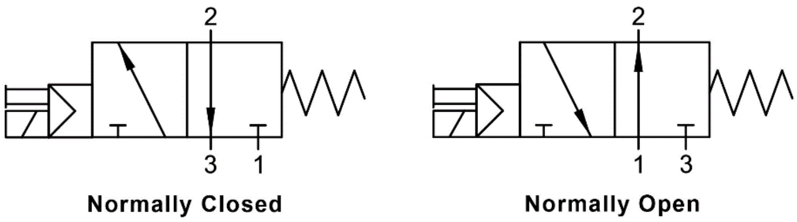


Figure 36 3/2-way Pneumatic Solenoid Valve

5-way Pneumatic Solenoid Valve

The 5-way solenoid valves are the type of solenoids that have 5 ports which comprise 1 intake, 2 outlets, and 2 exhaust ports. These types of solenoid valves are usually used for double-acting pneumatic cylinders or any other equipment which has 2 ports through which the air can be sent into the system and exhausted at the same time as represented in Figure 37.

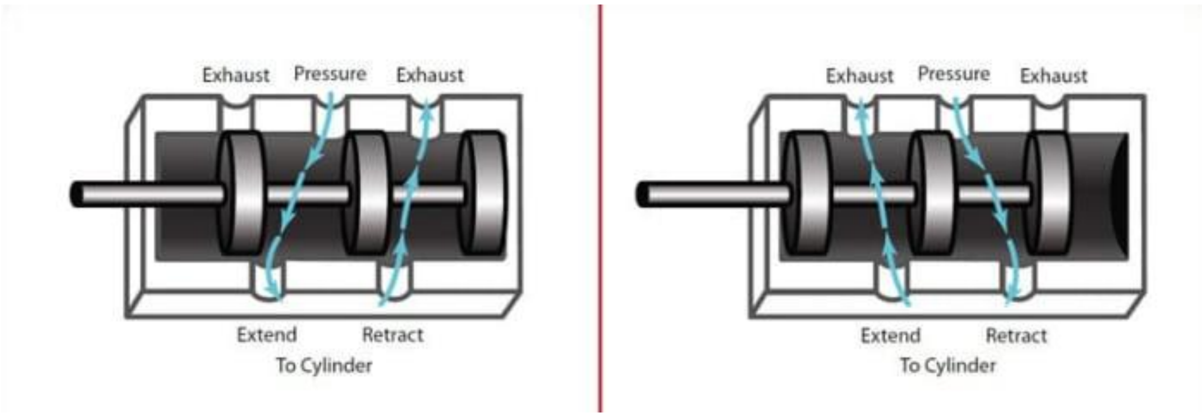


Figure 37 5/2-way Solenoid Valve Airflow Configuration

Depending on the number of ports that are active at a given time the 5-way type valves are classified into 5/2-way and 5/3-way.

5/2-Way Valve

The 5/2-way pneumatic solenoid valve has 5 ports and 2 states. It has 1 pressure port, 2 ports that supply air to the system that is to be controlled, and 2 exhaust ports. The two states of the valve are:

- Pressure port (P,1) connects to port (A,2), while port (B,4) vents through the exhaust port (EB,5)
- Pressure port (P,1) connects to port (B,4), while port (A,2) vents through port (EA,3).

The 5/2-way pneumatic solenoid valve can be configured in both mono-stable and bi-stable configurations to best suit the needs of the actuation required for the pneumatic system. These two configurations are represented in Figure 38 and Figure 39 respectively.

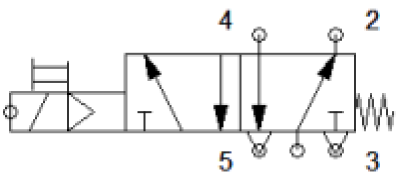


Figure 38 5/2-way Monostable valve

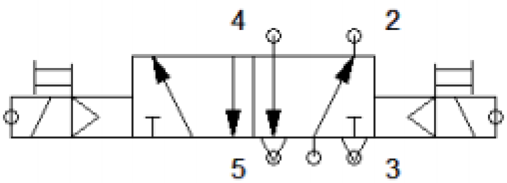


Figure 39 5/2-way Bistable Valve

5/3-way Pneumatic Solenoid Valves

The 5/3-way pneumatic solenoid valves, like the 5/2-way pneumatic solenoid valves, are used to power double-acting pneumatic cylinders. These pneumatic solenoid valves are used for any air-driven device that requires air to be supplied to alternate ports for the device to cycle between various stages of actuation to have precise control over the actuation.

The 5/3-way pneumatic solenoid valves are designed to allow compressed air to flow to one port of a double-acting air actuator while simultaneously allowing air to exhaust from the other port on the same air actuator at the same time.

In a 5/3-way pneumatic solenoid valve, the internal spool can be shifted to a centre position. The typical spool movement is end-to-end inside the valve. With a 2-position valve, the spool shifts from one end to the other end. In a 3-position valve, the spool can be positioned to stop in the middle to accomplish a specific actuation functionality.

The 5/3 pneumatic solenoid valves are usually designed to have two internal spring actuators that, when the valve is not being operated, shift that valve spool to the centre position automatically. This is particularly useful when there is a power loss as it would return to the centre and provide a particular functionality that is desirable for the safety of the system or to use this to perform a specific task when neither of the two solenoids is actuated.

The centre position is classified into 3 separate categories based on the functionality that is achieved by moving the spool to the central position. They are as follows

- Blocked Centre
- Open Centre
- Pressure Centre

Blocked Centre

In this position, all the pneumatic solenoid valve ports are blocked. Air cannot flow through the valve to either of the two actuator ports and from the actuator ports to the exhaust ports as the supply path and the exhaust path is both closed. In this position, since air cannot travel through the valve to the air cylinder or from the cylinder back through the valve when the spool is in the central position and shifts into Blocked Centre, the pneumatic cylinder will not move in either direction. This is represented in Figure 40.

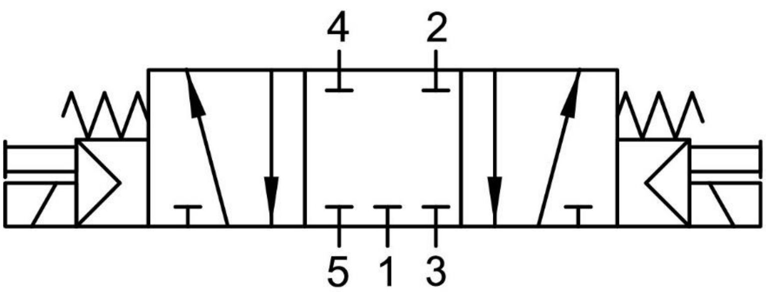


Figure 40 5/3-way Pneumatic Solenoid with Blocked Centre

Open Centre

When the 5/3 pneumatic solenoid valve is shifted into its central position in an “Open Centre” 3-position style valve, the supply line to the valve is blocked, and both cylinder ports are open through the valve to exhaust. With this spool selection, the pneumatic solenoid valve is designed to exhaust air from both sides of the cylinder to keep it in a specific position and this would allow the piston to be moved manually. This configuration is represented in Figure 41.

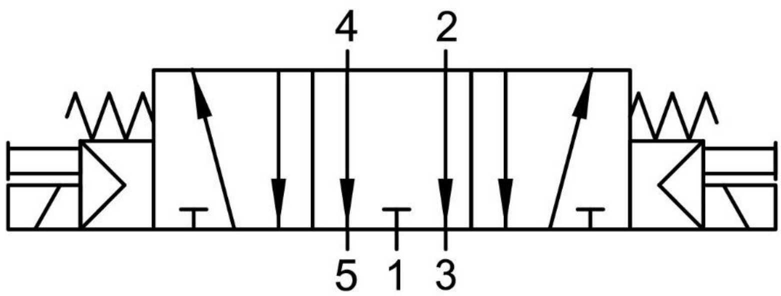


Figure 41 5/3-way Pneumatic Solenoid with Open Centre

Pressure Centre

In the “Pressure Centre” position, air will flow from the supply to both air actuator ports, and the exhaust ports are blocked. This results in the pneumatic piston staying in a fixed position and the continual supply of air prevents the piston from moving in any direction even under external load on the pneumatic piston. This configuration is represented in Figure 42.

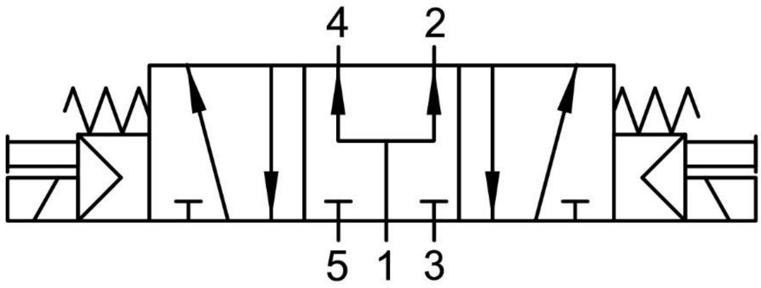


Figure 42 5/3-way Pneumatic Solenoid with Pressure Centre

3 Project:

3.1 Corbellati Missile

The Corbellati Missile is a Hybrid Hypercar with an 1800 HP IC Engine, a LeMans engine developed by Mercury Racing, and a 470 HP Electric Motor, HVH-250-090-DOM from Borg Warner, in a Parallel Hybrid Configuration coupled with a custom-designed gearbox with 4 speeds for ICE mode, 2 speeds for EV mode and 4 speeds for Hybrid mode. The gearbox is designed to have a maximum number of interchangeable gears for ease of manufacturing. The vehicle does not have a reverse gear and it is handled by the Electric motor in all the modes. The gearbox disposition is represented in Figure 43.

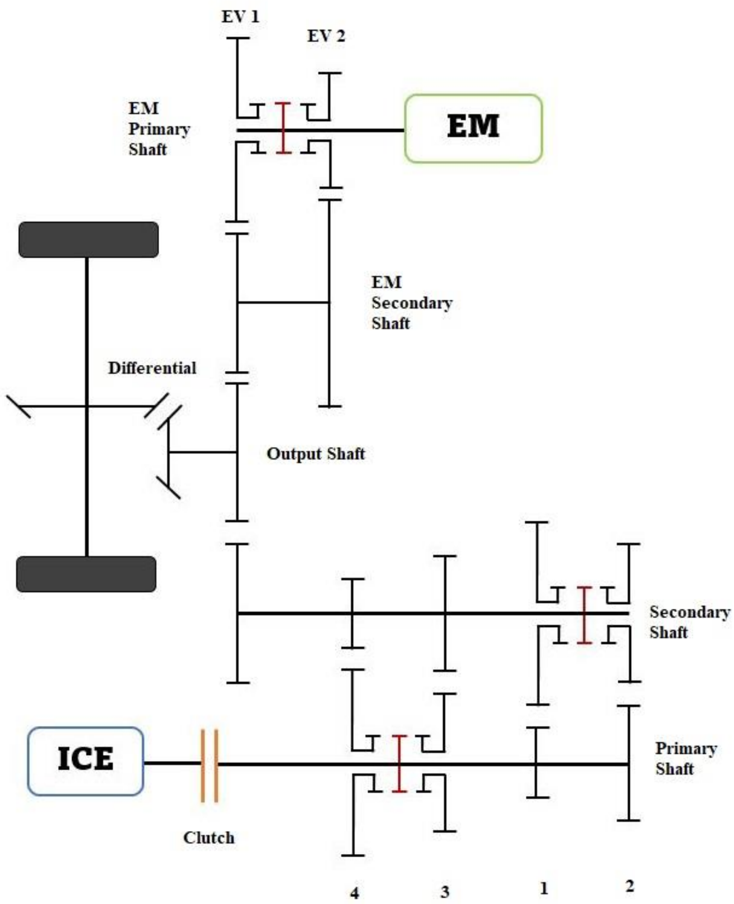


Figure 43 Corbellati Missile Gearbox Disposition

The V8 engine of the Missile produces 2400 Nm of torque and a clutch pack for the same was required. A racing clutch system RCS 184-S2.6-D-S-49, Figure 44, with a 4 Disc double-sided clutch designed to handle 2616 Nm of torque was chosen with special splines for our requirements. This system is equipped with a hydraulic Concentric Slave Cylinder, Figure 45, instead of the usual lever-actuated release bearing which was designed for DTM applications to be able to handle high amounts of release force while being compact.

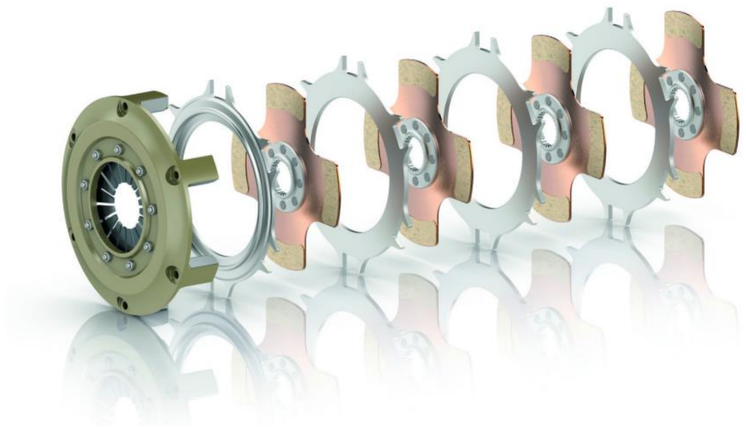


Figure 44 Clutch Pack used in Corbellati Missile

Figure 45 Clutch Release Bearing from SACHS used in Corbellati Missile

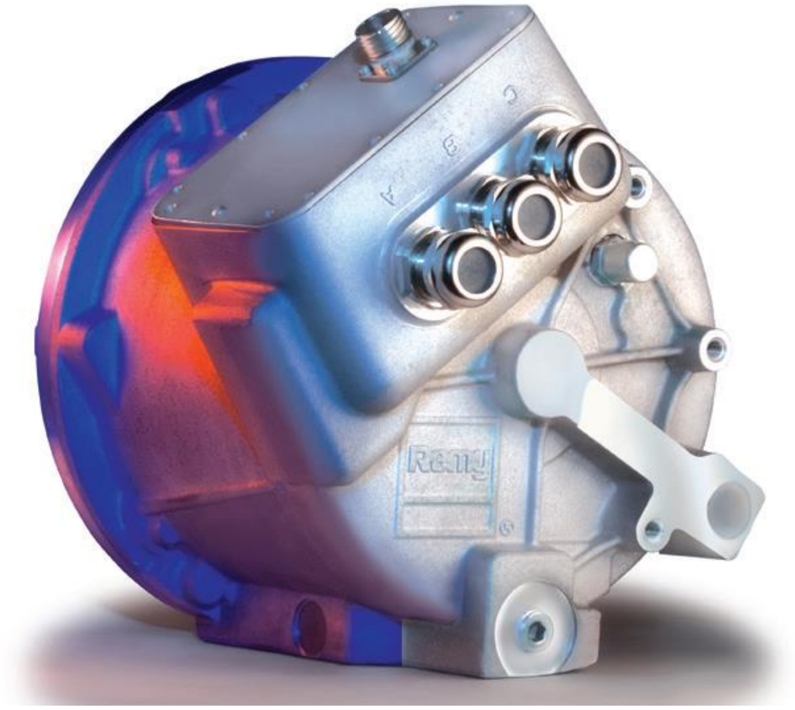


Figure 46 Electric Motor from Borg Warner used in Corbellati Missile

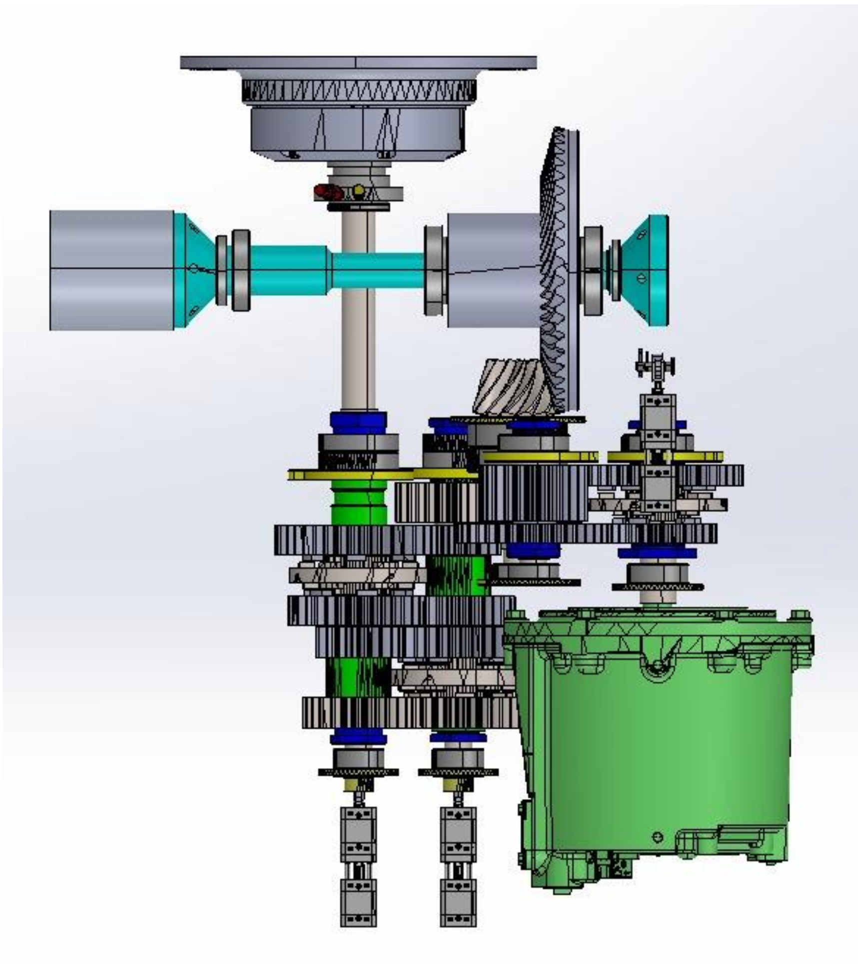


Figure 47 Corbellati Missile Transmission

3.2 Clutch Pneumatics

3.2.1 Requirements for the Clutch Actuation Mechanism

The criteria for choosing the Clutch actuation mechanism are

- A pneumatics-based system that can make use of the existing pneumatics available for the other system in the vehicle.
- Capable of achieving faster actuation speeds and high repeatability for cyclic usage.

3.2.2 Clutch Clamp Force Calculations

The clutch system that is being used in the vehicle is a Motorsports clutch developed by Sachs Performance which is a multi-plate clutch with 4 clutch plates of diameter 184mm with contact area on both sides of the plates with Sinter 2.6. The clutch system is a Diaphragm Spring Multiplate Clutch with a Push-type diaphragm spring. The model for the clutch pack is RCS 4.184-S2.6-D-S-49.

Using the official drawings provided by the manufacturer, Sachs Performance, we can derive the data required for the calculations. The following are the data for the clutch

External Diameter R_e = 0.1868 m
Internal Diameter R_i = 0.134 m
Average Friction Clutch Radius R_s = 0.1604 m
Friction coefficient μ = 0.4
Number of frictional surfaces n = 8 (2 frictional surfaces per clutch plate and the clutch pack has 4 clutch plates)
Maximum Engine Torque C_{eng} = 2350 Nm
Here, for our calculations we considered the β value to be 1.6 as the torque value is very high and usually found in trucks or high-performance vehicles.

Using the formulae

$$C = R_S \,.\, \mu \,.\, F_A \,.\, n \qquad \textit{Eq. 11}$$

$$p = \frac{F_A}{\pi\,.(R_e^2 - R_i^2)}$$ *Eq. 12*

$$C = C_{eng}\,.\beta$$ *Eq. 13*

We can calculate the clamping force to be F_A = 7.323 kN

3.2.3 Hydraulic Line Pressure Calculations

Using the total clamp force on the clutch plates, diaphragm spring pivot length and the slave cylinder piston dimensions the pressure in the hydraulic lines can be calculated. For the data regarding the slave cylinder, the technical drawings by Sachs Performance were used.

Clamp force F_A = 7.323 kN
Length from the clamp to pivot (B) = 0.04238 m
Length from pivot to outer edge (A) = 0.0217 m
External radius of the slave cylinder piston R = 0.0265 m
Internal radius of the slave cylinder piston r = 0.024 m

Using the formulae

$$A_{rb} = \pi\,.(R^2 - r^2)$$ Eq. 14

$$R_b = \left(F_{clamp}\right)\frac{A}{B}$$ *Eq. 15*

$$p_{hyd} = \frac{R_b}{A_{rb}}$$ *Eq. 16*

We can calculate the hydraulic pressure to be p_{hyd} = 9.459 MPa

3.2.4 Master Cylinder Calculations

With the help of the hydraulic line pressure and the dimensions of the existing master cylinder, we can calculate the clutch pedal force required for the operation of the hydraulic clutch.

Where,

Radius of the master cylinder piston r_p = 0.00953 m

Hydraulic Line pressure p_{hyd} = 9.459 MPa

Using the formula,

$$A_m = \pi . r_p^2 \qquad \textit{Eq. 17}$$

$$F_p = A_m \; . \; p_{hyd} \qquad \textit{Eq. 18}$$

We can calculate the Pedal force F_p = 2694.7 N

3.2.5 Pneumatic Cylinder Calculations

The pneumatic pressure available for use in the vehicle is 7 bar and the pneumatic cylinder should be able to operate under that pressure input to provide the necessary clutch pedal force.

Using the formula,

$$\emptyset = \sqrt{{4 . F_p} \big/ {\pi \; . \; P_n}} \qquad \textit{Eq. 19}$$

Where,

Pedal Force F_p = 2694.7 N

Operating pneumatic pressure P_n = 7 bar

We can calculate the Bore diameter of the pneumatic piston as Φ = 0.070 m.

3.2.6 Pneumatic Cylinder Selection

With the data obtained and the formulating of a requirement list, the correct pneumatic piston can be chosen. The requirement can be outlined as

- The pneumatic cylinder should have a minimum piston diameter of 70 mm.
- The operational stroke for the pneumatic cylinder should be 35mm according to the stroke of the master cylinder.
- The pneumatic cylinder should have a female mounting option to place the custom-fabricated pushrod to be used in place of the one present in the master cylinder.
- The pneumatic cylinder should have a magnetic piston so that a magnetic position sensor could be used to get feedback from the piston and use it to optimize the control of the pneumatic cylinder.

We chose the Double Acting Magnetic pneumatic cylinder with a piston diameter of 100mm with a stroke length of 60 mm produced by Artec Pneumatics (Figure 48) with a slot for the Magnetic position sensor. The sensor has a 96mm range analogue output Magnetic position sensor made by a company called SICK. This would be the piston and sensor that will be used to act as the pedal for a clutch system thereby making the manual clutch an electronically controlled Pneumatic actuated clutch.

Figure 48 Pneumatic Double-Acting Magnetic Cylinder

Figure 49 SICK MPS Sensor

3.2.7 Working of the Pneumatic Clutch Actuation System

The pneumatic system designed to actuate the clutch is a very simple implementation where the clutch pedal of a manual clutch system, shown in Figure 50 is replaced by a pneumatic piston Figure 51. For the operation of the pneumatic cylinder, a 5/3-way bi-stable and 5/2-way mono-stable pneumatic solenoid valves have been chosen. A custom-made spindle diameter of 12mm to connect the pneumatic piston to the piston of the master cylinder was designed.

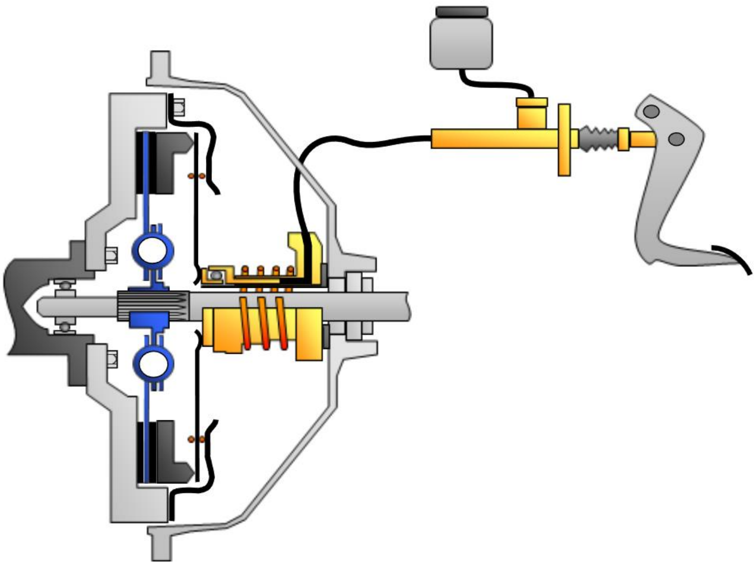


Figure 50 Clutch System with a manual lever for Clutch Actuations

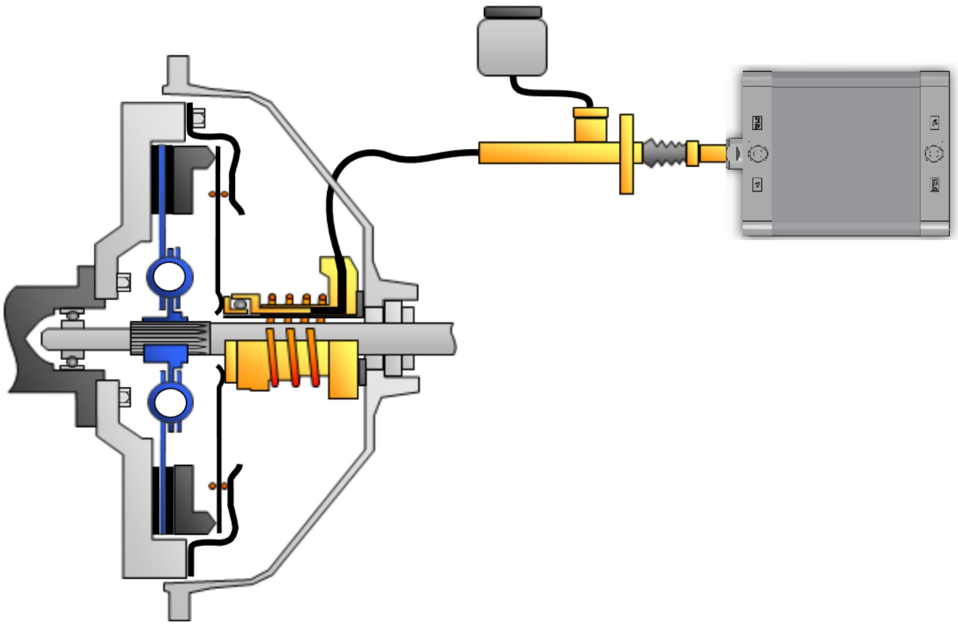


Figure 51 Representation of the proposed Clutch Pneumatic Actuation system for the Corbellati Missile

The Pneumatic solenoid valve would send the air from one port of the solenoid which would extend the pneumatic cylinder piston and the piston rod with a custom-made pushrod would act on the piston of the master cylinder which would send the hydraulic fluid towards the concentric slave cylinder. This hydraulic pressure extends the slave cylinder piston and applies force on the diaphragm of the clutch and disengages the clutch. To re-engage the clutch the hydraulic pressure is slowly reduced by actuating the other ports of the solenoid valves which reverse the airflow direction into the pneumatic piston thereby reducing the force applied by the pneumatic piston on the master cylinder piston. The extension of the pneumatic piston with the 5/2-way solenoid valve is shown in Figure 52 and the retraction stroke of the pneumatic piston is shown in Figure 53.

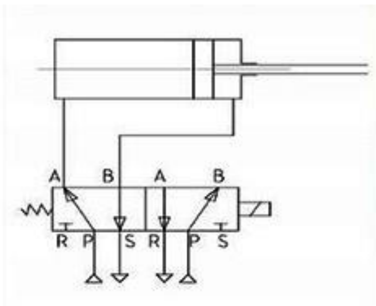


Figure 52 Pneumatic Cylinder Extension Stroke

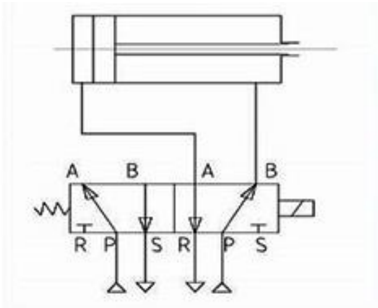


Figure 53 Pneumatic Cylinder Return Stroke

The control of the solenoid valve provides much better control over the clutch actuation. The clutch actuation is not always engage-disengage function and to move the car from a standstill position the clutch needs to be modulated which can be achieved by controlling the pneumatic piston extension and retraction rate.

The retraction rate can mainly be controlled by opening and closing the pneumatic circuit continuously with a specific frequency and this needs to be done at the desired rate to achieve the perfect clutch re-engagement. The switching frequency of the pneumatic solenoid valves needs to be kept in consideration while formulating a control strategy for the solenoid control. The usual operational switching frequency is between 5ms and 50ms and the solenoid valve is controlled by varying the amount of time for which the operational voltage of 12V DC is provided to the coils. A PWM signal with varying switching frequencies can be used with a 5/2-way mono-stable solenoid valve.

An alternative option for the same actuation system would be to use a 5/3-way solenoid valve is bi-stable coils and spring retraction normally closed solenoid valve instead of the 5/2-way mono-stable valve. The actions that the 5/3-way pneumatic solenoid needs to perform would remain the same but the control strategy to achieve the desired clutch engagement and disengagement with modulation for the moving-off condition would be handled differently. Instead of providing the coils with a PWM signal the same type of signal needs to be sent in an alternating way to both the coils so that one would actuate the other coils to counter the motion of the thereby slowly moving the plunger to open and close the pneumatic circuits and as the 5/3-way solenoid chosen has a blocked centre the piston would move and stay in a fixed position till the coil is re-energized to move it further or retract it.

These proposed configurations need to be tested to check for their functionality, response times, load capacity, cyclic functionality, actuation pressures at the slave cylinder, solenoid control strategies, and the pneumatic cylinder’s ability to perform the actuation as intended. This would require the development of a complex control system with various feedback loops acting as input based on which the set functions are carried out as programmed and the development of such a complex system would entail a lot of research and development into the various control system being employed by the other manufacturers in the passenger vehicles and understanding the equipment that is going to be used in the prototype being developed.

3.3 Test Bench

The purpose of a test bench is to evaluate the proposed pneumatic clutch and gearbox actuation configuration to check for issues with compatibility, functionality, and controllability of the pneumatic actuation system according to the required functionality.

For the test bench, the requirements are

- Should be able to simulate the actuation of the master cylinder.
- Ability to check the functionality of the system with the 5/2-way and 5/3-way solenoid valves.
- Ability to test the custom pushrod for the master cylinder.
- Ability to verify the control strategy for the pneumatic solenoid for clutch modulation and engage-disengage functionality.
- Ability to test the shifting of the gears for upshifting and downshifting.
- Ability to test the system in ICE, EV and Hybrid modes.
- Data acquisition from the sensors to later analyse the functionality of the system and actuation strategy.

The architecture of the test bench is designed to house the Pneumatic piston, the Master cylinder, hydraulic lines connected from the master cylinder to the slave cylinder, the slave cylinder and a strain gauge. When the pneumatic piston is actuated by the solenoids it moves the push rod connecting the pneumatic piston to the piston of the Master cylinder which compresses the hydraulic fluid and that in turn pushes the piston of the Concentric Slave cylinder which pushes a flat metal plate onto the stain gauge. The force applied by the concentric slave cylinder is what is measured by the stain gauge which gives us an accurate idea if the force applied is enough to engage and disengage the clutch. The entire system is controlled by an Arduino board that connects the various pneumatic solenoids, the sensors mounted on the pneumatic piston, oil pressure gauges and the strain gauge to the computer where the data is displayed.

When the Arduino sends a signal for the actuation of the clutch the solenoid valve that operates the clutch, it starts to actuate the pneumatic piston to engage or disengage the clutch based on the given logic. To modulate the clutch the pneumatic valve is turned off and on multiple times a second to smoothly move the piston forward slowly increasing the force applied on the master cylinder thereby producing a varying hydraulic pressure with a gradual rise in pressure that moves the slave cylinder to the clutch and in case of the test bench onto the plate that exerts force evenly onto the strain gauge. The signals from the magnetic position sensor on the pneumatic piston, the hydraulic line pressure gauge and the strain gauge are sent to the Arduino which is then relayed to the display that shows all the data acquired from the start of the test to the end of the test at a frequency of 250Hz. The test scenarios are run continuously to replicate multiple actuations for both clutch modulation where the piston moves slowly to engage the clutch and the other where it needs to engage and disengage the clutch fast to aid faster gear shifting.

The cut section view of the pneumatic cylinder and the master cylinder setup is shown in Figure 54. The CAD model for the Pneumatic cylinder was taken directly from the manufacturer’s website while the master cylinder was designed completely in Solidworks using the dimensions of the OBP Master Cylinder using the data from the technical datasheet.

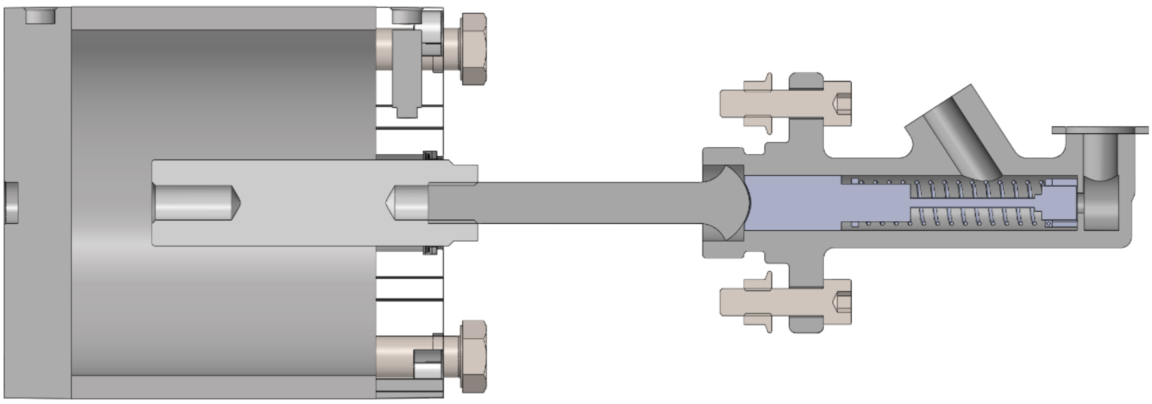


Figure 54 Cut section of the Pneumatic Piston and the Master Cylinder

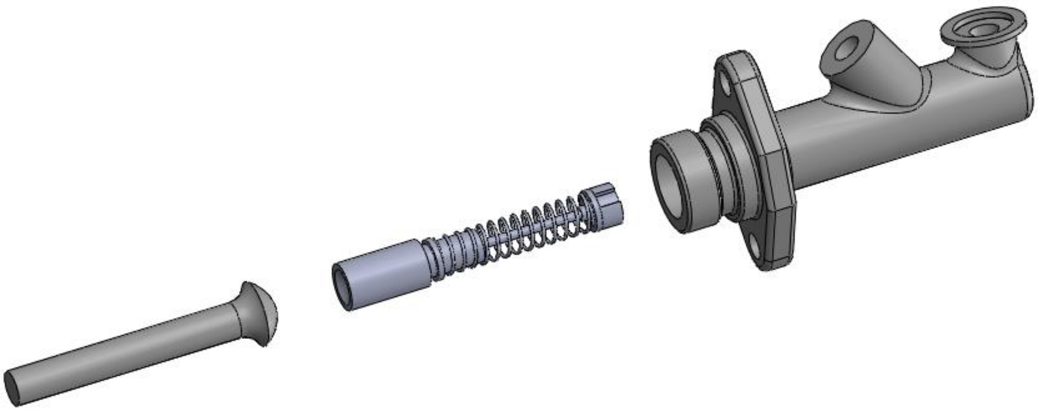


Figure 55 Master Cylinder CAD Model

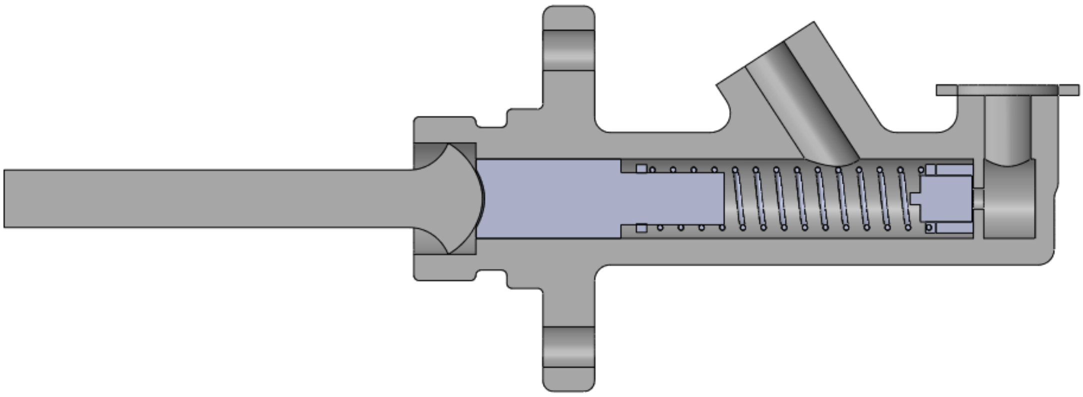


Figure 56 Master Cylinder with the Custom Push Rod

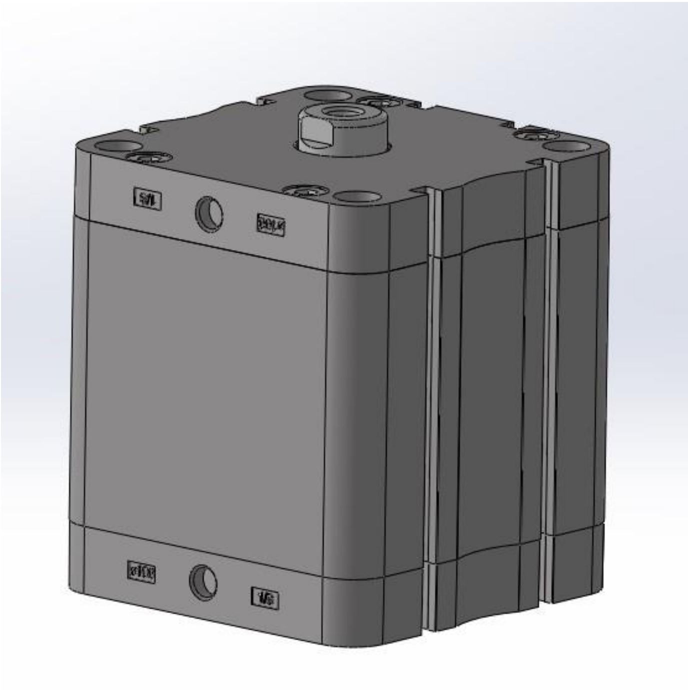


Figure 57 Pneumatic Piston from ARTEC

For the test setup, we can use a pneumatic cylinder of 100 mm bore diameter with a 60 mm store used to actuate a clutch master cylinder which was a standard 0.75-inch piston diameter used for most clutch systems. The custom push rod was designed to replace the push rod of the master cylinder with the dimensions on the other side to match the thread available in the pneumatic cylinder piston. The cut section of the custom push rod in the master cylinder is shown in Figure 56.

The clutch actuation to engage and disengage the clutch is done by energizing or de-energizing the coils of a pneumatic solenoid but when it comes to the modulation function of the clutch the operational logic of the power to the coils needs to be modified. To verify the correct solenoid for the application we choose both the 5/2-way and 5/3-way solenoids to test out the clutch actuation mechanism to see which one is better and which one can provide the smooth actuation function. For this, logic needs to be made on controlling the 5/2-way and 5/3-way solenoids individually as they have different functionalities and operate slightly differently. The magnetic position sensor data and the data from the strain gauge can be used to check the functioning of the 5/2-way solenoid valve as well as the 5/3-way valve. This would show the working of the clutch and the pneumatics responsible for the actuation under various conditions and that helps in deciding the right solenoid valve and the actuation logic to act as the controller for our pneumatic circuit.

The verification of the control strategy is dictated by the results observed with the pneumatic shifting of the gears according to the developed strategy for the right inputs. As we know the working of the mechanical gearbox and the working of the shift mechanism manually, the simulation of the pneumatic solenoids with the control logic would verify the system and determine the level of functionality of the developed system.

The Arduino acts as a controller for the test bench. The code was written in a way that it takes input from the user through assigned switches connected to the ports and executes either the ICE mode, Hybrid mode or EV mode control strategies. The solenoids to be activated are programmed into each mode and then based on the user upshift or downshift the code changes the gears. After the execution of a gear actuation, the user gets visual feedback with an indication of the gear shifting and if the user tries to downshift or upshift beyond the gears present, 0 and 4 respectively, then it would show a pop-up on the digital display and computer to say the gear cannot be further changed. The code includes a command to run cyclically shifting up and down and changing between modes randomly without user input to test the cyclic repeatability of the function. This gives us an accurate understanding of the repeatability of the system and the functionality of the solenoid valves connected to the controller.

The Arduino provides a 5V output to actuate the solenoids but as the actuation voltage for a solenoid valve is between 12V and 24V DC we need a voltage converter or an interface module that takes 5V as input and bridges a 12V DC input for the power source to actuate the solenoid valve. We use the bridging interface where a custom-developed MOSFET module was made that takes the 5V input from the Arduino and then bridges a circuit that sends 12V to the appropriate circuit on the solenoid valves. These being Solid State MOSFET modules, do not have a physical moving metal plate to bridge the circuit, thus allowing a faster and high frequency on and off control to facilitate modulation in the pneumatic piston motion. The interfacing of the Arduino with these MOSFET modules can be seen in Figure 59.

The Pneumatic Circuit for the test bench is illustrated in Figure 60 where the connections between the pistons and the solenoid valves are shown along with the connections to the compressed air source which is an air tank connected to a compressor which fills the tank with air as it senses a loss in pressure. All the solenoids are connected to one distribution block to ensure that there is a simpler connection to all the solenoids from the air source and it would be an easier piece to diagnose any leaks or to replace the connections if necessary as it would be placed in an accessible place in the vehicle. The 2 ports on the distribution block are blocked off as they are connected to the pneumatic pistons that actuate the rear spoiler of the car running on the same pneumatic circuit but not an application of this project.

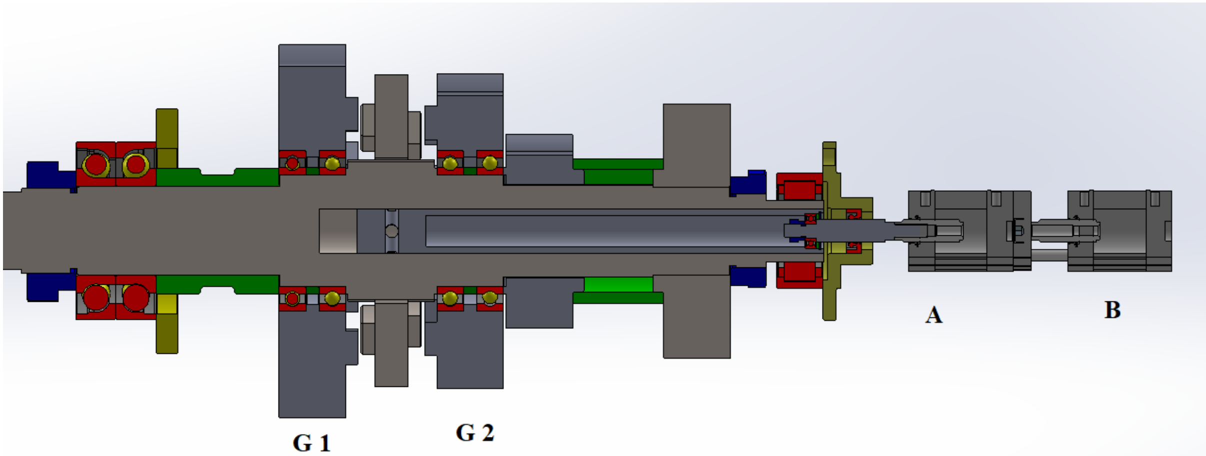


Figure 58 Cut Section of the Corbellati Missile Gear Box Primary Shaft

Figure 58 shows the cut section of the primary shaft of the gearbox to illustrate the connection between the dog-clutch and the pneumatic pistons. The shifting mechanism is a shaft inside the primary or the secondary shaft that is linked to the dog clutch with the help of 4 long pins. The dog is moved by sliding the shifting mechanism shaft forward or backwards with the help of 2 pneumatic pistons per shaft to have 3 possible positions for the dog-clutch. The configuration of the pistons in the Figure 58, and the location of the dog-clutch indicate that it is in Neutral position, and this is the set default position in the controller for solenoids and the pneumatic configuration achieved by having piston A in the retracted position and the piston B in the extended position. From Figure 58, if both the pistons extend then the gear G-1 is engaged and if both the pistons retract then the gear G-2 is engaged. This is the same configuration for the gears on the secondary shaft and the EV primary shaft. The Gear G-1 and G-2 are a representation of a shiftable gear on the shaft and not any gear. On the primary shaft gears 3 and 4 for ICE are selected and on the secondary shaft gears, 1 and 2 are selected. For the EV gear selection, both gears are on the EV shaft and a combination of these two helps in the selection of the Hybrid Gears. This is represented in Figure 43.

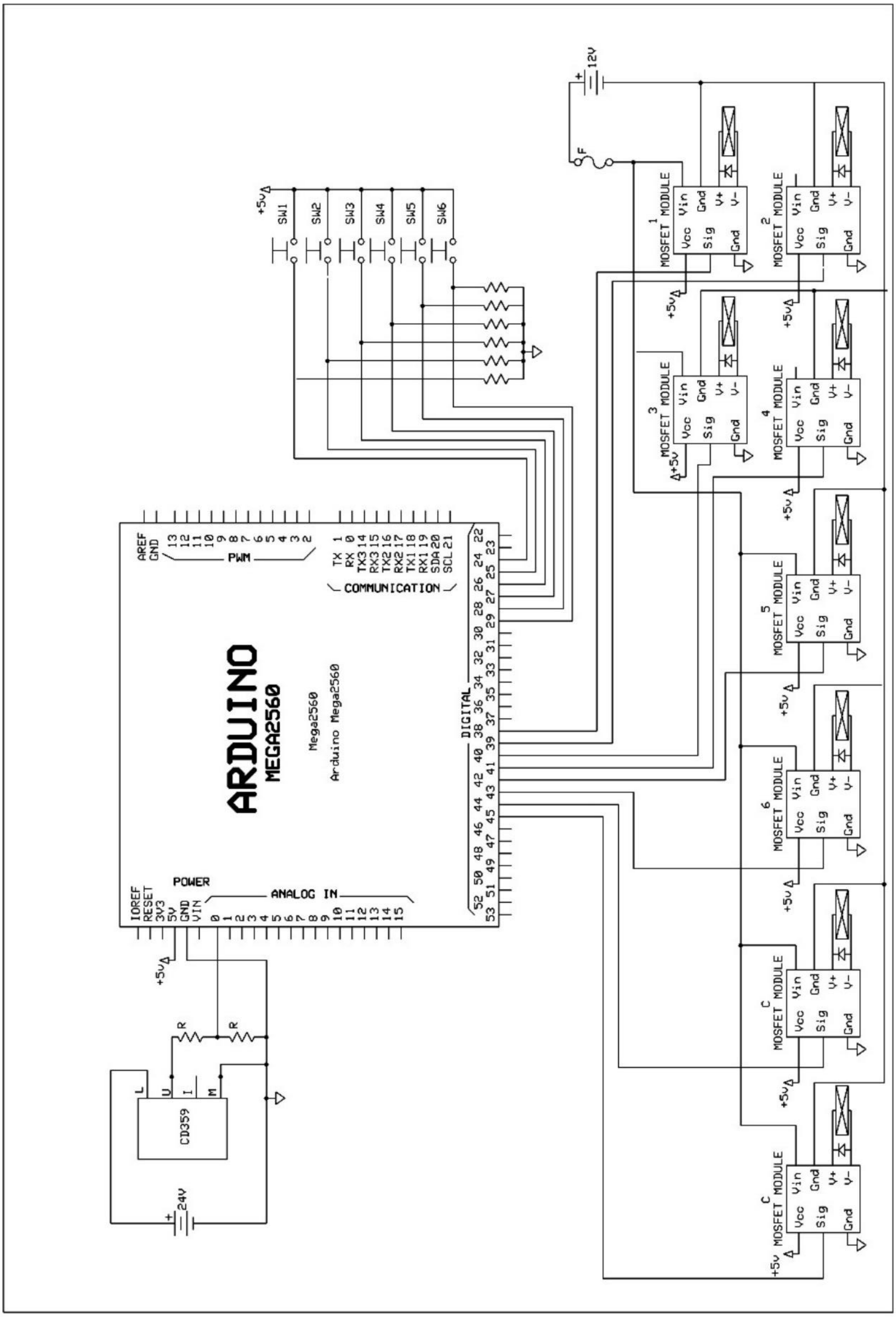


Figure 59 Arduino Circuit with the MOSFET modules for Solenoid Valves

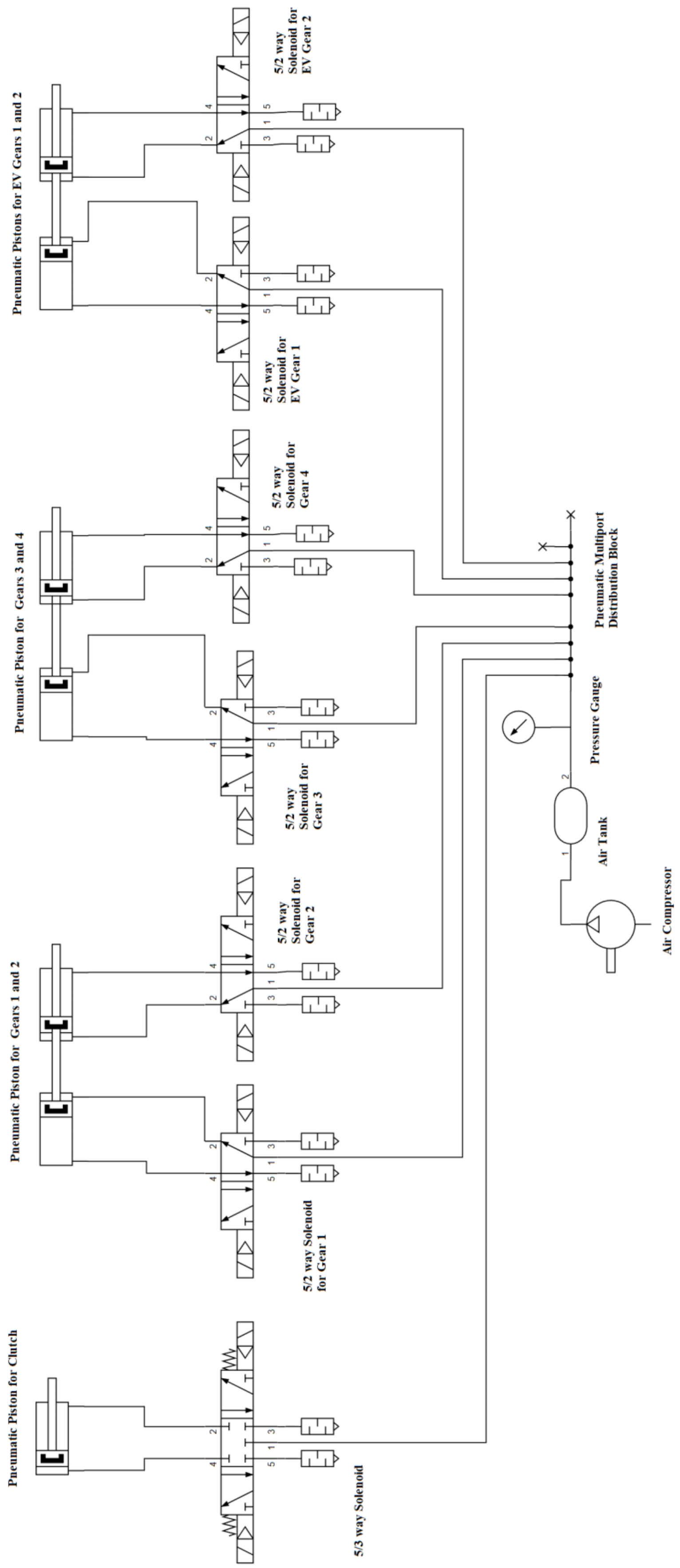


Figure 60 Pneumatic Circuit for the Gearbox

The results of the test bench simulation showed us that the chosen pneumatic pistons and solenoid valves would work in a range of 3.5 bar to 7 bar pneumatic pressure thus giving us the option to keep using the system while there is a drop in pressure. This is useful in case of any emergency actuations that are required to move the vehicle if the engine is not running or there is any other issue, and the car needs to be moved into a safe place. The test also proves that the connections made to the pneumatic pistons move to a specific position to engage neutral in case of a loss in pneumatic pressure or the loss in voltage works and can be relied on to shift to neutral as per the requirements for a failsafe condition. The test results showed that the required release force was achieved for the clutch as applied by the pneumatic piston and that gives us a varying range in which we can operate the pneumatic piston by varying the pneumatic pressure to achieve a varying force applied on the master cylinder. The tests gave us an insight into the logic being used to implement the control of the pneumatic solenoids, the failsafe scenarios and how the logic was necessarily changed to reduce the gaps between two successive actuation for the modulation of the clutch to achieve a smooth actuation.

3.4 Simulation of the actuation logic on Simulink

3.4.1 Requirements to perform the SIMULINK simulation

The simulation of the actuation logic needs to be done to understand the gearbox maps and use them to implement in the vehicle's control unit to allow the gearbox and clutch pneumatic to actuate at the correct instance and to have a smooth actuation to ensure better ride quality and no damage to the gearbox or the clutch.

The purpose of the simulation on MATLAB SIMULINK is as follows

- To understand the clutch actuation and implement it in the desired actuation curve that corresponds to the implementation in the vehicle.
- To simulate the gear actuation for an automatic vehicle.
- To simulate the clutch and gear actuation logic under the WLTP cycle to understand the operation of the system in an urban driving environment.

3.4.2 Vehicle Modelling in Simulink

3.4.2.1 Driver and Driving Cycle

The driving cycle chosen for the simulation is a WLTP Class 3 cycle. WLTP stands for Worldwide Harmonized Light Vehicles Test Procedure, and it was proposed to simulate the world's average driving conditions. The WLTP cycle is divided into various classes based on the Power mass ratio and other classifications can be seen in Table 2. The Class 3, as seen in Figure 61, has been chosen for our simulations as it is a representation of the driving conditions available in Europe and Japan. The parameters of this class can be seen in Table 3. The use of this testing cycle ensures that the simulation for the gear shifting, and the clutch actuation is done on a widely accepted standardized cycle to ensure maximum compatibility and development of a logic that can be used to make a gearbox control module for the vehicle that has been simulated with the road speed conditions available in Europe. [9]

Category	PMR	Speed Phases	Comments
Class 3	PMR > 34	Low, Middle, High, Extra-High	If v_max < 135 km/h, phase 'extra-high' is replaced by a repetition of phase 'low'.
Class 2	34 ≥ PMR > 22	Low, Middle, High	If v_max < 90 km/h, phase 'high' is replaced by a repetition of phase 'low'.
Class 1	PMR ≤ 22	Low, Middle	If v_max ≥ 70 km/h, phase 'low' is repeated after phase 'middle'. If v_max < 70 km/h, phase 'middle' is replaced by a repetition of phase 'low'.

Table 2 WLTP Test Cycles Classification

Phase	Duration	Stop Duration	Distance	p_stop	v_max	v_ave w/o stops	v_ave w/ stops	a_min	a_max
	s	s	m		km/h	km/h	km/h	m/s²	m/s²
Low	589	156	3095	26.5%	56.5	25.7	18.9	-1.47	1.47
Middle	433	48	4756	11.1%	76.6	44.5	39.5	-1.49	1.57
High	455	31	7158	6.8%	97.4	60.8	56.6	-1.49	1.58
Extra-High	323	7	8254	2.2%	131.3	94.0	92.0	-1.21	1.03
Total	**1800**	**242**	**23262**						

Table 3 WLTP Class 3 Cycle Parameters

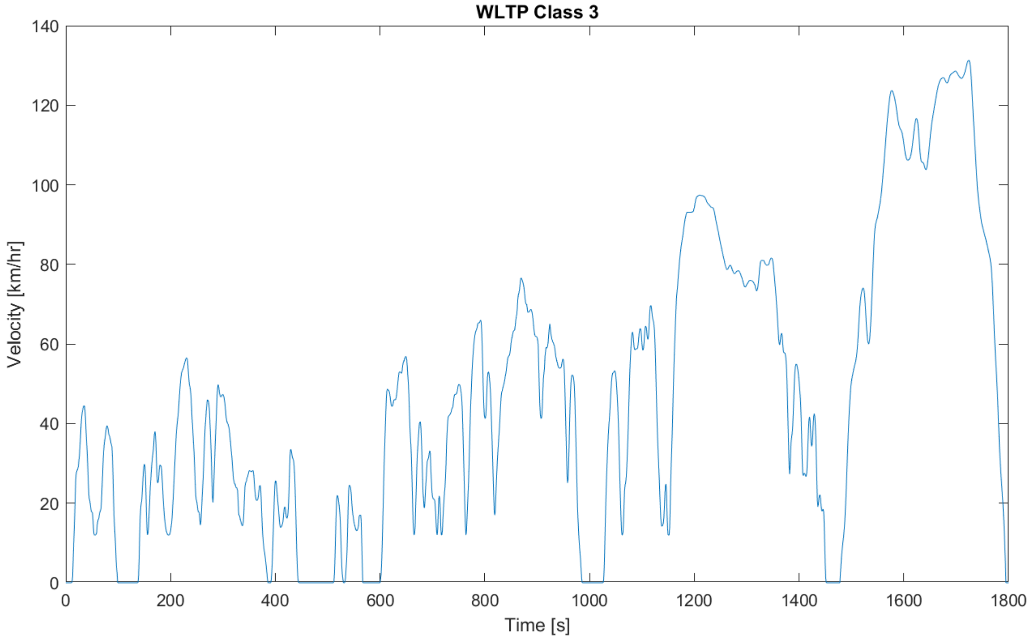


Figure 61 WLTP Class 3 Driving Cycle

The Longitudinal Driver block implements a controlled PI-type speed-tracking. Based on reference and feedback velocities from the Drive Cycle and the vehicle speed, the block generates acceleration and braking commands that can vary from 0 to 1. This block can be used to generate the commands necessary to track a longitudinal drive cycle for various simulations involving a driver and a drive cycle.

For our simulations, the WLTP driving cycle is chosen for Velocity reference (VelRef) and the Vehicle Velocity feedback (VelFdbk) is obtained from the vehicle body modelling. The grade at which the simulation is done is chosen to be 0. The longitudinal driver gives out an Acceleration command (AccelCmd) and Deceleration command (DecelCmd) which are used as input for the engine and the brakes, respectively.

Figure 62, represents the Driver and Driving cycle module used for this simulation in the SIMULINK environment.

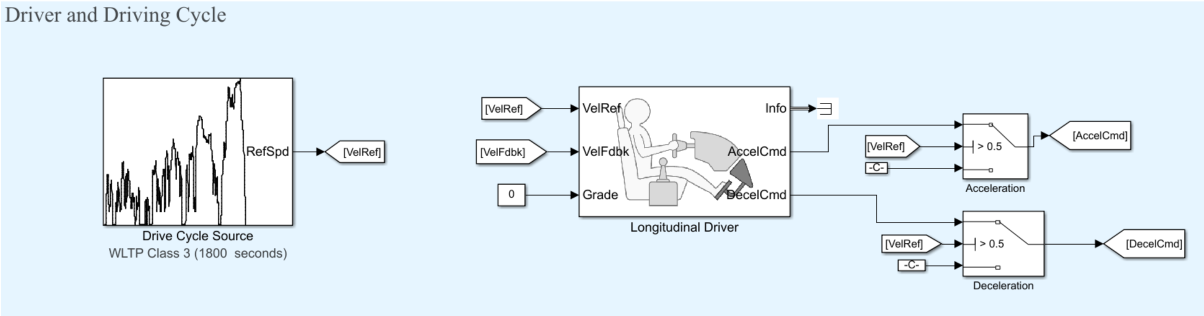


Figure 62 Driver and Driving Cycle Model

3.4.2.2 Powertrain System

The Powertrain System Module comprises the Engine, Clutch, Electric Motor, Gearbox, and Differential modules.

Engine Module

The vehicle's engine was modelled on SIMULINK by using a Generic engine block with the parameters of the engine set according to the engine being used in the vehicle. The acceleration command is given by the Longitudinal Driver based on the driving cycle and that acts as an input to the engine block. Ports B and F are mechanical rotation ports and the output from the engine is sent out from port F as a physical signal that is used as an input to the Clutch block. The port P is used to add an engine sensor to gather the rotational data from the engine and display it in the final results.

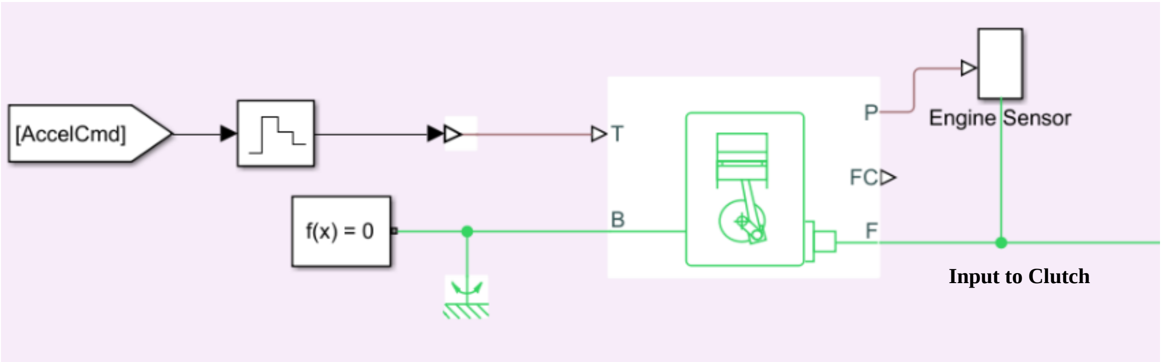


Figure 63 Engine Module on SIMULINK

Clutch Module

The vehicle's clutch system is designed to handle very high amounts of torque and it was developed by Sachs Performance. The clutch parameters have been put in the SIMULINK environment with the Clutch model based on the technical datasheet of the clutch which can be seen in Figure 65. The command for the clutch actuation is received from the Gearbox control unit.

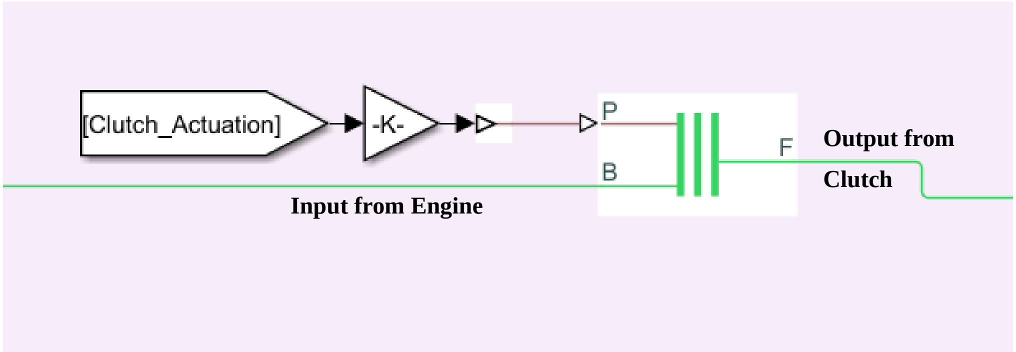


Figure 64 Clutch Module on SIMULINK

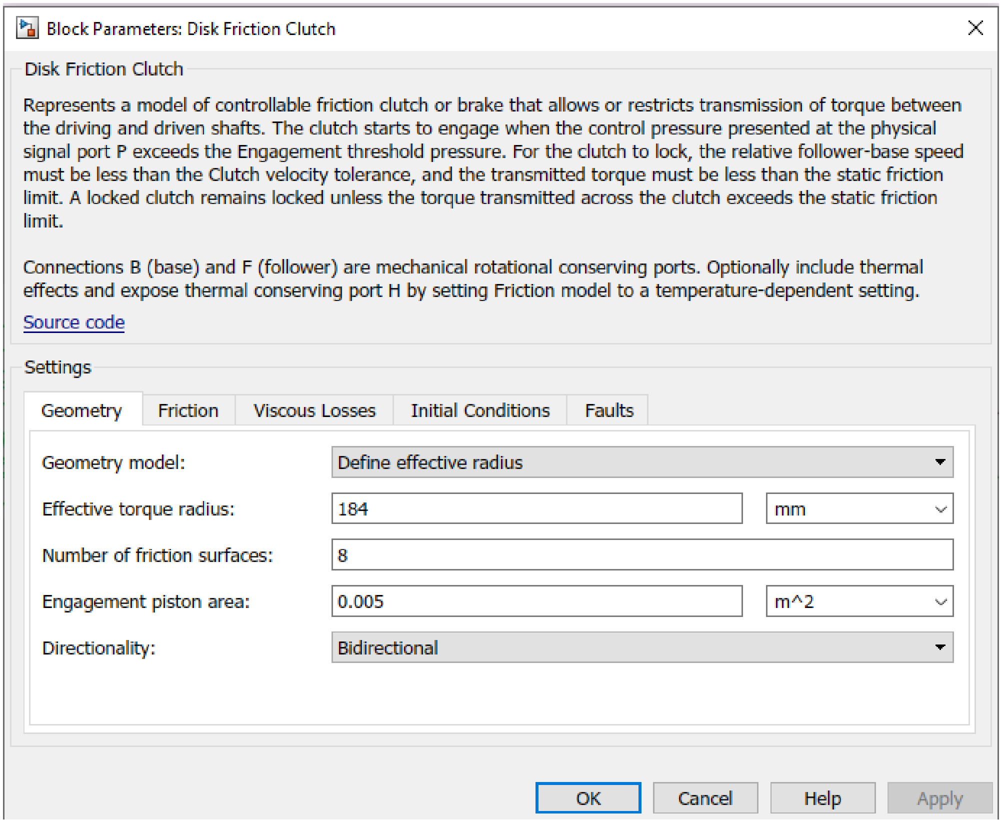


Figure 65 Clutch Parameters on SIMULINK

Electric Motor Module

The Electric Motor module consists of the DC Motor, H-Bridge, PWM Voltage Controller, Battery and a Current Source. The Parameters of the DC Motor were taken from the data sheet provided by Borg Warner for HVH250-090 Electric Motor. The battery module uses the parameters of an in-house designed and developed 700V battery made with Lithium-ion pouch cells. The throttle input from the driver is converted into a PWM voltage used to operate the current to the Electric motor thereby controlling the output torque. Figure 66 shows the connections made for the Electric Motor Module which is then connected to a gearbox.

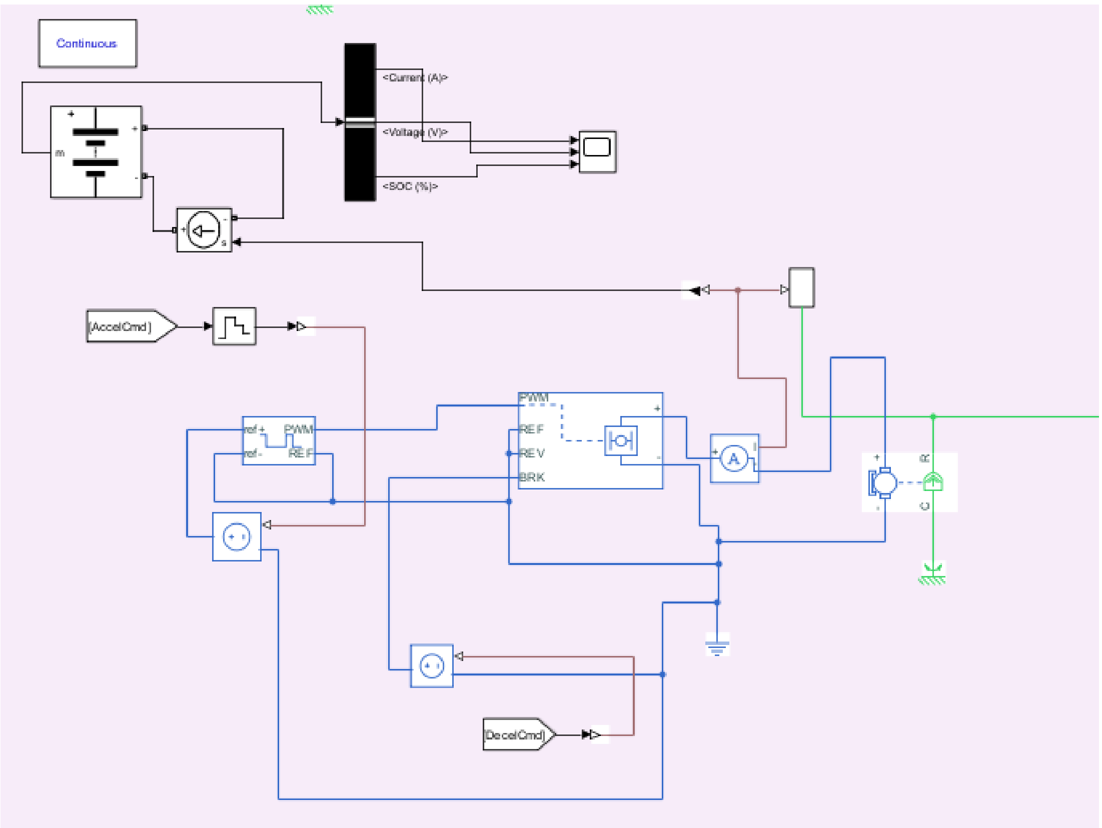


Figure 66 Electric Motor module on SIMULINK

Gearbox Module

The gearbox, as shown in Figure 43, houses 4 sets of gears to be used with the IC engine, 2 sets of gears to be used with the Electric Motor and can also be run in a combination for the Hybrid mode. The gear shifting is achieved by moving the dog clutch towards the gears with pneumatic cylinders. This functionality is shown in Figure 58. This Simulink module uses the pneumatic actuator positioning as a matrix command to control all the 6 pneumatic cylinders simultaneously and engage any of the gears corresponding to the logic being applied and the drive mode the vehicle is in. This has been programmed into the model for the actuators.

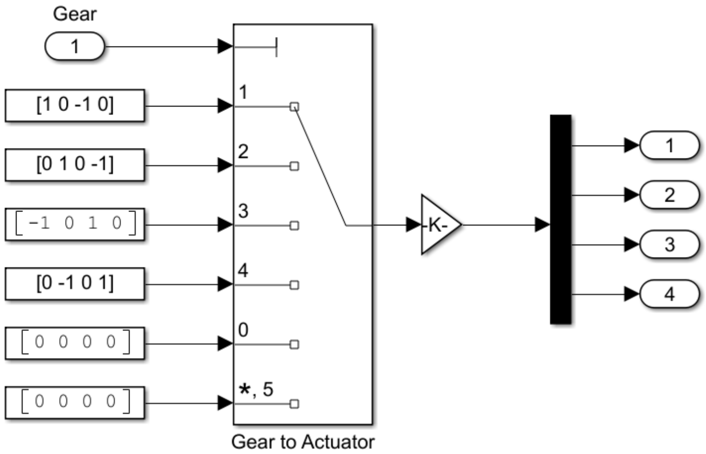


Figure 67 Gear to Actuator Command Module for ICE and Hybrid Modes

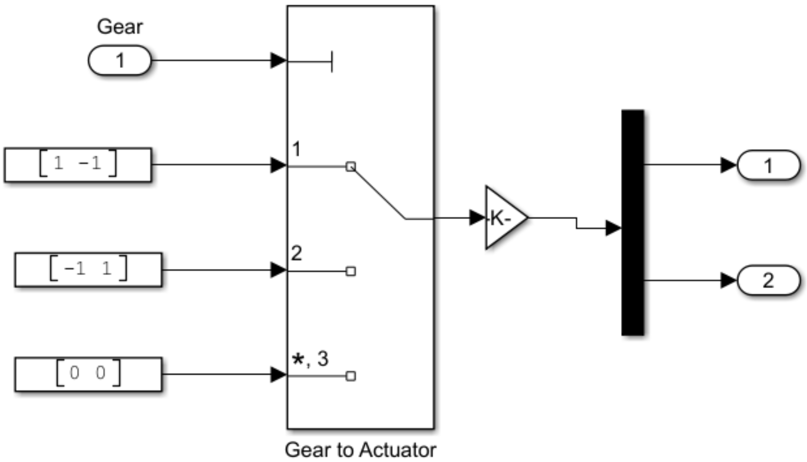


Figure 68 Gear to Actuator Command Module for EV Mode

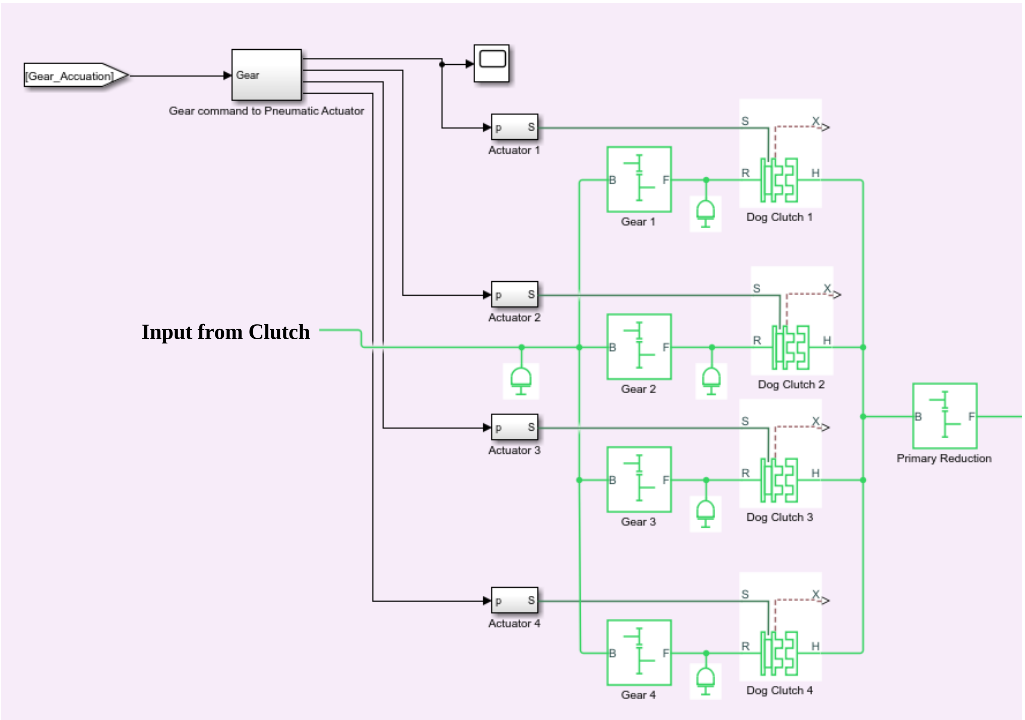


Figure 69 Gearbox Module of ICE on SIMULINK

The gearbox module of ICE was made as shown in Figure 69, and the data for the gear ratios and the dog clutch were taken from the company and programmed into the model. The output from the clutch acts as the input for the gearbox as it is a physical connection, and it is connected to the B ports for the gears to drive them the output from this is obtained from port F and is transmitted to the final drive when the corresponding dog clutch is actuated by the pneumatic actuators.

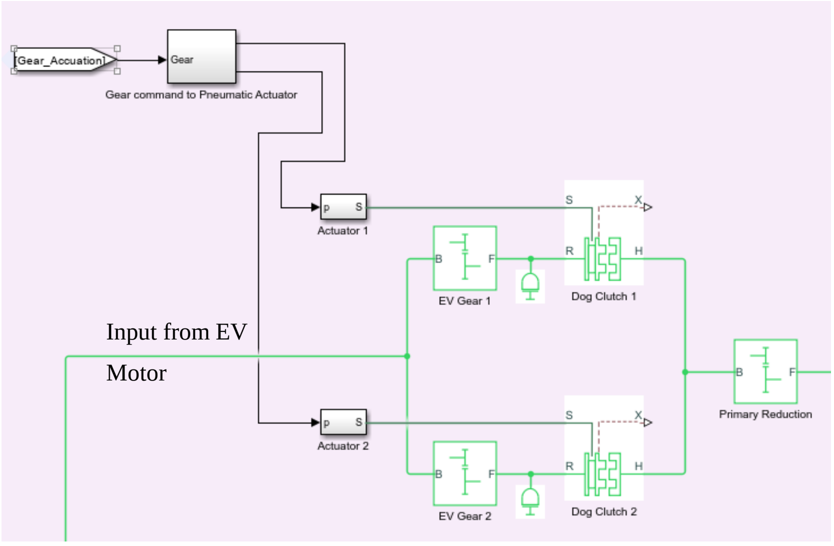


Figure 70 EV Mode Gearbox Model on SIMULINK

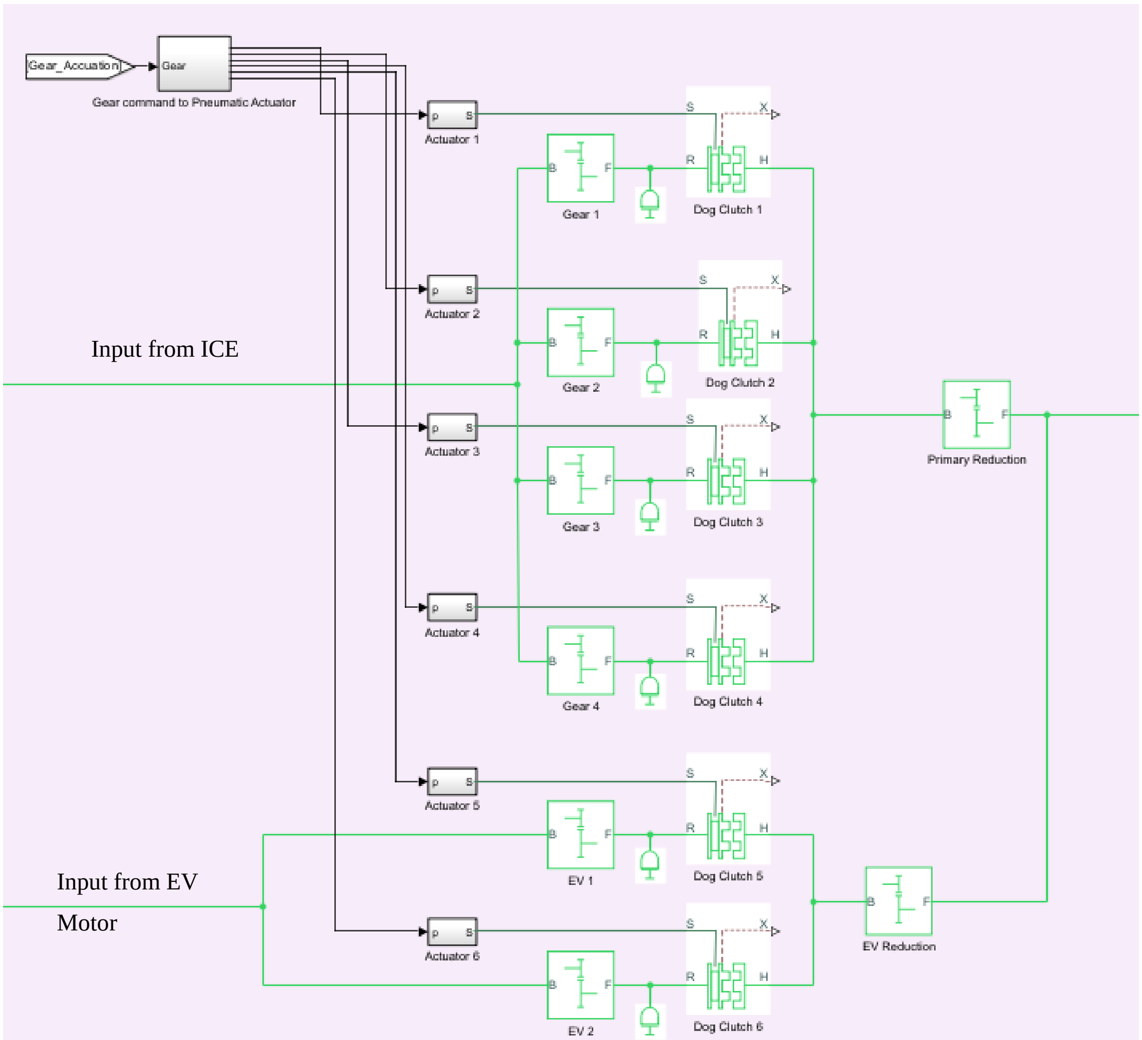


Figure 71 Hybrid Mode Gearbox Module on SIMULINK

The gearbox for the EV side is shown in Figure 70, and a combination of the two gearboxes through a common output shaft is shown in Figure 71. For the vehicle in the hybrid mode, it was decided by the gearbox development team at Corbellati to run as a 4-speed hybrid configuration although an 8-speed configuration is possible for the initial project. A proposed revision of the Engine and Electric Motor are in the works that use the 8-speed configuration.

Differential Module

The differential data was obtained from the company and was used in the Simulink model to get the correct final ratio. The output from the gearbox acts as the input for the differential. The output is sent to the 2 wheels of the rear axle as this is a model for a rear-wheel-drive vehicle.

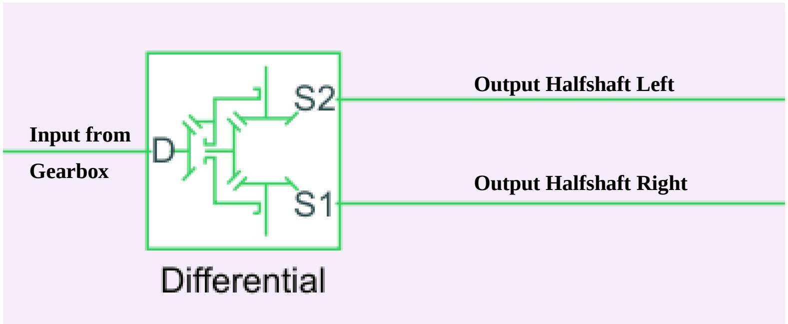


Figure 72 Differential Module on SIMULINK

The Engine, Clutch, Gearbox, and Differential comprise the Powertrain Module for ICE Mode and which can be seen in Figure 73, and the output from this module would act as the input for the models in the Vehicle body module.

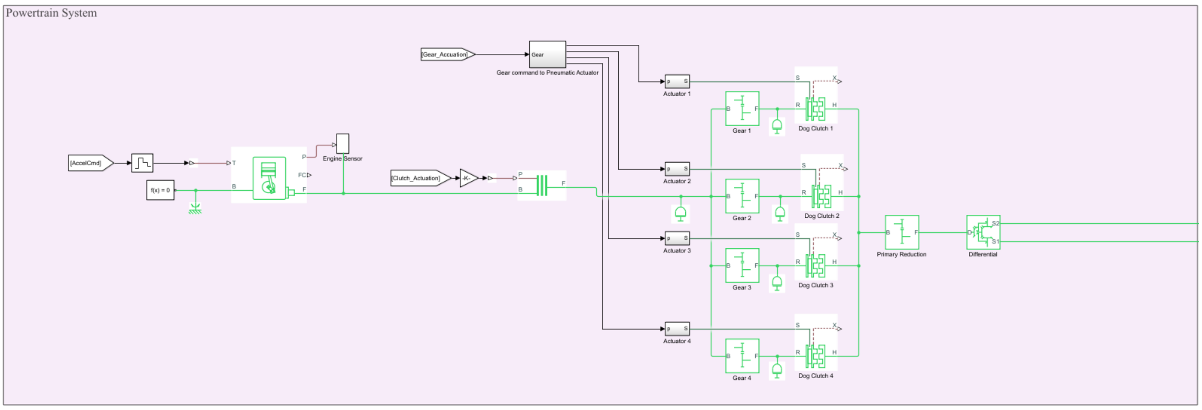


Figure 73 Powertrain Model for ICE Mode on SIMULINK

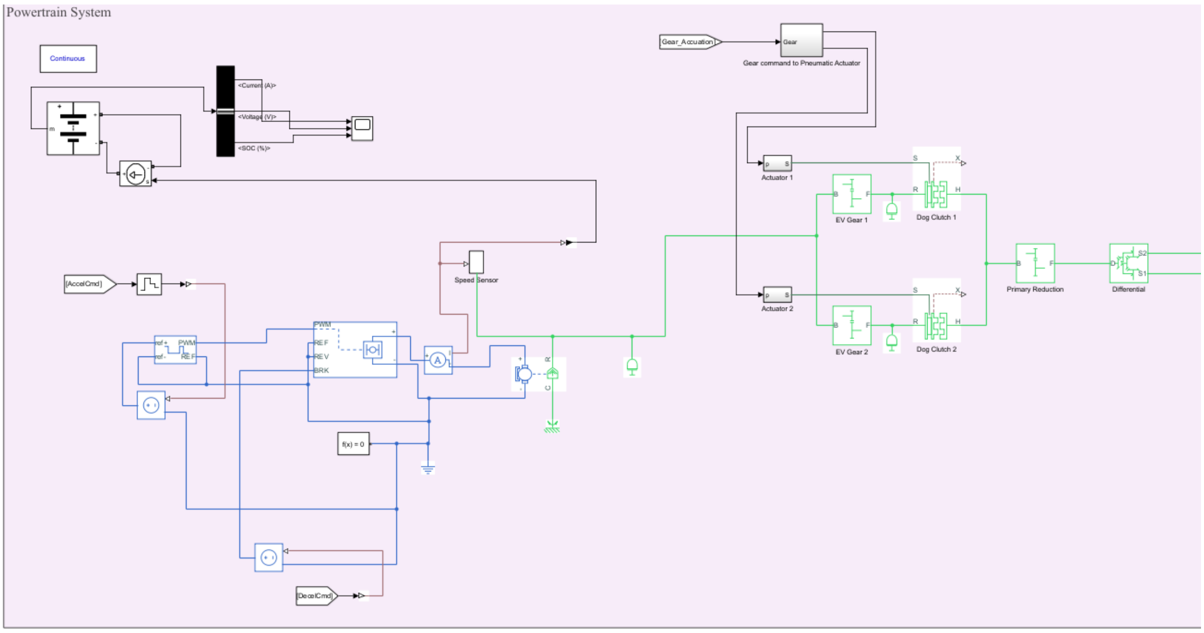


Figure 74 Powertrain Module for EV Mode on SIMULINK

3.4.2.3 Gearbox control unit

The gear shifting uses a Simulink threshold function to decide between upshift, downshift and a steady state condition. The function has a conditional command that it would only upshift or downshift when there is a change in the vehicle speed required and compare the throttle input from the driver and the vehicle speed, as seen in Figure 81. The output is a gear number into which the vehicle is requested to shift, as seen in Figure 79 and Figure 80 for ICE and Hybrid modes and EV mode respectively. During the shifting, the corresponding substrates are followed to send out the clutch actuation and the gear actuation commands.

The gear maps were developed for the vehicle to travel in the city and the maximum speed with the map is set to 200 km/hr to mimic an electronic limit on the top speed the 4 gears were interpolated according to that limit for ICE and Hybrid modes of operation. The Upshift threshold maps, Figure 75, show the distribution of gears for varying throttle inputs and vehicle speeds. The Downshift threshold maps, Figure 76, show the distribution of the gears for downshifting logic.

The gearbox was designed to go to a maximum speed of 540 km/h in Hybrid mode but for our city driving simulation, an alternative lower speed electronically limited gear threshold maps are being implemented. The maps were taken from the gearbox department and plotted in the State flow function with the help of a 2-D Lookup table that has the values of the Throttle input from the driver, Vehicle Speed and the corresponding Gear. For Upshifting and Downshifting the maps we input into these lookup tables and based on the command request sent by the Simulink function the gear value is taken from the Upshifting table for Upshifting or from the Downshifting table for Downshifting.

The ICE and the Hybrid modes share the same gear map as they have the same 4-speed configuration while the EV mode has a separate map due to its 2-speed configuration.

In Figure 75 & Figure 76 for the ICE and Hybrid Modes, the Pink line indicates the gear threshold for Gear 1, Orange indicates Gear 2, Blue indicates gear 3 and Purple indicates gear 4 and Light Blue indicates Neutral (gear 0). Based on these lines the system will decide which gear it needs to be in while taking the throttle, vehicle speed and previous gear as input.

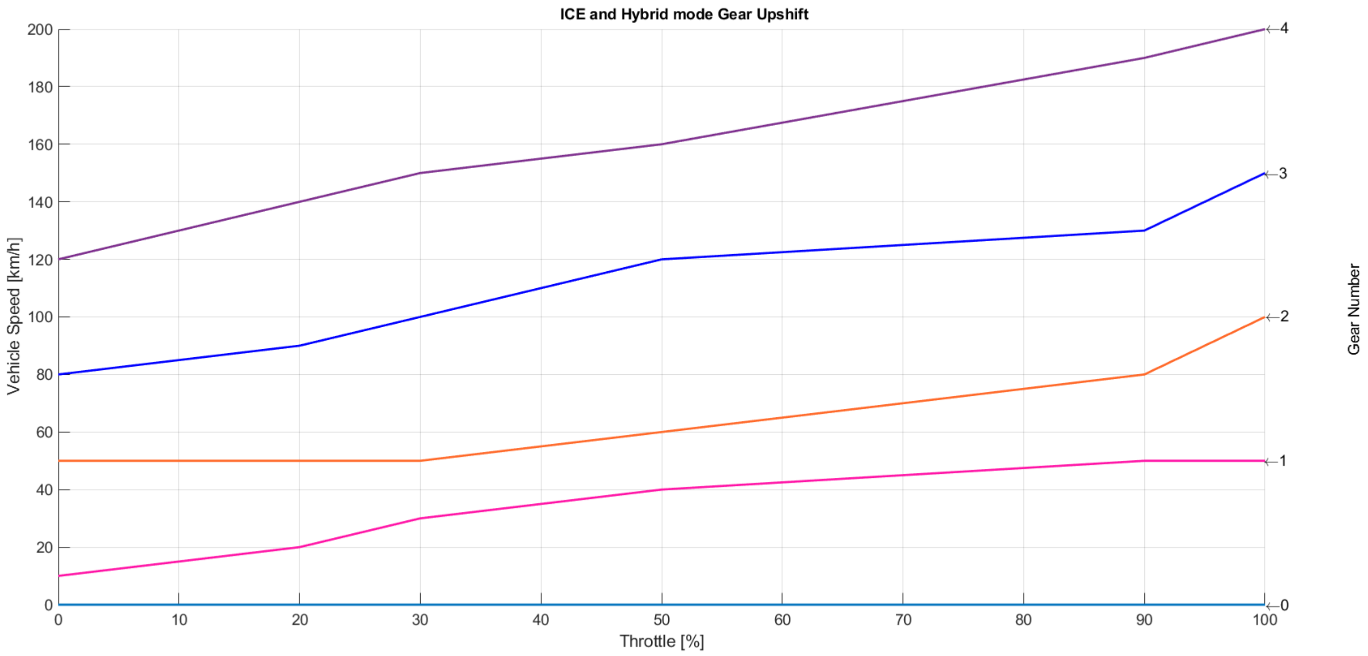


Figure 75 Gearbox Map for ICE and Hybrid Modes Upshift Threshold

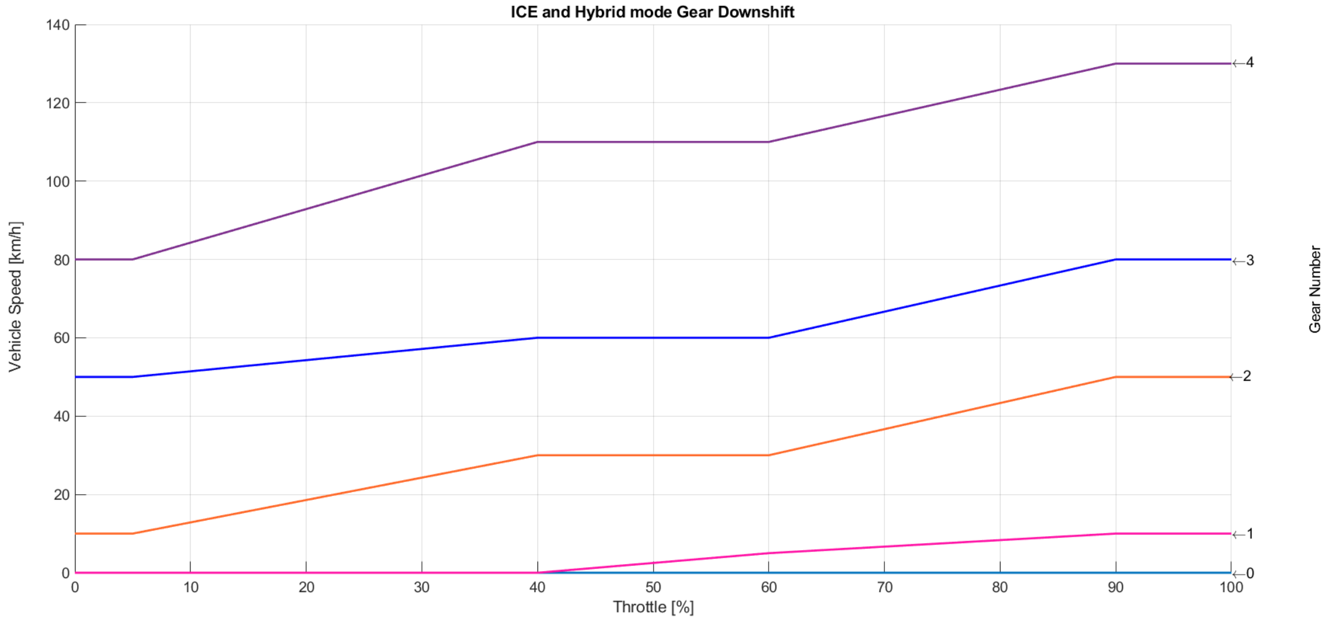


Figure 76 Gearbox Map for ICE and Hybrid Modes Downshift Threshold

In Figure 77 & Figure 78 for the EV mode gear maps, the Pink line indicates the Neutral (gear 0) position, Blue indicates Gear 1 and Red indicates Gear 2 corresponding to the input throttle and vehicle speeds.

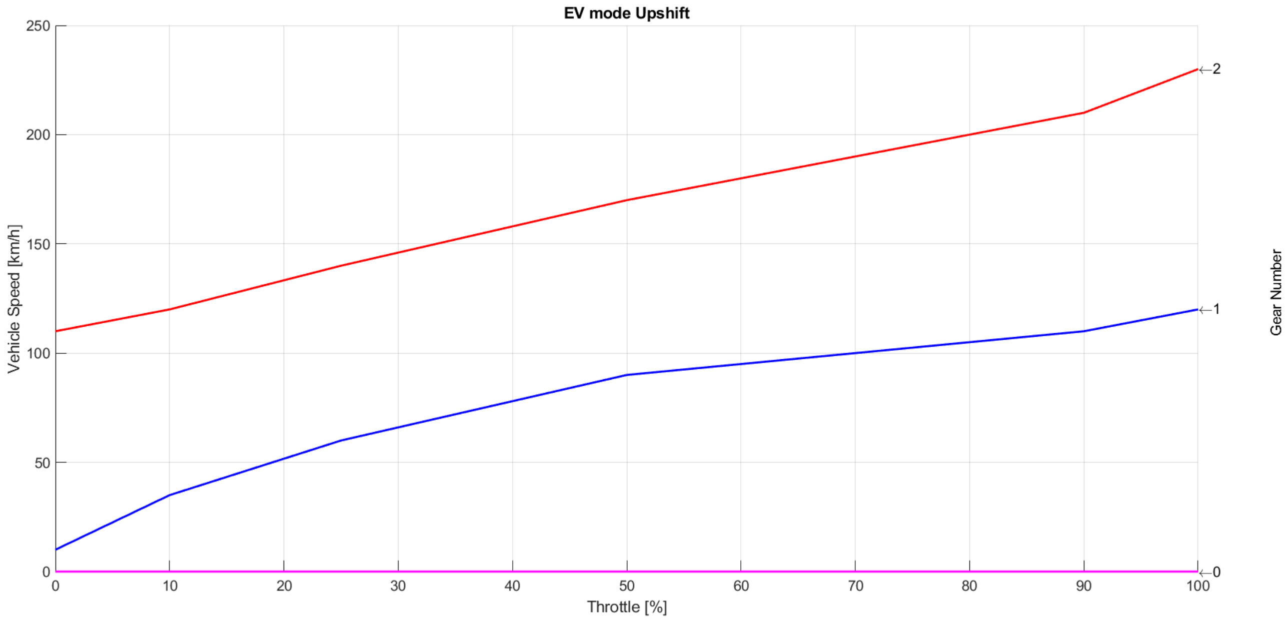


Figure 77 Gearbox Map for EV Mode Upshift Threshold

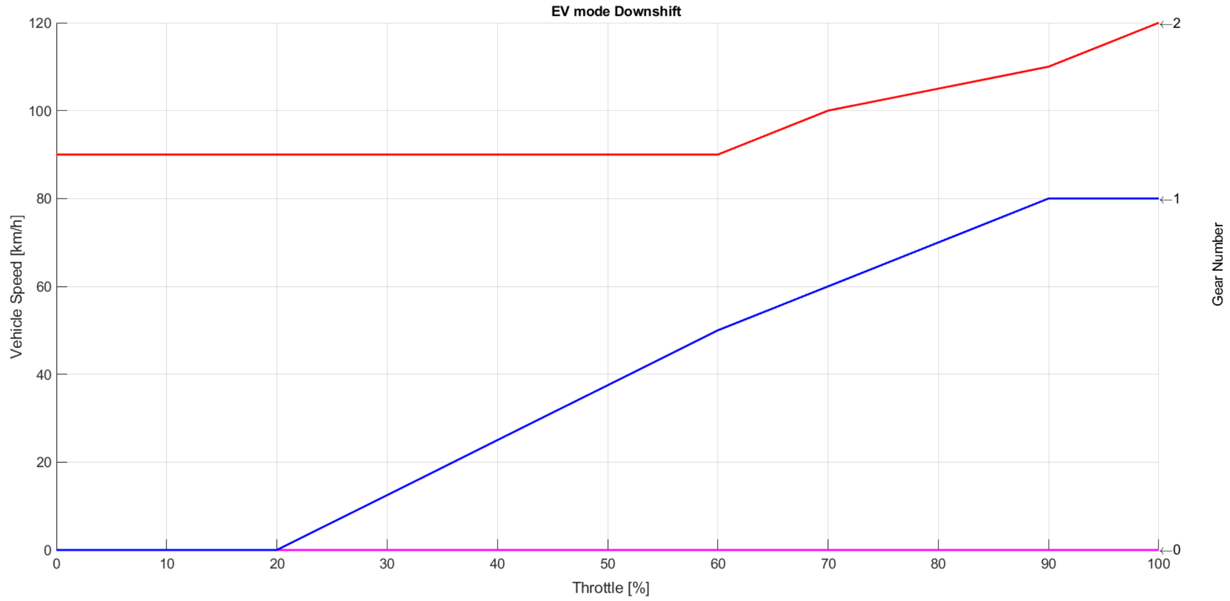


Figure 78 Gearbox Map for EV Mode Downshift Threshold

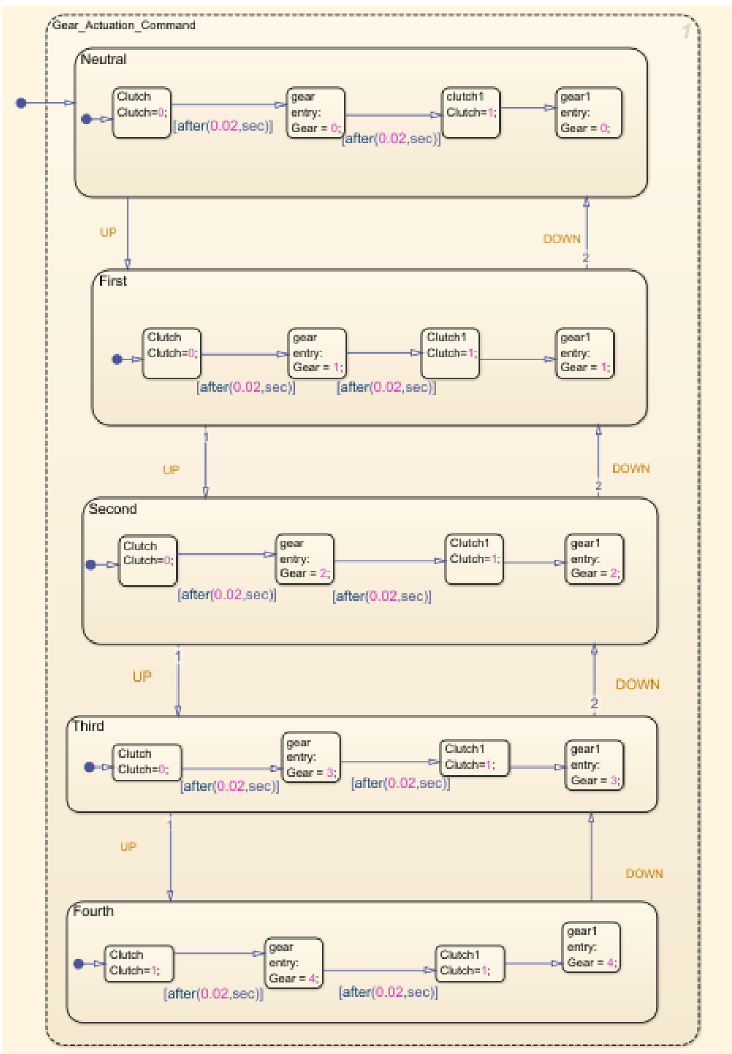


Figure 79 State flow Logic for Gear Shifting for ICE and Hybrid Modes

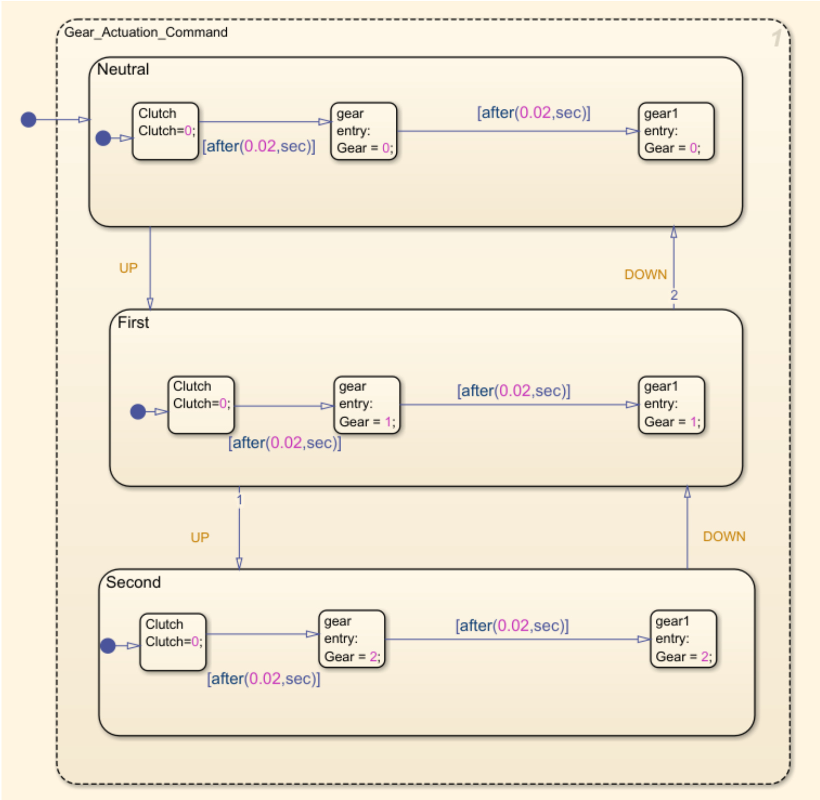


Figure 80 State flow Logic for Gear Shifting for EV Mode

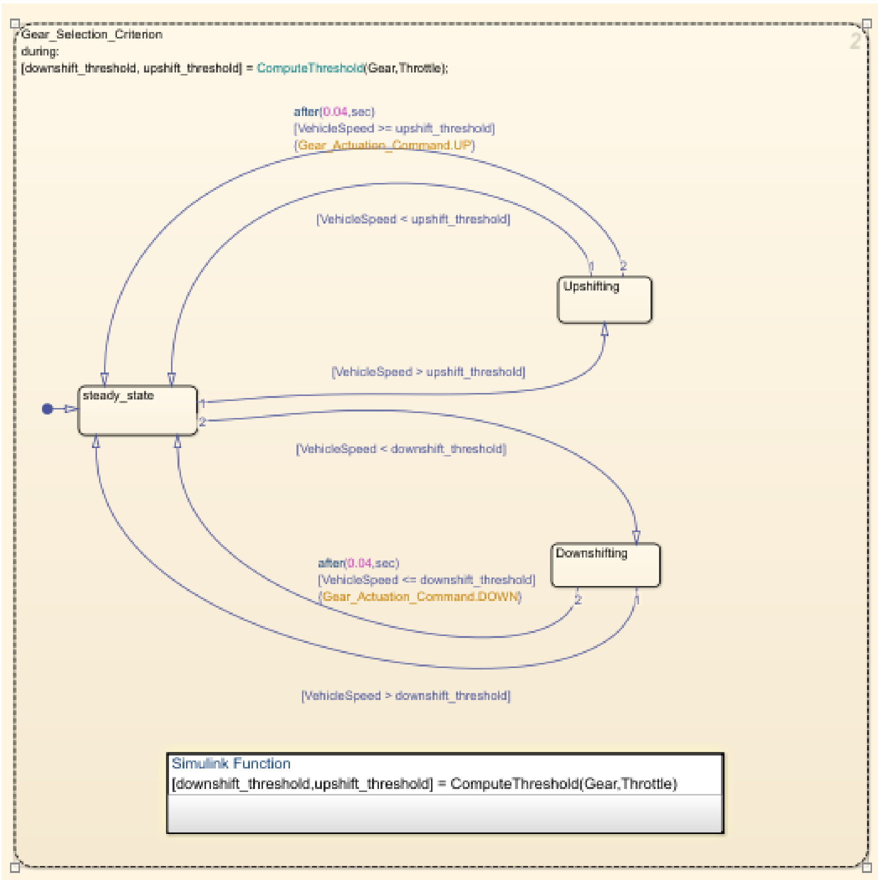


Figure 81 State Flow Logic for Gear Selection for all modes

The overall Gearbox Control Unit module, shown in Figure 82, was made in the SIMULINK environment and it uses the vehicle feedback and Acceleration command as throttle input to compute gear and clutch actuation and send out the Gear Actuation and the Clutch Actuation commands which are used by the actuators to perform the necessary actuation. The State Flow logic and the Control Unit Module are essentially the same for all three modes except for the implementation of only 2 Gears in EV mode and 4 Gears in ICE and Hybrid Modes.

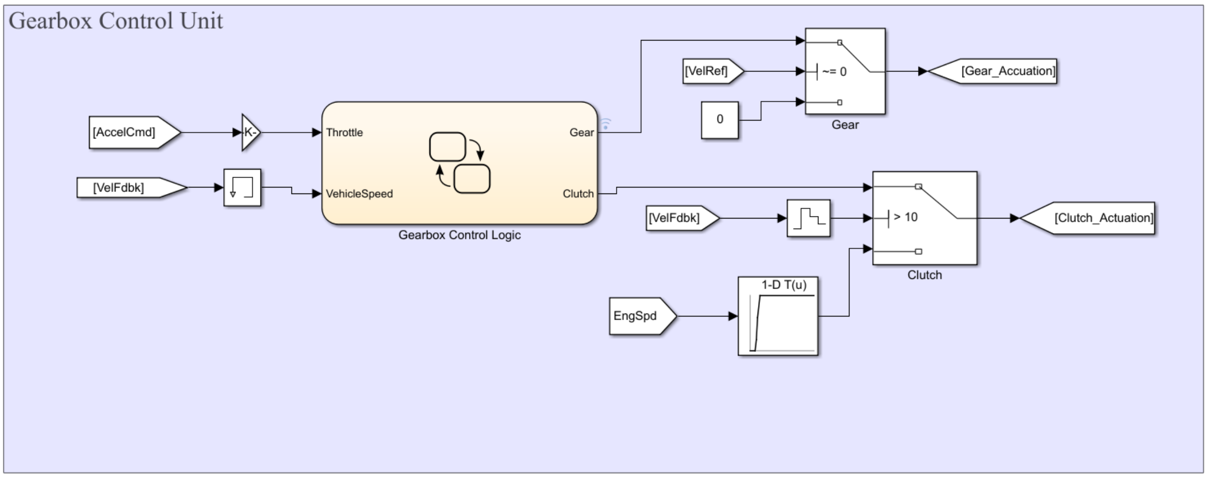


Figure 82 Gearbox Control Unit

3.4.2.4 Vehicle System

The vehicle system comprises the rear axle with two wheels and their respective disk brakes and the Vehicle body SIMSCAPE module. This vehicle body module represents the actual vehicle in longitudinal motion in the SIMULINK environment and this has all the vehicle parameters that would affect the dynamics of a vehicle under motion the data for this module has been provided by the company for use in the model. The output from this module is Vehicle speed which is used as a Feedback Velocity (VelFdbk) which is used by a lot of modules to compute the actuation for the clutch, acceleration and deceleration commands, etc.

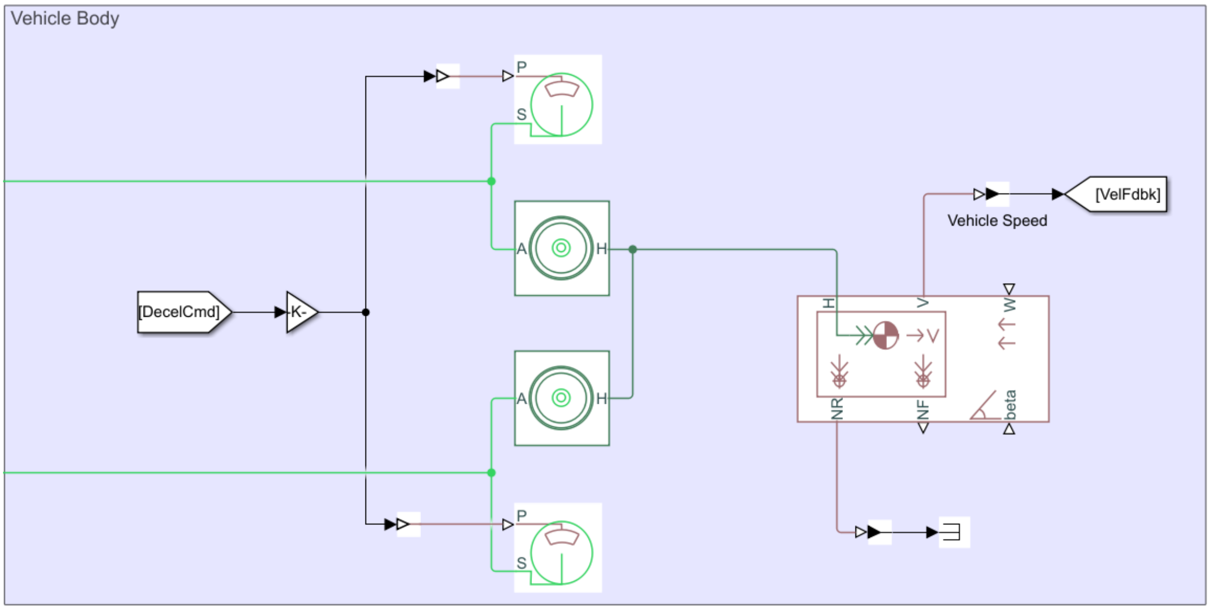


Figure 83 Vehicle Body Model

3.4.3 Results Discussion

The SIMULINK model was run for 1800 seconds which is the simulation time for a WLTP cycle that involves distinct levels of vehicle speeds ranging from low to very high reaching a maximum speed of just under 135 km/hr. The results are shown below for the 3 modes – ICE, Hybrid and EV Modes.

3.4.3.1 ICE Mode

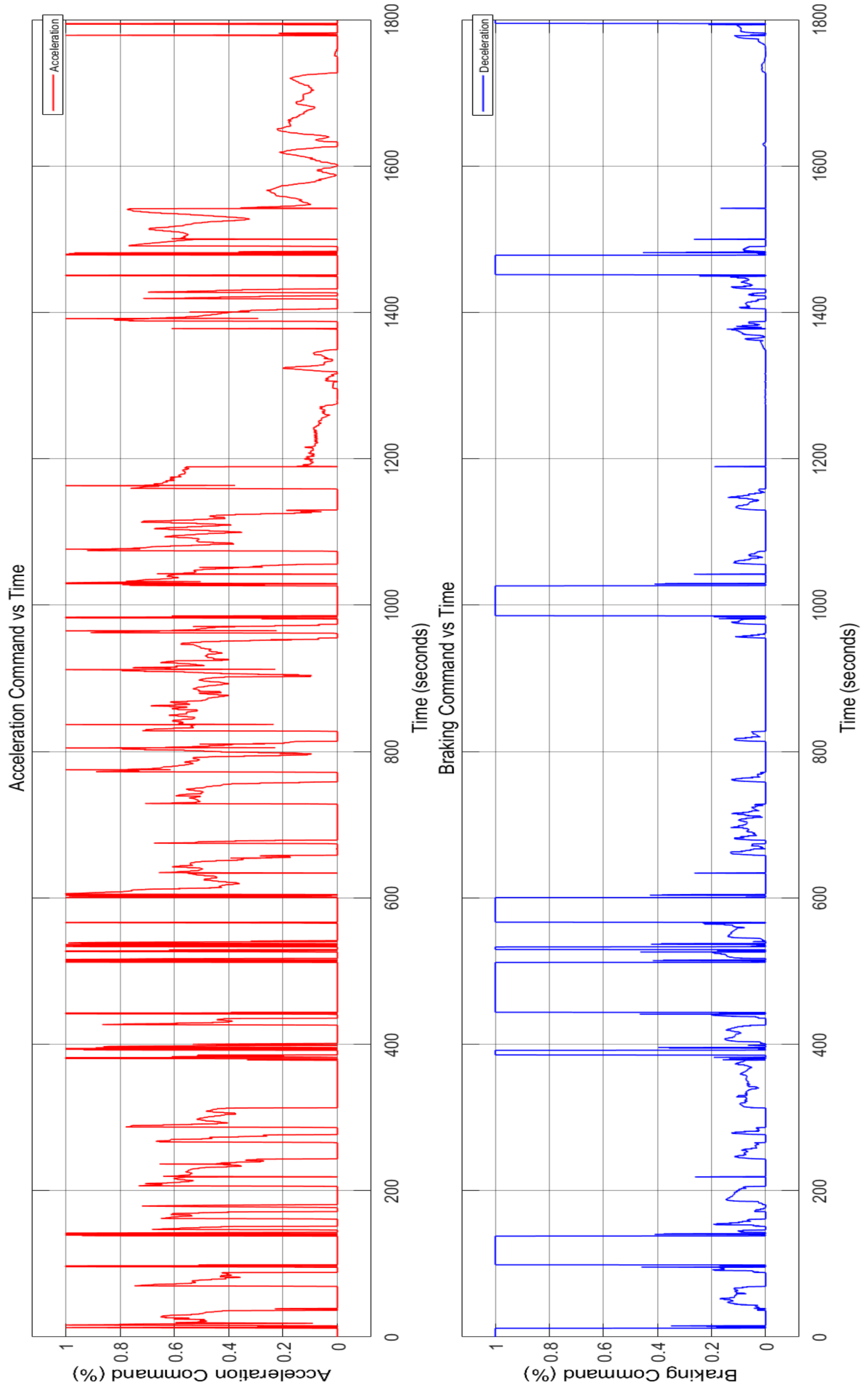


Figure 84 Acceleration and Braking Command Graphs for ICE Mode

Figure 84, shows the Acceleration and Braking/Deceleration graphs which correspond to the driver input about the WLTP cycle while running the vehicle in the ICE mode. We can infer from the graph the varying inputs from the driver to achieve the desired vehicle speeds.

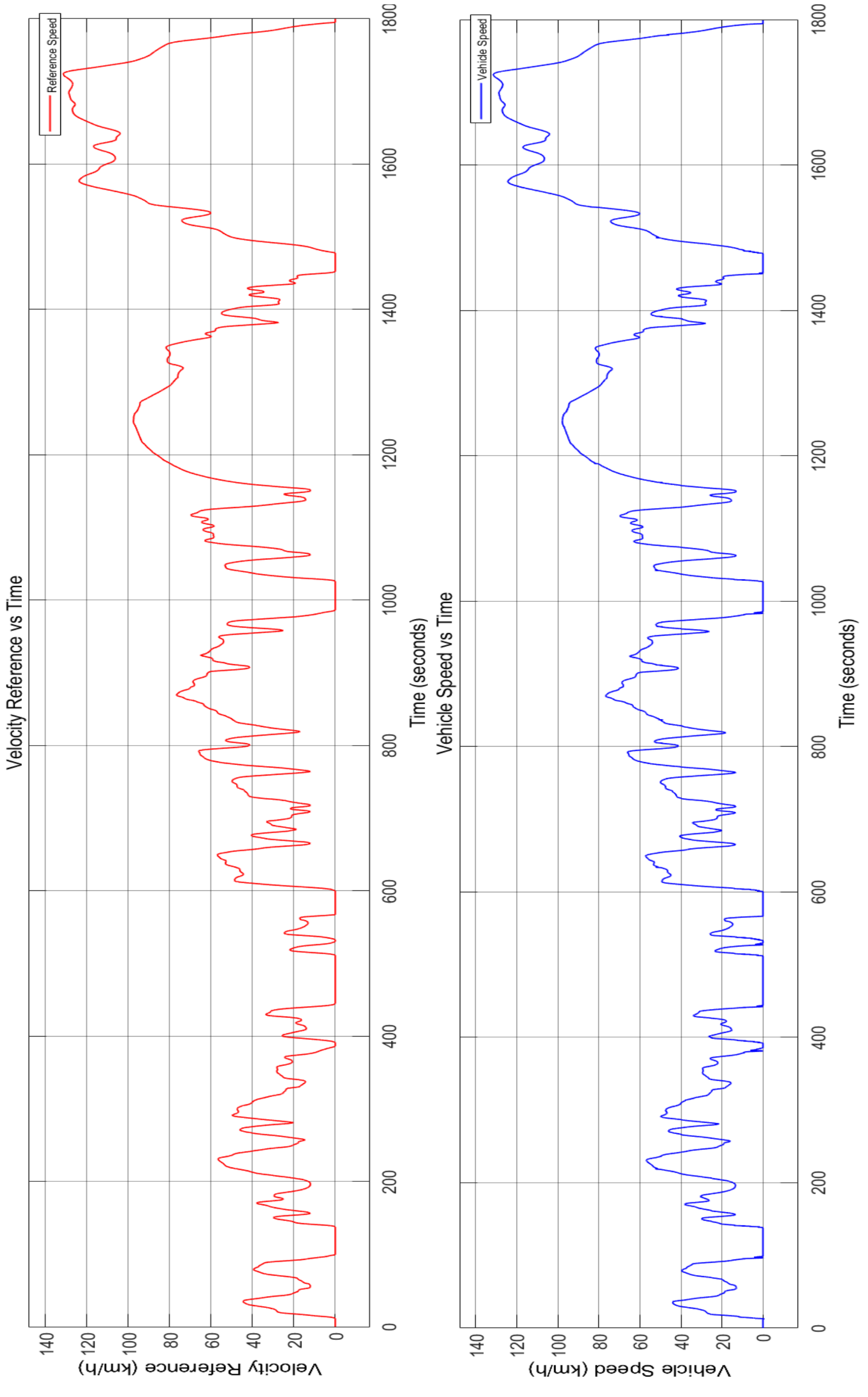


Figure 85 Vehicle Speed and Reference Speed Graphs for ICE Mode

Figure 85, shows the Vehicle Speed and Reference Speed graphs which correspond to the WLTP cycle while in ICE mode. We can infer from the graph that the output vehicle speed is matching the reference speed based on the inputs from the driver. These speeds are achieved by continuously varying the amount of throttle while comparing the reference speed from the WLTP cycle and the velocity feedback obtained from the wheels.

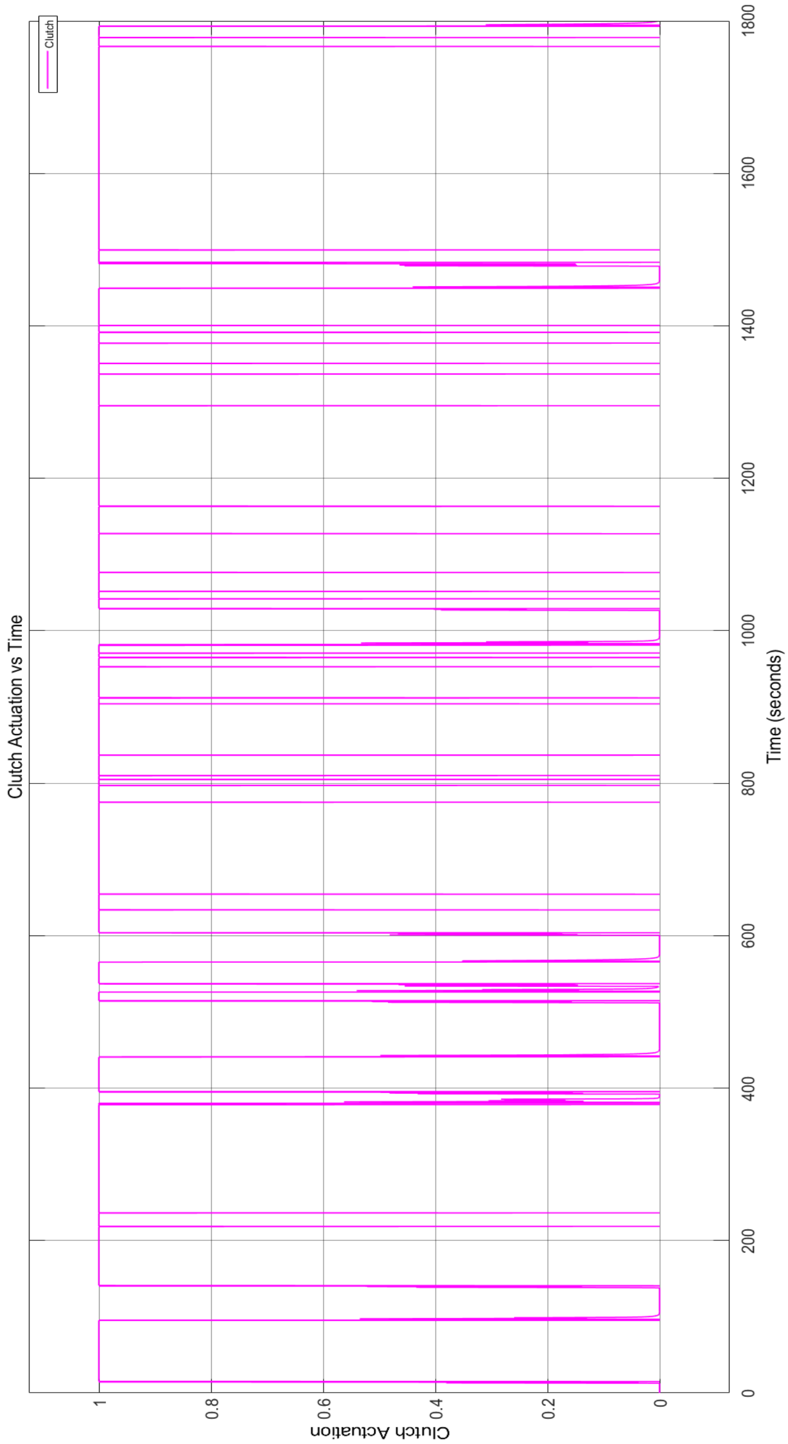


Figure 86 Clutch Command Graph for ICE Mode

Figure 86, shows the Clutch Actuation Command which is responsible for the engaging and disengaging of the Clutch during the gearshift, and we can infer that the logic used for this in the Gearbox Control Unit is working as intended and can simulate the clutch modulating while starting from a stop and during normal speeds, the clutch command is instantaneous and is according to the applied gear shifting logic.

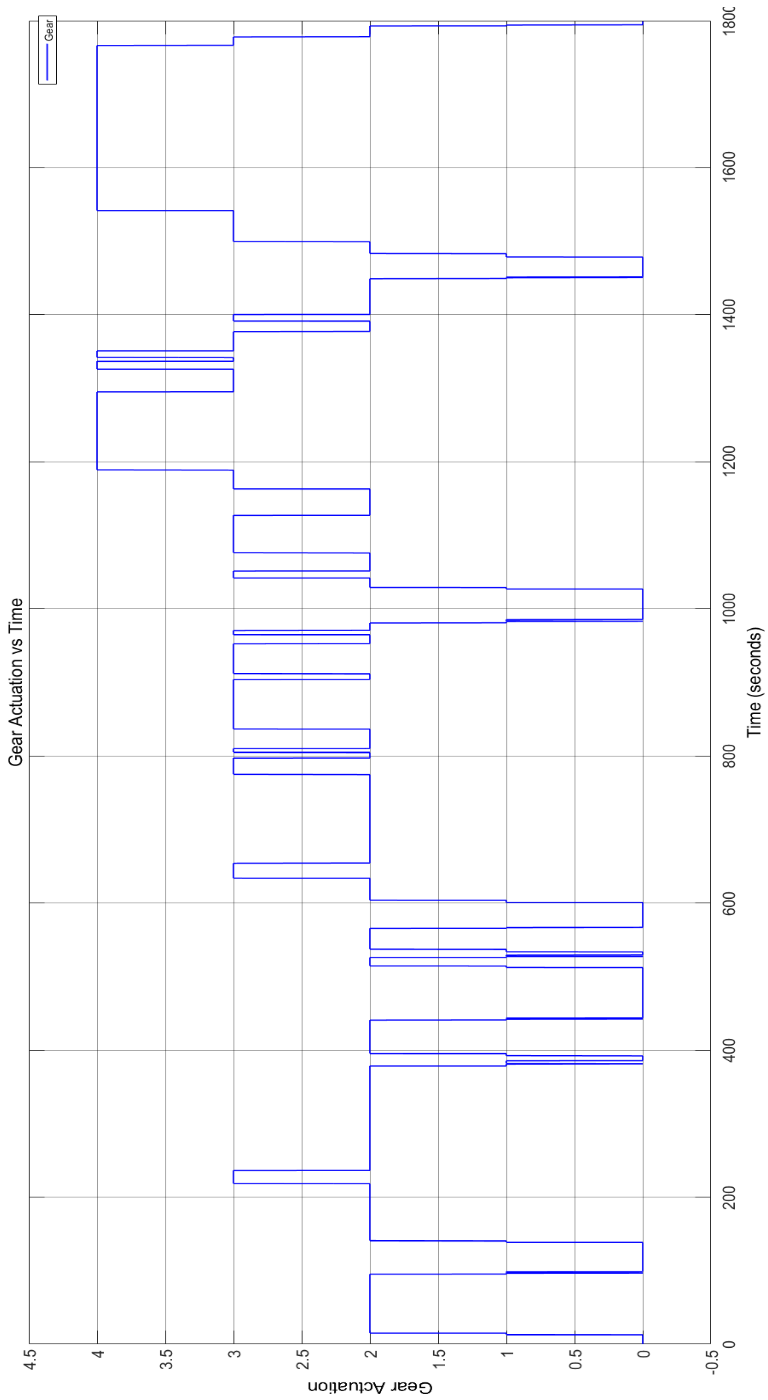


Figure 87 Gear Selection Command for ICE Mode

Figure 87, shows the Gear Selection command graphs which correspond to the driver throttle input and the feedback velocity. We can infer that the gear selection is following the gear maps programmed into the gearbox control unit to shift into the right gear while upshifting and downshifting while slowing down. The gear shifting times are following the State flow delay requests and can be modified to achieve the desired shifting times while considering the time of actuation based on the mechanical linkage.

3.4.3.2 Hybrid Mode

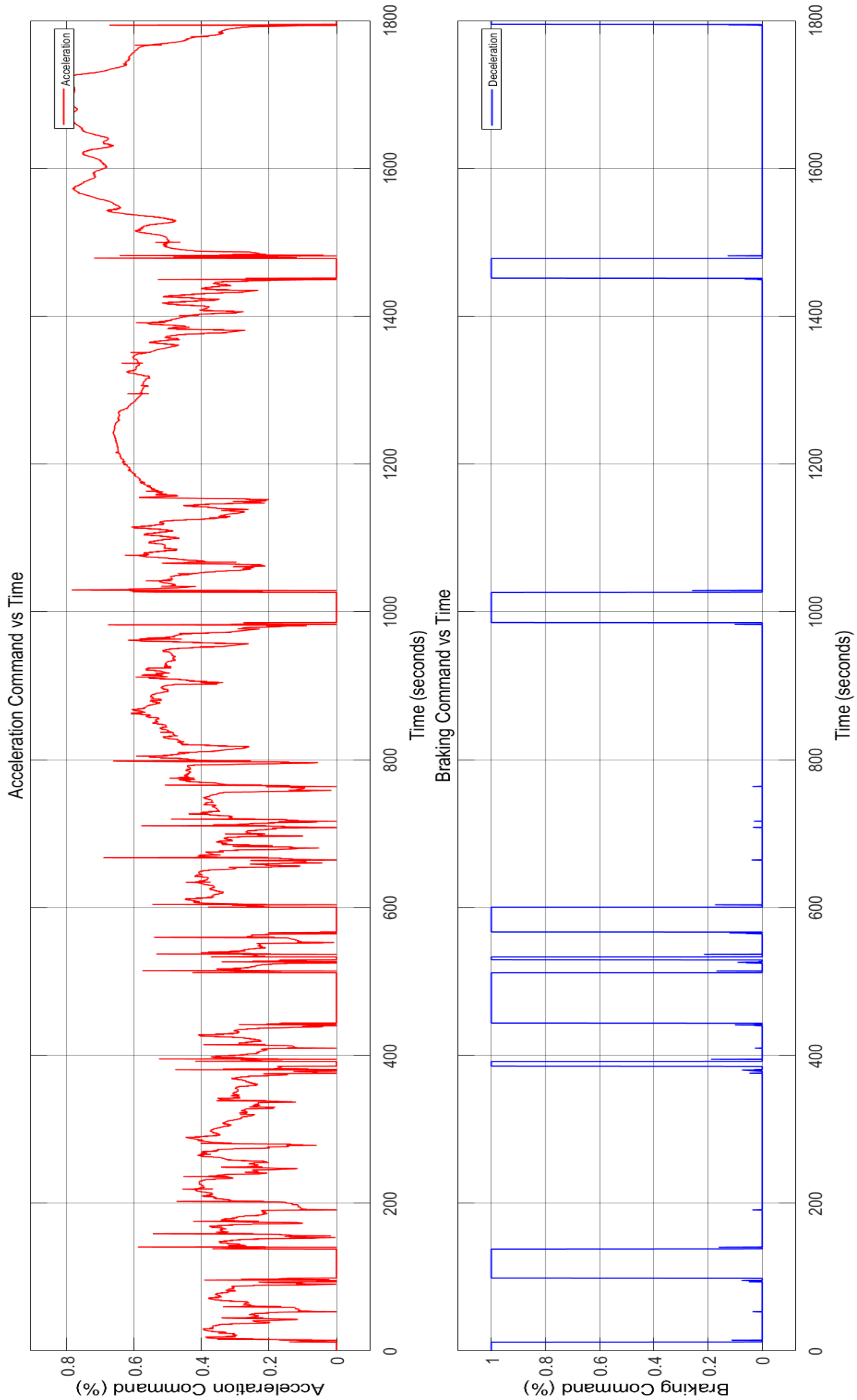


Figure 88 Acceleration and Braking Command Graphs for Hybrid Mode

Figure 88, shows the Acceleration and Braking/Deceleration graphs which correspond to the driver input to achieve the WLTP cycle Vehicle reference speed when the vehicle is in Hybrid Mode. Based on the reference speeds and the feedback velocity the acceleration and braking commands are varied to match the reference and feedback velocities.

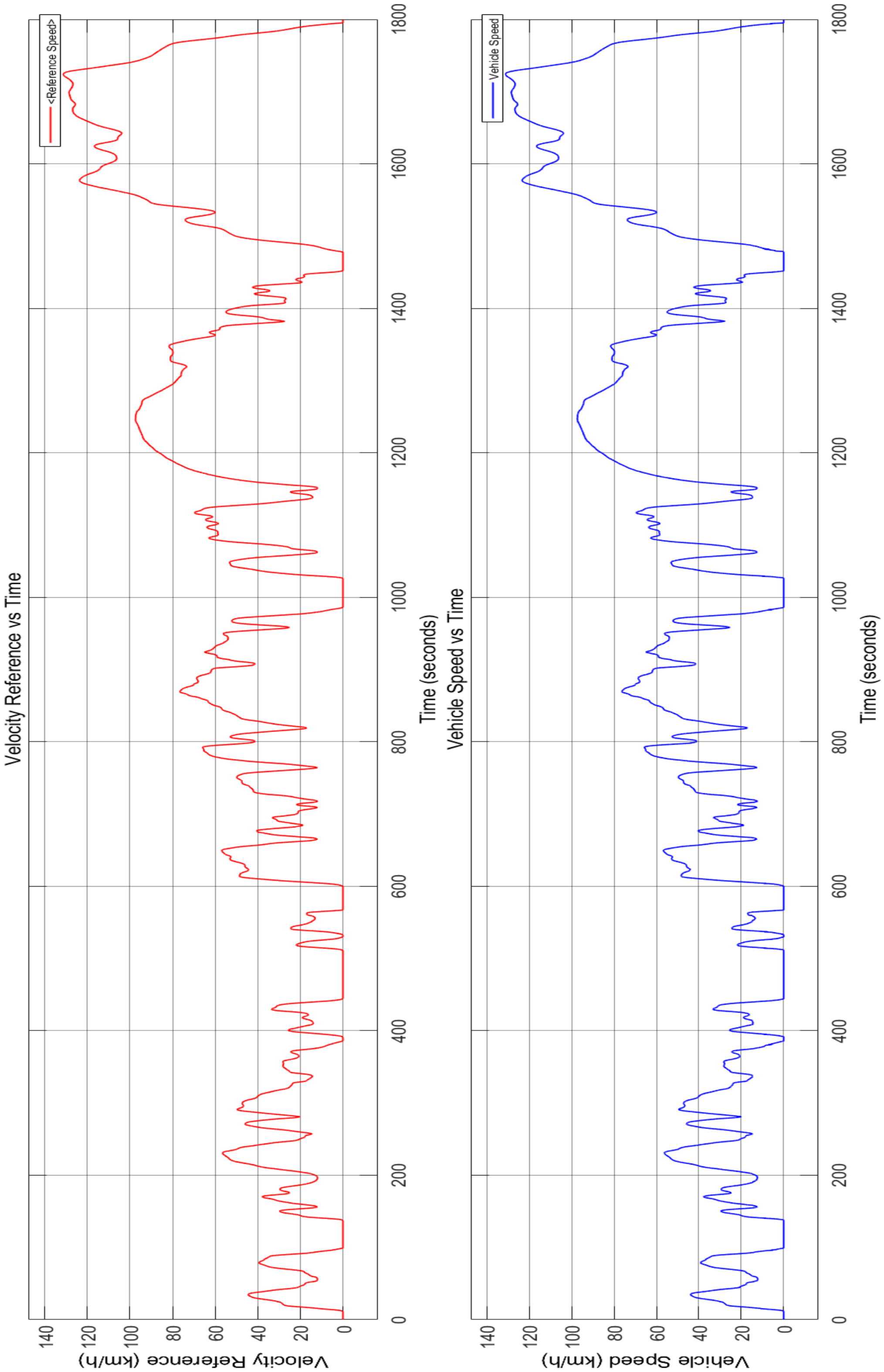


Figure 89 Vehicle Speed and Reference Speed Graphs for Hybrid Mode

Figure 89, shows the Vehicle Speed and Reference Speed graphs for Hybrid mode. We can infer from the graph that the output vehicle speed is matching the reference speed which is done by way of throttle control from the driver.

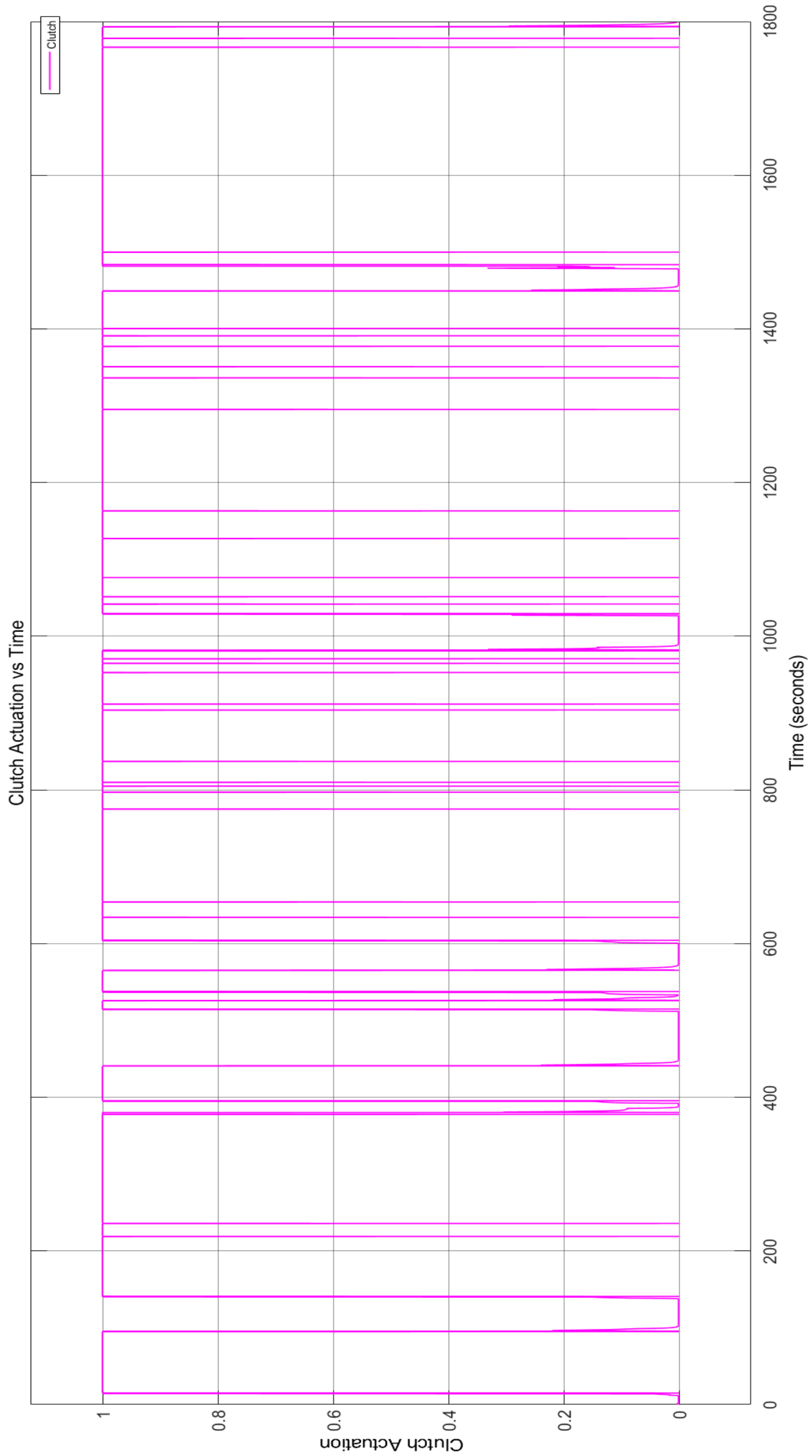


Figure 90 Clutch Command Graph for Hybrid Mode

Figure 90, shows the Clutch Actuation Command which is responsible for the engaging and disengaging of the Clutch during the gearshift, and from the graph, we can infer that the logic used for this in the Gearbox Control Unit is working as intended and can simulate the clutch modulating while starting from a stop (0 km/hr) and during normal speeds, the clutch command is instantaneous and is according to the applied gear shifting logic.

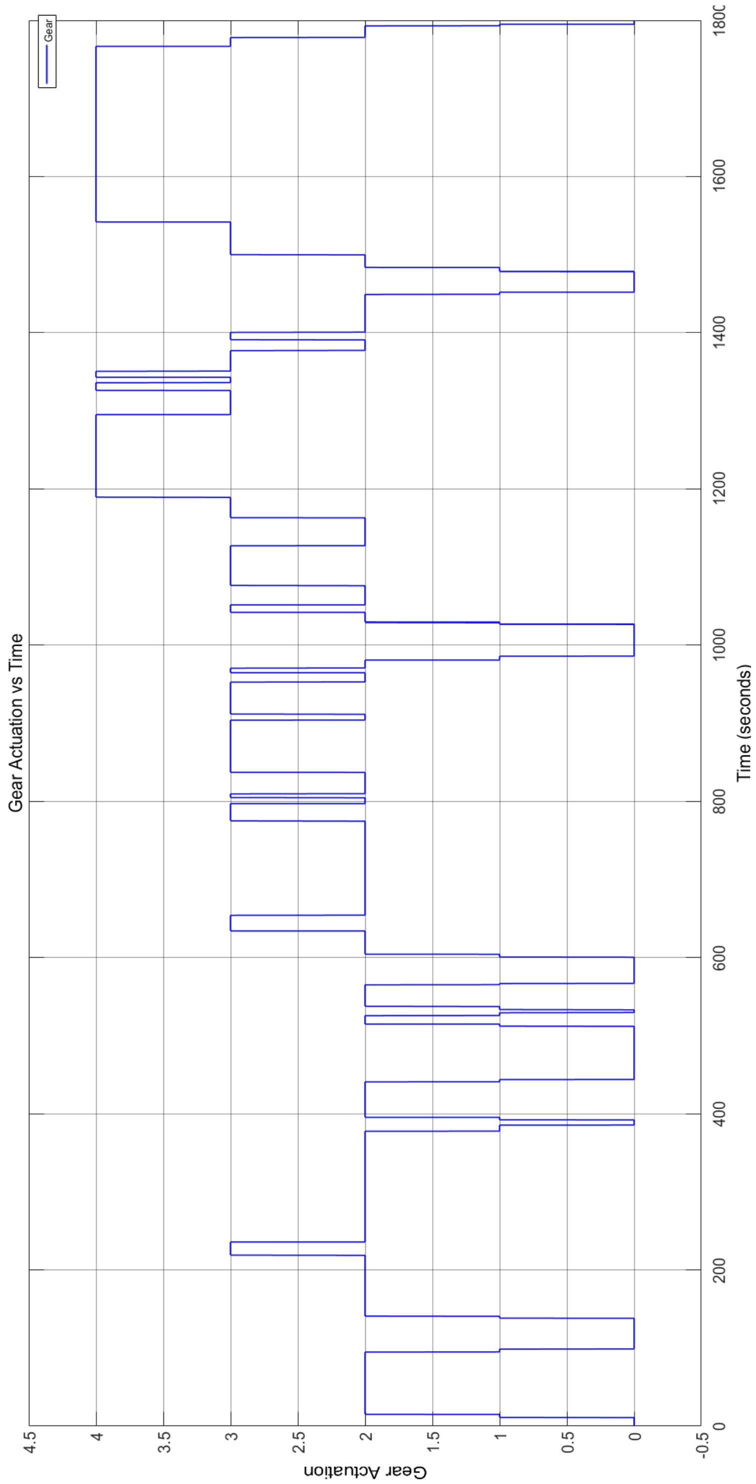


Figure 91 Gear Selection Command for Hybrid Mode

Figure 91, shows the Gear Selection command graphs which correspond to the driver input and the feedback velocity about the WLTP cycle. We can infer from the graph that the gear selection is following the gear maps programmed into the gearbox control unit to shift into the right gear while upshifting or downshifting while slowing down. As in hybrid mode, we have a possibility of an 8-speed and 4-speed configuration with the 4-speed being part of this project model, it can be easily modified into the 8-speed logic and can simulate the gear shifting according.

3.4.3.3 EV Mode

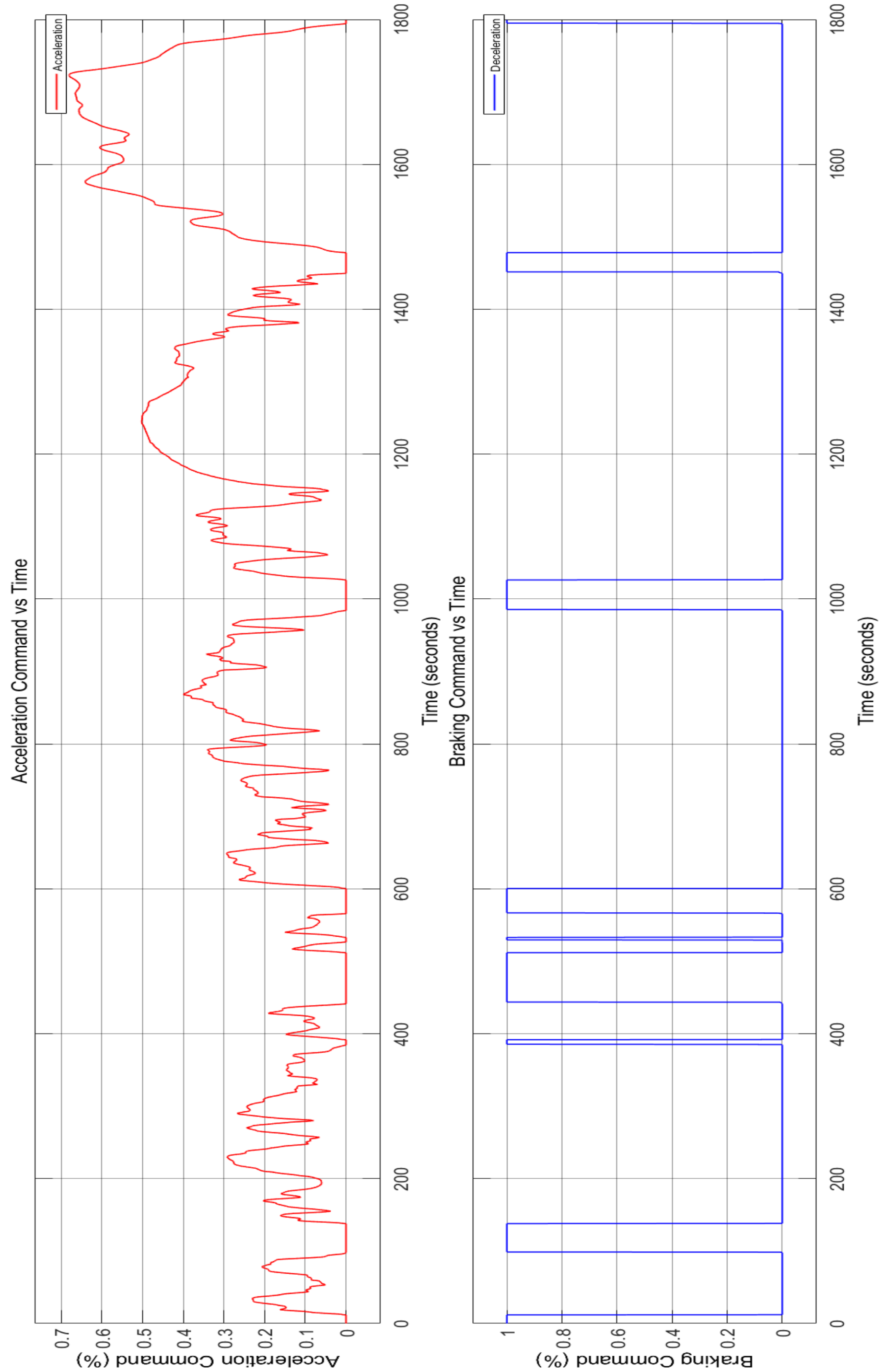


Figure 92 Acceleration and Braking Command Graphs for EV Mode

Figure 92, shows the Acceleration and Braking/Deceleration graphs which correspond to the driver input while in EV mode. We can infer from the graph the inputs provided follow the reference velocity and the throttle modulation is done to closely match the reference and actual vehicle velocities.

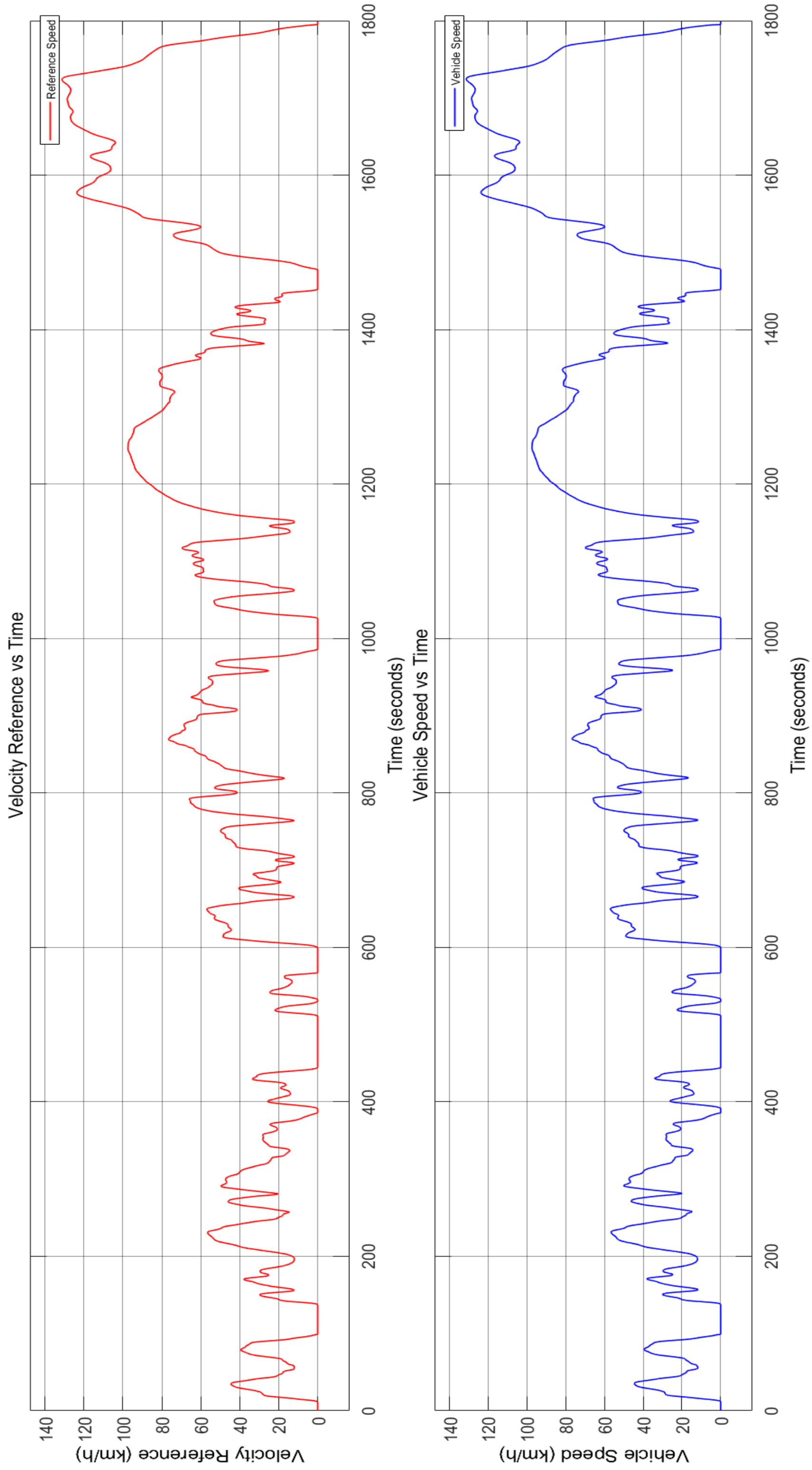


Figure 93 Vehicle Speed and Reference Speed Graphs for EV Mode

Figure 93, shows the Vehicle Speed and Reference Speed graphs in EV mode. We can infer from the graph that the output vehicle speed is matching the reference speed that the driver uses to provide input to the system.

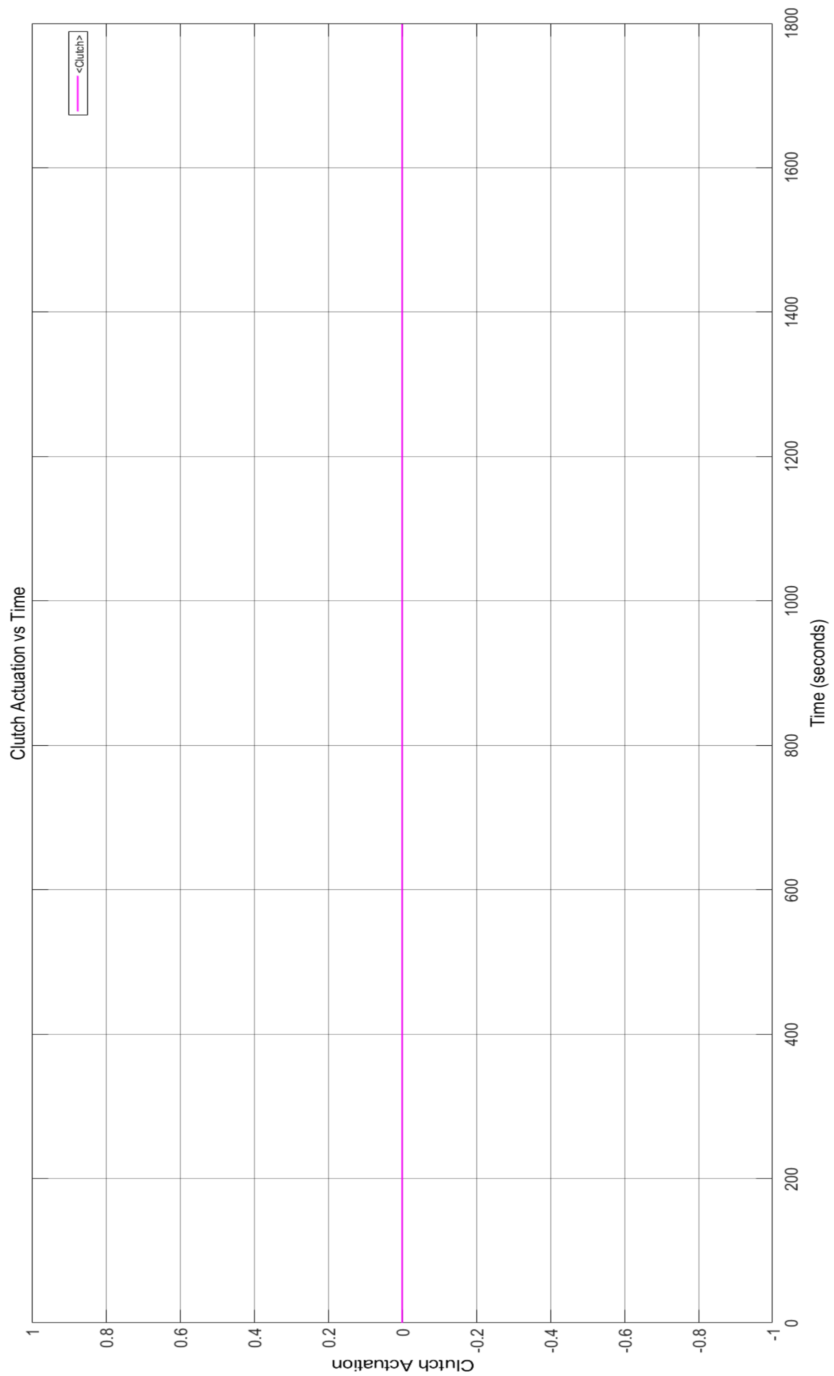


Figure 94 Clutch Command Graph for EV Mode

Figure 94, shows the Clutch Actuation Command while in EV mode. It is necessary to have a clutch command while in EV mode because the output shaft is common for both the EV and ICE gear sets thus needing to disengage the clutch.

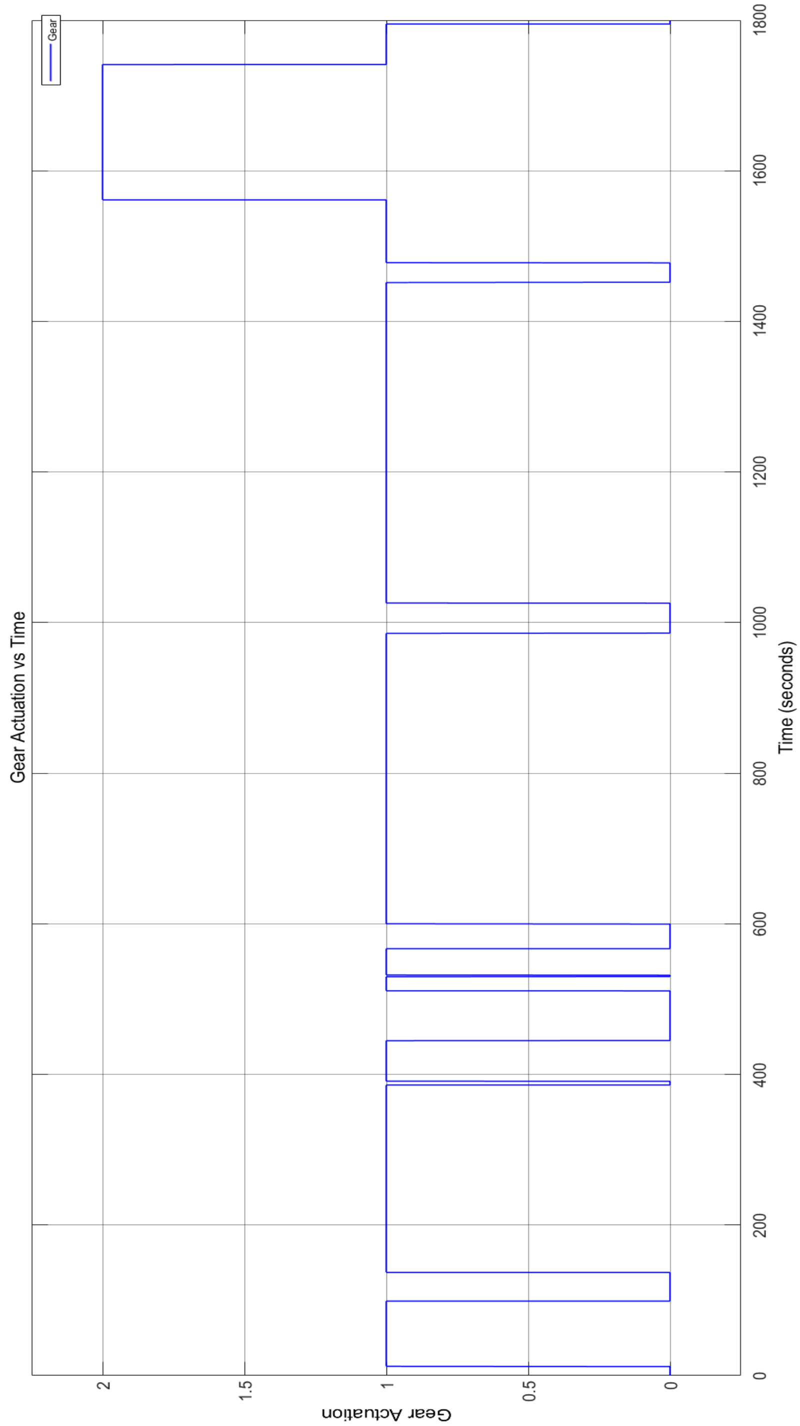


Figure 95 Gear Selection Command for EV Mode

Figure 95, shows the Gear Selection command graphs which correspond to the driver input and the feedback velocity in EV mode. We can infer from the graph that the gear selection is following the gear maps programmed into the gearbox control unit to upshift and downshift.

4 Conclusion

This thesis deals with the development of a clutch and gear actuation system with the use of pneumatic components and the primary aim was to decide the pneumatic components that would be required to use for the system. The choice of a pneumatic system for the clutch actuation mechanism was mainly due to the presence of a pneumatic compressor and an air tank in the car and other key components in the vehicle like the rear active spoiler and suspension working on compressed air.

The calculation was carried out to understand the required amount of pedal force at which the clutch can be actuated at the working pressure of 7 bar as the compressed air available for this system in the vehicle was 7 bar. The choice for the pneumatic cylinder was done based on the pedal force required and market research on the various types of pneumatic cylinders. The Artec PDM.100.060 was chosen for our application. The pneumatic controller for the same was chosen from the manufacturer Haufner and two models were chosen of which one was a 5/2-way mono-stable valve and the other was a 5/3-way bi-stable valve which will both be used for the testing. The positioning sensor chosen was from the manufacturer SICK and it is a Magnetic position sensor with 96mm of sensing range and gives an analogue output.

The second part of the project was to define a test bench where a basic test rig was made on Solidworks with the pneumatic cylinder, master cylinder, and the custom pushrod for the same was modelled. Once the components are manufactured the clutch logic and the gearbox actuation logic can be tested by using a wide range of Hardware-in-loop (HIL) testing equipment. The actuation timings, the accuracy of actuation, and the force required for the actuation can be measured and this would give us an idea of how the system works and what changes need to be made either in the hardware or in the logic implementation to achieve the right actuation for the clutch.

The third part of the project was to propose a logic of operation for the clutch actuation and the gearbox actuation and simulate the actuation mechanisms for the possible ICE, Hybrid and EV modes on a driving cycle on SIMULINK. The WLTP cycle was chosen for this purpose as it represents the most common driving cycle in Europe and some other parts of the world and that makes the best possible city driving cycle to test the logic for lower speed gear and clutch actuation. The gear maps corresponding to these operating conditions are observed to work as planned with the logic used in the SIMULINK model and this confirms that these maps can be implemented in the vehicle.

The main outcome of the project was that it paved the way to automating the clutch actuation for a gearbox and the implementation of pneumatics for this automation. The simulations run for the testing of the working logic to actuate the pneumatic actuators showed that the approach being followed was correct and can be used. The test bench proposed for the testing of the actuation system can be made and used to understand the working of the system and verify if it can perform as intended and if any mechanical modifications are required to get the system to work and give the necessary clutch actuation. The results of the initial implementation of the gearbox control logic on the Arduino board showed the required functionality of the system with clutch and gear actuation based on the user input was working and this was later implemented in a HiL test bench for testing and

5 Future Work

The research work involved in this project paves way for further implementation of the clutch actuation system along with the gearbox actuation. The logic made in the SIMULINK simulation can be used to develop Hardware-in-loop (HIL) testing equipment to test out various pneumatic equipment being used for the clutch actuation system and the gearbox. The maps available for the gearbox and clutch actuation can be further used to develop a gearbox control unit that can be implemented in the prototype.

The testing involved in the verification of the chosen pneumatic system would ensure that the finalized model would have been well tested in the test benches and would ensure the perfect functionality in the prototype.

The scope of this project spans a wide range of applications with pneumatic clutch and gearbox actuation systems for high-performance vehicles as faster shifting times and extremely low latency is highly desirable. The company further plans on developing a control system for the pneumatic actuation systems to be implemented in the vehicle with a custom-developed control module based on the logic simulated in SIMULINK and after intensive testing on various test benches. The future work would entail the development of a pneumatic system with redundancy, high controllability and accuracy, and a robust system.

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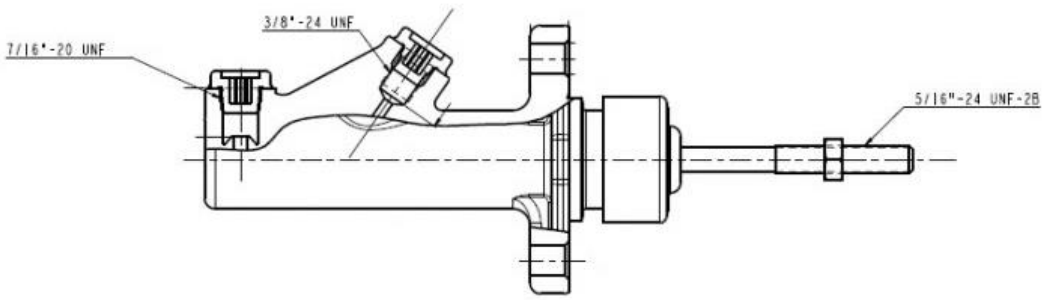
Appendix

Appendix I – Master Cylinder Technical Datasheet



SPECS

Master Cylinder Dimensions	
Bore Size	**3/4"**
Area (in^2)	**0.440**
Stroke	**1.40**
Volume (cu in)	**0.61**
Master Cylinder Description	
Type	**Single Outlet**
Outlets	**1**
Material	**Aluminum**
Finish	**Bare**

Appendix II – Clutch System Technical Datasheet

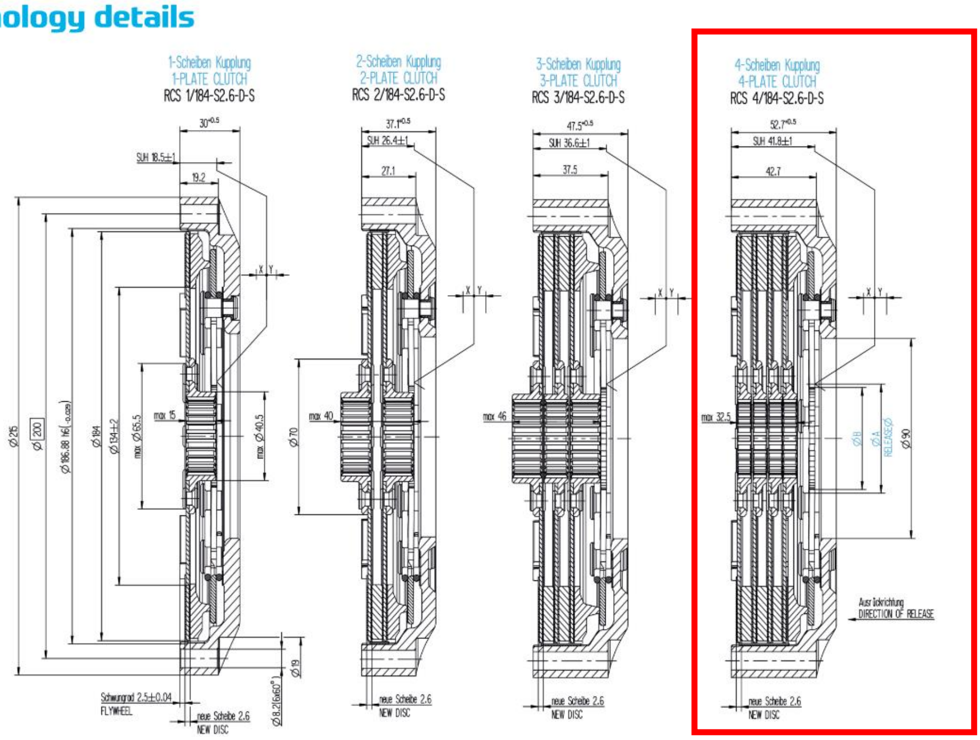
RCS 184-S2.6-D-S-XX

If you are looking for a clutch that can easily cope with the harsh demands of modern day motorsports, then you have found what you are looking for. This motorsports clutch is an excellent synthesis of robustness and reliability whilst maintaining its competitive character, due to its favourable mass and inertia properties. Keeping this in mind and knowing that it is used in race winning drivetrains, it additionally provides high thermal stability making it fit for off-track race applications, absorbing the unexpected misuse events.

Examples of application: Rally, circuit racing, touring cars

Advantages:Two spring loads available. Two release diameters. Protection plates forhousing increases wear resistance.

Technology details

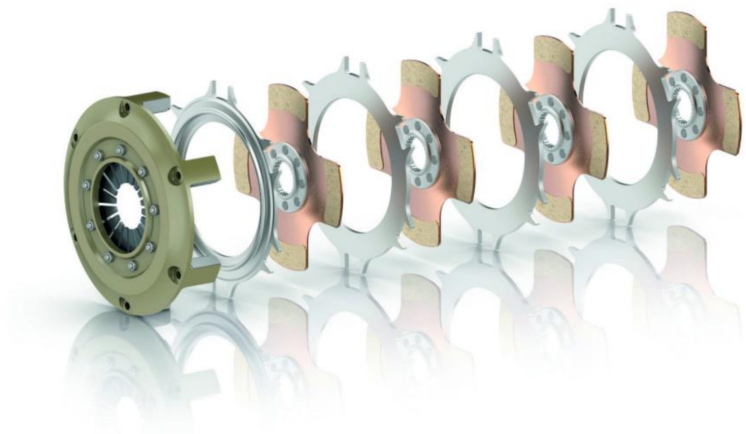


Technical specifications
RCS 184-S2.6-D-S-XX

Selection criteria			Technical specifications						Purchase order number			
Clutch torque [Nm]	Dimension Ø A Release Ø [mm]	Dimension Ø B Spring inner Ø [mm]	Release force max. [N]	Wear range [mm]	Release travel X [mm]	Wear travel Y [mm]	Mass [kg]	Inertia [kgm²]	Housing Part number	Pressure plate Part number	Qty intermediate plate 003019000336 Qty	Qty driven disc* Qty
1-Disc Clutch												
385	49	46	2400	1.5	5.0 +0.5	5.0	2.195	0.01255	003072999543	003002001897	0	1
654	49	46	4300	1.5	5.0 +0.5	5.0	2.247	0.012568	003072999542	003002001897	0	1
2-Disc Clutch												
769	49	46	2400	1.5	5.0 +0.5	5.0	3.055	0.017199	003072000130	003002001897	1	2
1308	49	46	4300	1.5	5.0 +0.5	5.0	3.105	0.017218	003072000125	003002001897	1	2
1308	44	37	3900	1.5		5.0	3.116	0.017167	003072000139	003002001897	1	2
3-Disc Clutch												
1154	49	46	2400	1.5	5.0 +0.5	5.0	4.06	0.022262	003072000134	003002001898	2	3
1962	49	46	4300	1.5	5.0 +0.5	5.0	4.111	0.022852	003072000126	003002001898	2	3
1962	44	37	3900	1.5	5.0 +0.5	5.0	4.172	0.023029	003072000140	003002001898	2	3
4-Disc Clutch												
2616	49	46	4300	1.5	5.0 +0.5	5.0	4.937	0.027736	003072000127	003002001897	3	4

More clutch torque without safety margin!
*) The order number for driven discs, according to the required hub spline configuration and spline selection can be found on the following pages.

Selection of driven disc order number:
- Select the suitable hub configuration
- The order number of the driven discs is defined with the ZF Code / spline size application
- A configuration example can be found on pages V13 & V14

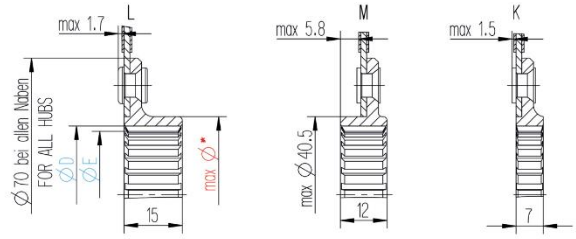


Hub configuration RCS 184-S2.6

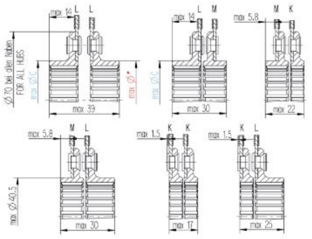
Flywheel Side

Gearbox Side

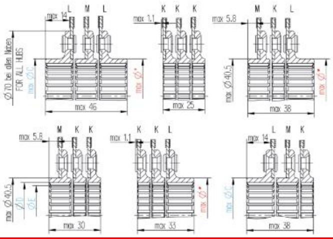
1-Disc Clutch



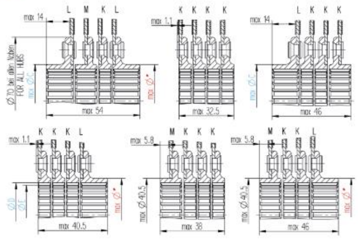
2-Disc Clutch



3-Disc Clutch

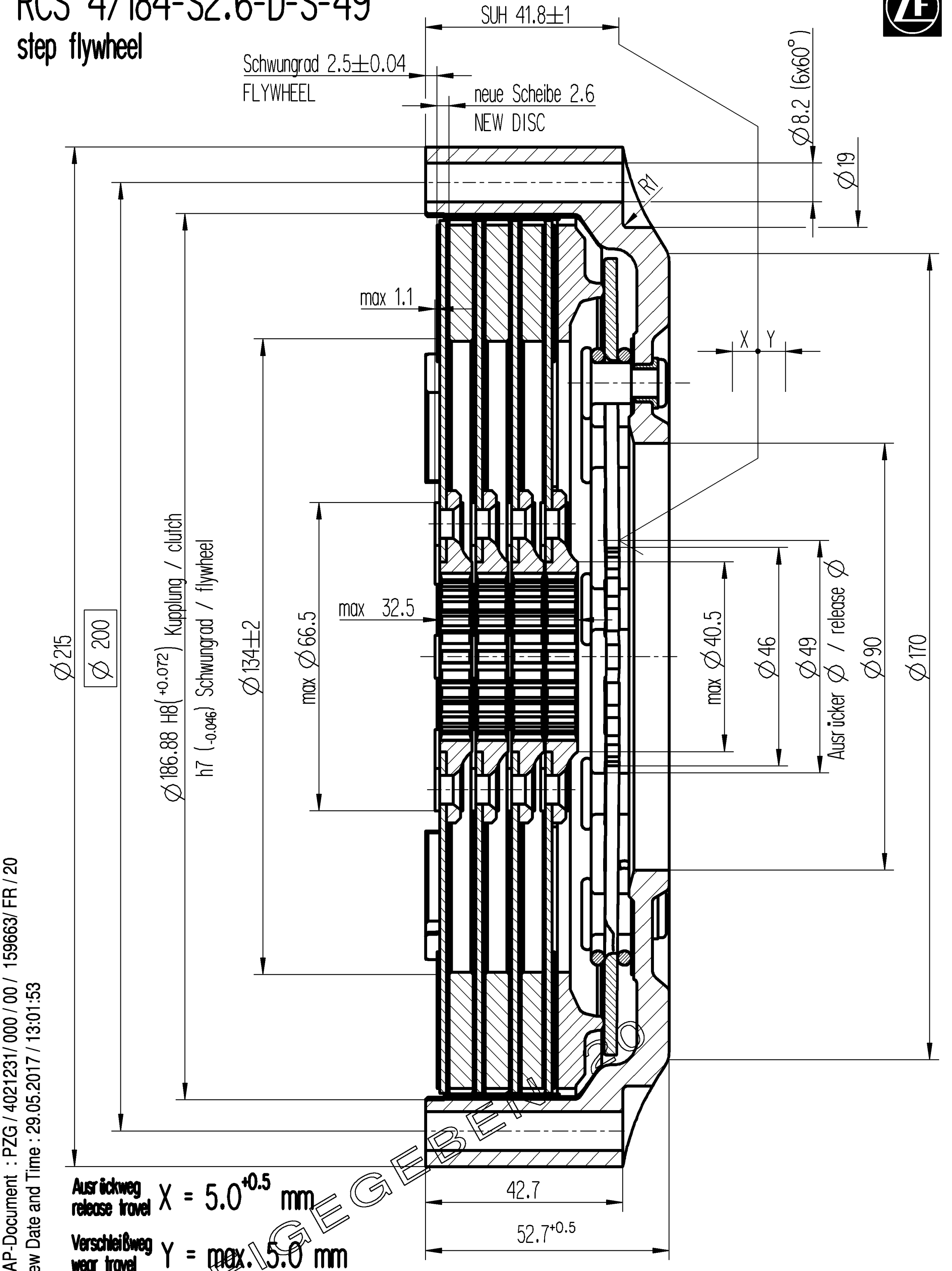


4-Disc Clutch



RCS 4/184-S2.6-D-S-49

step flywheel



Ausrückweg release travel $X = 5.0^{+0.5}$ mm

Verschleißweg wear travel Y = max. 5.0 mm

KPN SAP-Document : PZG / 4021231 / 000 / 00 / 159663 / FR / 20
Plot/View Date and Time : 29.05.2017 / 13:01:53

Stand/issued: Juli 2013
000004021231

Appendix III – Pneumatic Slave Cylinder Technical Datasheet

Slave Cylinder

ZF Race Engineering offers a specially developed and designed slave cylinder (CSC = Concentric Slave Cylinder) for push-type actuation of RCS racing clutches.

Advantages

Using housings, which are produced as a single component, we achieve the highest possible strength and prevent leakage.

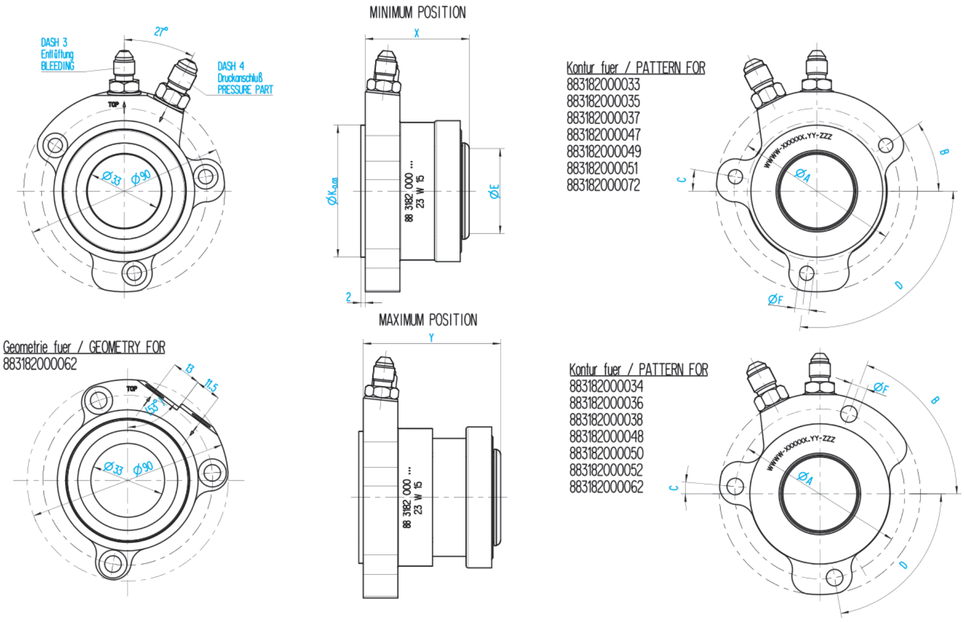
The release bearing is designed for high rpm levels of race engines. The release diameter of the slave cylinder is specially adjusted to the release diameter of ZRE RCS clutches and is available in three different diameters:
Ø 38 mm | Ø 44 mm | Ø 49 mm

ZRE slave cylinders are designed to generate the optimum friction in the system. This has been achieved by a special coating, as well as an optimal friction sealing system.

In general racing clutches have higher release forces. As a result of that, the pedal forces that the drivers must exert are also considerably higher. However, the slave cylinder developed specifically for racing clutches, features a higher hydraulic volume, which lowers the pedal forces for the driver.The cross-sectional area of this slavecylinders is 820.7 mm

ZRE slave cylinders are available in different connection geometries and in two different working ranges (strokes).

Technology in detail



A Pitch circle fixation holes
B/C/D Angle fixation holes
E Release diameter (contact diameter)
F Diameter fixation holes
K Centering diameter
X CSC in compression
Y CSC in rebound
***)** Hydraulic - adapter not included

Slave Cylinder Variations

Part number	A [mm]	B [°]	C [°]	D [°]	E [mm]	F [mm]	K [mm]	X [mm]	Y [mm]	Stroke [mm]	Replacement for
883182000033	74	33.5	10	97	38	6.6	58.99	46.5	61.5	15.0	---
883182000034	76	70	5	80	38	7.2	62.65	46.5	61.5	15.0	---
883182000035	74	33.5	10	97	44	6.6	58.99	46.5	61.5	15.0	---
883182000036	76	70	5	80	44	7.2	62.65	46.5	61.5	15.0	---
883182000037	74	33.5	10	97	49	6.6	58.99	47.0	62.0	15.0	---
883182000038	76	70	5	80	49	7.2	62.65	47.0	62.0	15.0	---
883182000072	74	33.5	10	97	38	6.6	58.99	50.0	62.0	15.0	883182999546
883182000047	74	33.5	10	97	38	6.6	58.99	43.3	55.3	12.0	---
883182000048	76	70	5	80	38	7.2	62.65	43.3	55.3	12.0	---
883182000049	74	33.5	10	97	44	6.6	58.99	43.3	55.3	12.0	---
883182000050	76	70	5	80	44	7.2	62.65	43.3	55.3	12.0	---
883182000051	74	33.5	10	97	49	6.6	58.99	43.8	55.8	12.0	---
883182000052	76	70	5	80	49	7.2	62.65	43.8	55.8	12.0	---
883182000062 *)	76	70	5	80	49	7.2	62.50	43.8	55.8	12.0	---

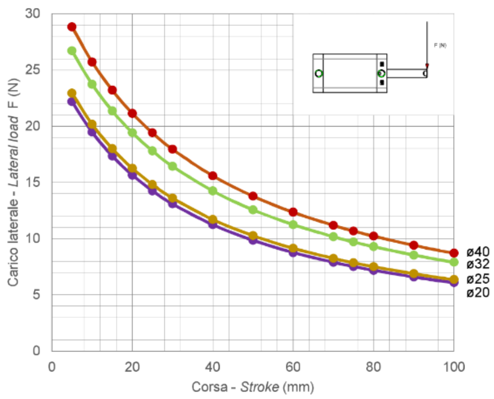
Appendix IV – Pneumatic Cylinder Technical Datasheet

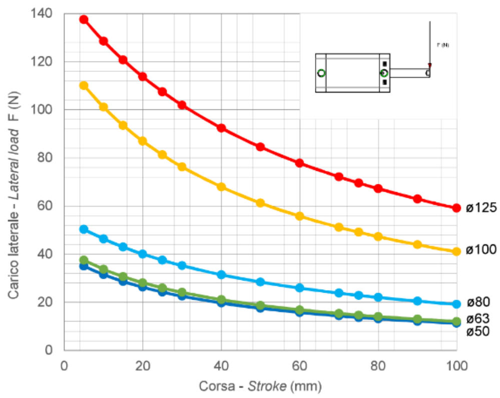
SERIE P

PDM PSM PSEM

DIAGRAMMA TEORICO CARICO AMMISSIBILE

THEORETICAL ALLOWABLE LOAD

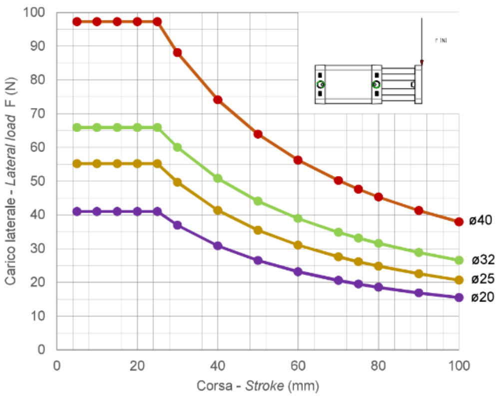


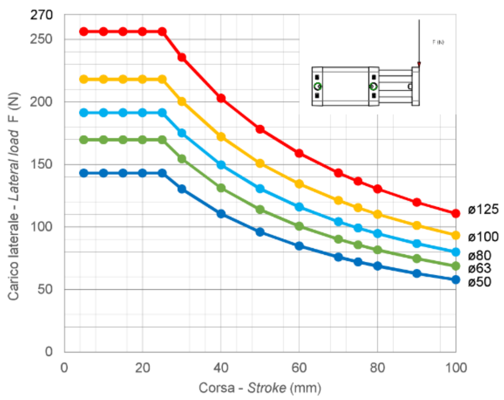


PDMA

DIAGRAMMA TEORICO CARICO AMMISSIBILE

THEORETICAL ALLOWABLE LOAD





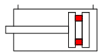
80

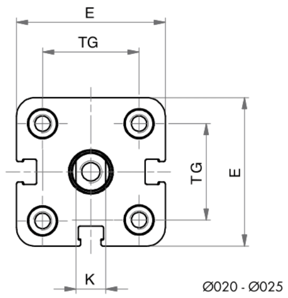
DOPPIO EFFETTO MAGNETICO

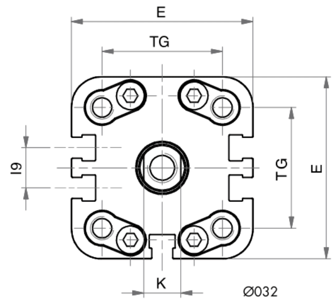
PDM

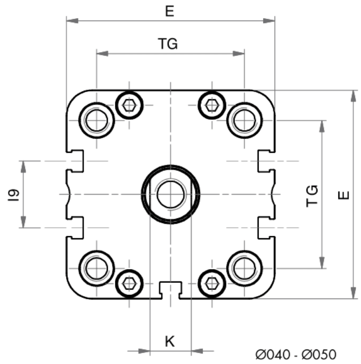
DOUBLE ACTING MAGNETIC

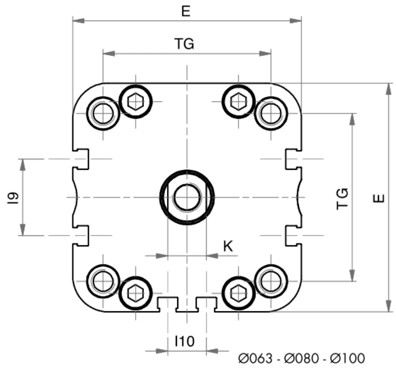
SERIE P

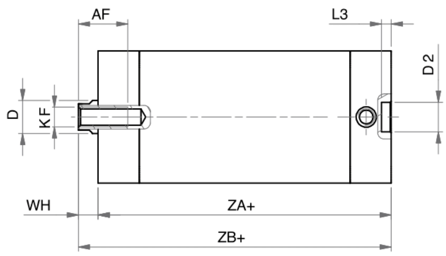


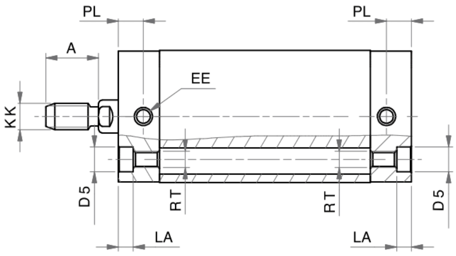












DOPPIO EFFETTO MAGNETICO

DOUBLE ACTING MAGNETIC

DIMENSIONI - *DIMENSIONS*

ø	020	025	032	040	050	063	080	100	125
A	16	16	19	19	22	22	28	28	40
AF	15	15	15	15	17	17	20	22	25
ø D	10	10	12	12	16	16	20	25	25
ø D2	9	9	9	9	12	12	12	12	12
ø D5	7,5	7,5	9	9	10,5	10,5	13,5	13,5	-
E	36	40	49	54,5	65,5	77	95,5	113,5	135
EE	M5	M5	G1/8	G1/8	G1/8	G1/8	G1/8	G1/8	G1/4
I9	-	-	10,8	12,8	21	25,8	30	50	50
I10	-	-	-	-	-	13	18	35	50
K	8	8	10	10	13	13	17	22	22
KF	M6	M6	M8	M8	M10	M10	M12	M12	M16
KK	M8	M8	M10x1,25	M10x1,25	M12x1,25	M12x1,25	M16x1,5	M16x1,5	M 20x1,5
LA	4,5	4,5	5	5	5	5	3	3	-
L3	3	3	3	3	4	4	4	4	4
PL	7,5	7,5	7,5	8	8	7,5	8	10,5	10,5
RT	M5	M5	M6	M6	M8	M8	M10	M10	M12
TG	22	26	32,5	38	46,5	56,5	72	89	110
WH	6,5	6	6,5	7	8	8	9	10	11
ZA+	37	39	44	45	45	49	54	67	81
ZB+	43,5	45	50,5	52	53	57	63	77	92

+ = aggiungere lunghezza corsa (mm) - *add stroke length (mm)*

Note: dado stelo compreso nella fornitura
Note: rod nut included in the supply

OPZIONE V (FEMMINA) - Z (MASCHIO) - *OPTION V (FEMALE) - Z (MALE)*

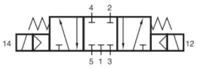
ø	020	025	032	040	050	063	080	100
A	16	22	22	24	32	32	40	40
AF	12	12	15	15	20	20	22	22
ø D	10	10	12	16	20	20	25	25
K	8	8	10	13	17	17	22	22
KF	M5	M6	M6	M6	M8	M8	M10	M12
KK	M8	M10x1,25	M10x1,25	M12x1.25	M16x1,5	M16x1,5	M20x1,5	M20x1,5

ø	CORSE STANDARD - *STANDARD STROKES*
020	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250
025	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250
032	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250 - 300 - 350 - 400
040	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250 - 300 - 350 - 400
050	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250 - 300 - 350 - 400
063	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250 - 300 - 350 - 400
080	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250 - 300 - 350 - 400
100	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250 - 300 - 350 - 400
125	5 - 10 - 15 - 20 - 25 - 30 - 40 - 50 - 60 - 70 - 75 - 80 - 90 - 100 - 125 - 160 - 200 - 250 - 300 - 350 - 400

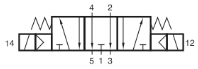
Appendix V – Pneumatic Solenoid Valves

2.5.3.1.2
page 130

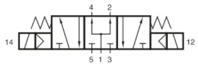
MH 53_ 501/MH 53_ 701/MH 53_ 801



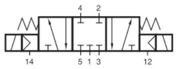
MH 531 501
MH 531 701
MH 531 801



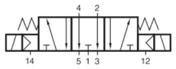
MH 532 501
MH 532 701
MH 532 801



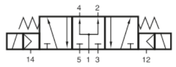
MH 533 501
MH 533 701
MH 533 801



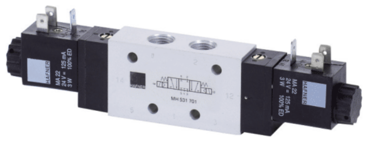
MEH 531 501
MEH 531 701
MEH 531 801



MEH 532 501
MEH 532 701
MEH 532 801



MEH 533 501
MEH 533 701
MEH 533 801



5/3-way solenoid valve with spring return to middle position, actuated by permanent signal.

Type 531 centre closed
Type 532 centre exhausted
Type 533 centre pressurised

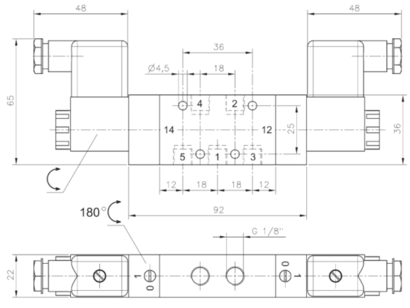
When ordering please complete the type number by 1, 2 or 3 according to the type required.

Available with solenoid operators:
230V/50Hz, 110V/50Hz, 24V/50Hz, 48V=,
24V =, 12V=.

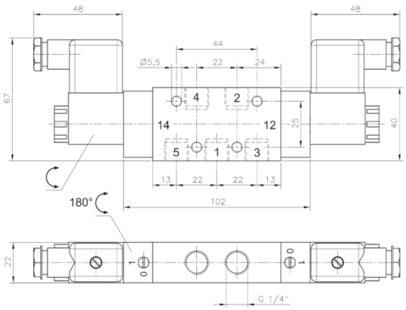
Valves are generally equipped with manual override. If requested without manual override please order M 53_ _ _ _.

Valves are also available with external pilot feed.
Type: MEH 53 _ _ _ _ *(please add 1 digit for type and 3 digits for size)*.
Ports 12 and 14 series 501 and 701: M5,
series 801: G 1/8".
Minimum actuation pressure: 3 bar.
Operating pressure: 0-10 bar.

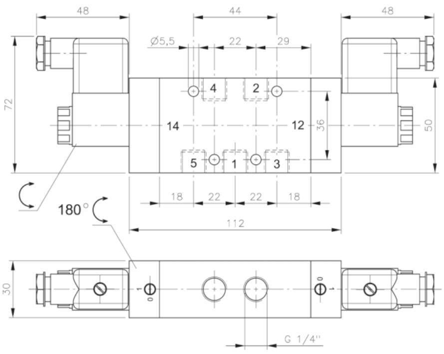
Version for vacuum on request.



MH 53_ 501



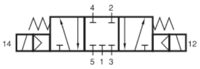
MH 53_ 701



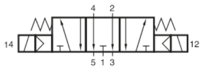
MH 53_ 801

Type	Port size	Air flow	Operating press.	Power consumption	Weight	
MH 53_ 501	G 1/8“	650 l/min	3 - 10 bar	3 W = / 5 VA ~	0,33 kg	Ex
MH 53_ 701	G 1/4“	1250 l/min	3 - 10 bar	3 W = / 5 VA ~	0,35 kg	Ex MK
MH 53_ 801	G 1/4“	1450 l/min	3 - 10 bar	3 W = / 5 VA ~	0,62 kg	Ex

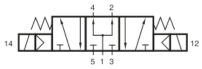
MH 53_ 101/MH 53_ 121/MH 53_ 181



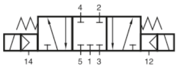
MH 531 101
MH 531 121
MH 531 181
MH 531 121 NPT



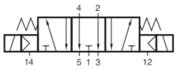
MH 532 101
MH 532 121
MH 532 181
MH 532 121 NPT



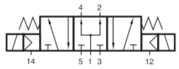
MH 533 101
MH 533 121
MH 533 181
MH 533 121 NPT



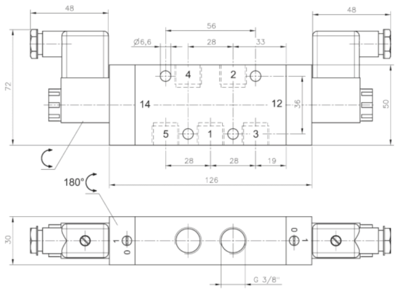
MEH 531 101
MEH 531 121
MEH 531 181



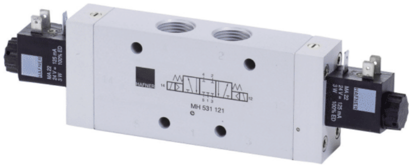
MEH 532 101
MEH 532 121
MEH 532 181



MEH 533 101
MEH 533 121
MEH 533 181



MH 53_101



5/3-way solenoid valve with spring return to middle position, actuated by permanent signal.

Type 531 centre closed
Type 532 centre exhausted
Type 533 centre pressurised

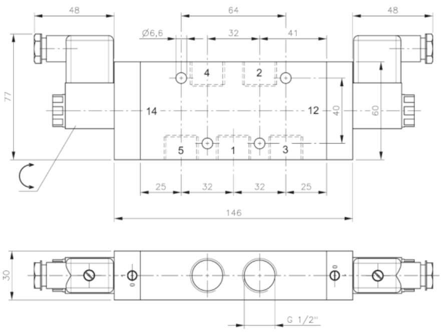
When ordering please complete the type number by 1, 2 or 3 according to the type required.

Available with solenoid operators:
230V/50Hz, 110V/50Hz, 24V/50Hz, 48V=,
24V =, 12V=.

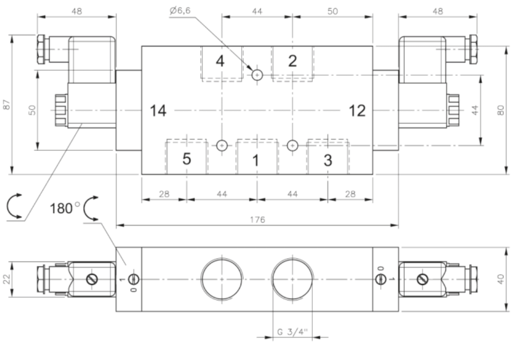
Valves are generally equipped with manual override.
If requested without manual override please order
M 53_ _ _ _.

Valves are also available with external pilot feed.
Type: MEH 53 _ _ _ _ *(please add 1 digit for type and 3 digits for size)*.
Ports 12 and 14: G 1/8".
Minimum actuation pressure: 3 bar.
Operating pressure: 0- 10 bar.

Version for vacuum on request.



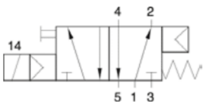
MH 53_121/MH 53_ 121 NPT



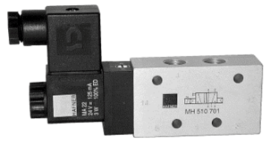
MH 53_181

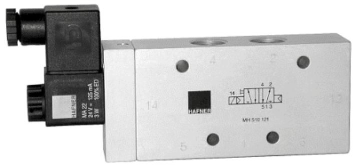
Type	Port size	Air flow	Operating press.	Power consumption	Weight
MH 53_ 101	G 3/8"	2250 l/min	1,5 - 10 bar	3 W = / 5 VA ~	0,66 kg
MH 53_ 121	G 1/2"	3000 l/min	1 - 10 bar	3 W = / 5 VA ~	0,84 kg
MH 53_ 181	G 3/4"	6000 l/min	1 - 10 bar	3 W = / 5 VA ~	1,45 kg
MH 53_ 121 NPT	1/2" NPT	3000 l/min	1 - 10 bar	3 W = / 5 VA ~	0,84 kg

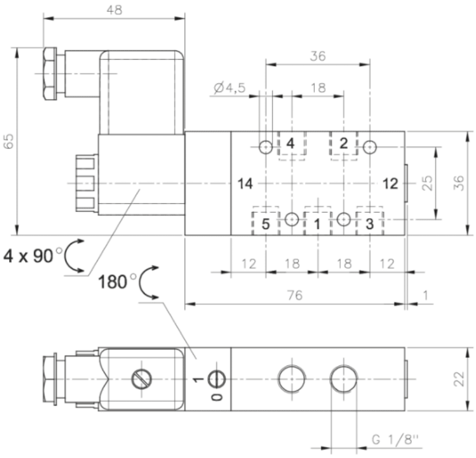
HAFNER



MH 511 501
MH 511 701
MH 511 121







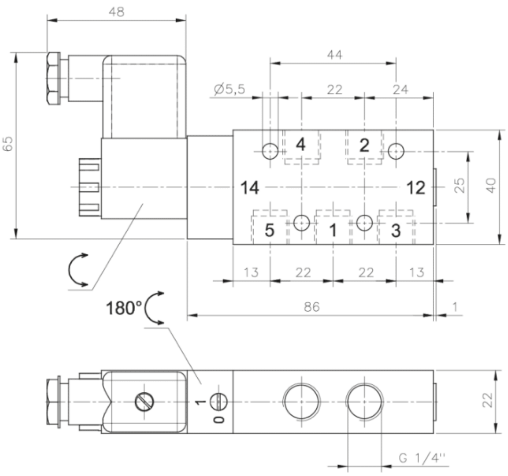
MH 511 501

5/2-way solenoid valve actuated by permanent signal and equipped with combined air and mechanical spring return.

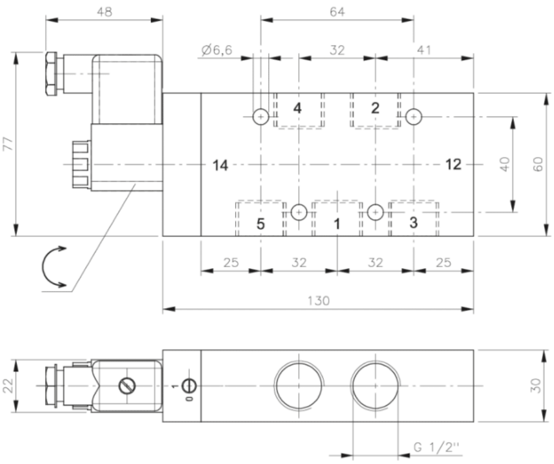
Available with solenoid operators:
230V/50Hz., 110V/50Hz., 24V/50Hz., 48V=, 24V=, 12V= .

Valves are generally equipped with manual override to turn. If requested without manual override please order M 511 _ _ _.

Connector as shown on the photo is included.



MH 511 701



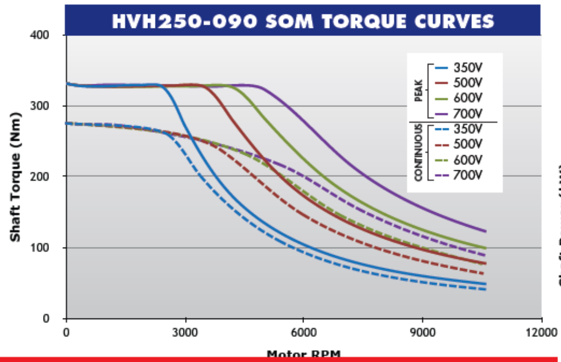
MH 511 121

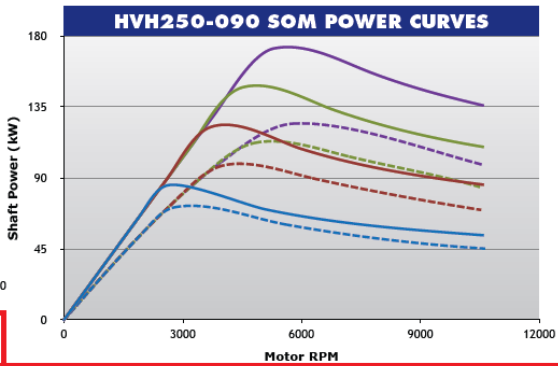
Type	Port size	Air flow	Operating press.	Power consumption	Weight
MH 511 501	G 1/8"	650 l/min	3 - 10 bar	3 W = / 5 VA ~	0,23 kg
MH 511 701	G 1/4"	1250 l/min	3 - 10 bar	3 W = / 5 VA ~	0,22 kg
MH 511 121	G 1/2"	3000 l/min	3 - 10 bar	3 W = / 5 VA ~	0,70 kg

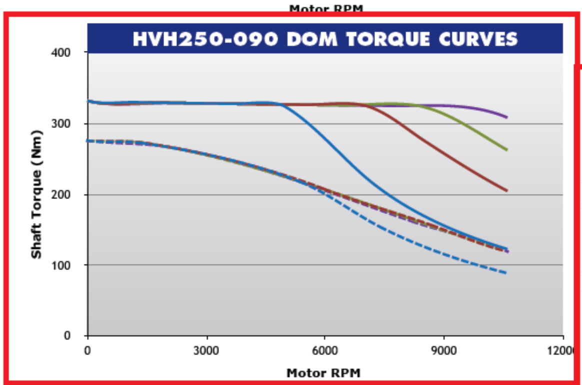
Appendix VI – Electric Motor

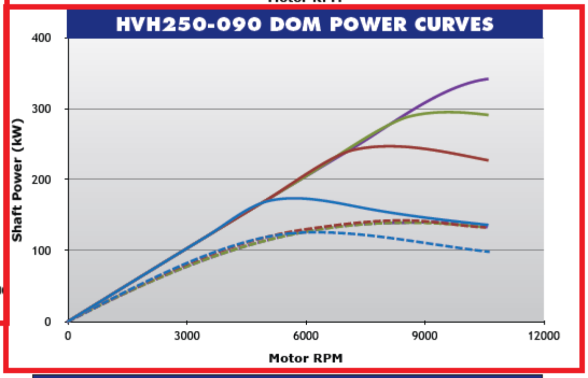
HVH250-090 Electric Motor

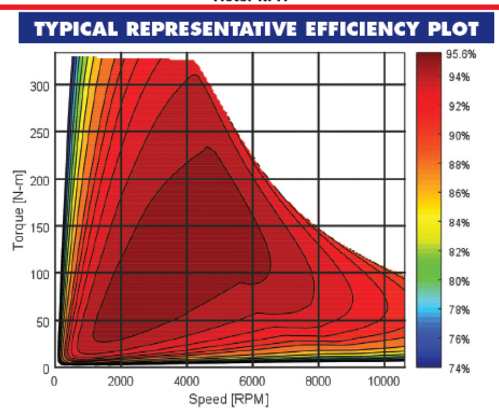
Product Details











OPERATING CONDITIONS	
Typical Coolant Inlet Temperature	up to 90 C
Typical Coolant Flow Rate	5 to 30 LPM
DC Bus Voltage	up to 700 V
Peak Current	300*/600** Arms
Rated Peak Operating Time	60 sec
Standard Cooling Medium	Dexron VI
MOTOR MASS DATA	
Motor Assembly	49 kg
Motor Rotational Inertia	0.067 kg·m^2

*Series wound stator **Dual path stator

Note: Graphs above are based on actual test data. The torque and power ratings are based on typical operating conditions as noted on the performance graphs. There are several variables that may change the motor performance, including coolant flow rate, operating temperature, inverter settings and parameters, etc. For actual performance, the motor must be evaluated in its final system and application. All specifications are subject to change.

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