

A New Control Strategy of Wet Dual Clutch Transmission (DCT) Clutch and Synchronizer for Seamless Gear Preselect

Mohammad Adhitya, Rashad Mustafa, Artur Plötner and Ferit Küçükay
 IAE, TU Braunschweig

ABSTRACT

In this paper, a new gear preselect strategy of wet dual clutch transmission (DCT) is proposed by delaying the gear preselect to the time near the power shift time in order to create a seamless gear preselect. Software in the loop (SiL) method is used to investigate the proposed strategy. High fidelity 13 degree of freedom lumped inertia drivetrain models, i.e., synchronizer model and electro hydraulic model, are constructed and validated using laboratory test data. A closed loop control for electro hydraulic system is developed to control the clutch and synchronizer considering the optimum synchronizer trigger time. Furthermore, the model used in this study is constructed considering the valve spool dynamic in electro hydraulic system and the sleeve dynamic including axial drag force in synchronizer system. The genetic algorithm has been used to obtain the optimum trigger time which is close to the power shift time to sustain uninterrupted torque during gear shifting. A comparison between the conventional sport mode gear preselect strategy with the proposed gear preselect strategy is performed. The obtained results of the proposed strategy show a fuel reduction of 0.8% in New European Driving Cycle (NEDC), within the allowable limits of the shift time.

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1. INTRODUCTION

Recently, many high performance vehicles have been using dual clutch transmission (DCT) as their transmission to ensure fast gear shifting with uninterrupted torque for achieving higher acceleration and speed. On the other hand, the requirement to reduce fuel consumption is more strictly stated by the official regulator that typically will diminish the vehicle performance if the subsequent improvement of the vehicle is not further made. Thus, the vehicle manufacturers are forced to increase the efficiencies of all vehicle components to meet these new regulations. As a complex system, the improvement of DCT efficiency can be achieved by either improving the software system as a relatively easy and low-cost method or changing the DCT hardware component which is considerably expensive.

Structurally, DCT is a merging of two manual transmissions which are well known for its robust construction. DCT employs two separate clutches which hold different gears. The first clutch holds the odd 1st, 3rd, 5th, 7th gears, whereas the second clutch grasps the even 2nd, 4th and 6th gears. In order to work properly, DCT utilizes an intelligence system to control the actuator mechanism

through the electrohydraulic system. Therefore, the gear in DCT can be shifted automatically. However, most of DCT still provide the manual gear shift mode which is preferable to the enthusiast drivers.

Many researchers have been continuously working on the DCT for better understanding of its system [1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18,19,20,21,22,23,24,25,26,27,28,29,30,31]. Alvermann et al. had proposed the clutch control in gear shifting for better comfort [7], whereas the synchronizer working load due to the drag torque in transmission was investigated by Walker et al. [2, 4, 28, 29]. Furthermore, the electrohydraulic system for gear shift actuator and the shift schedule to increase the drivability and DCT quality were explored and optimized by Mustafa et al. and Liu et al., respectively [5-6, 19]. Differentiating from the previous researches, an enhancement of the wet DCT efficiency is proposed and performed in this work by improving the gear preselect strategy by knowing the limitation of DCT mechanical construction and electrohydraulic system. The obtained results of the proposed method are also compared with those of the conventional sport mode gear preselect strategy.

2. GEAR PRESELECT STRATEGY

The DCT gear shift process is merely a simultaneously engine torque transfer from current closed clutch to the next open clutch to prevent torque interruption. This mechanism is similar to that of the automatic transmission which is known as clutch to clutch power shift, but completely different with that of the manual transmission i.e., disengaging from one current gear and then engaging to the next gear. Moreover, gear preselect can be defined as a gear engagement process of one constant mesh gear to the open clutch through the synchronizer system before the gear shift. This method is used to guarantee the readiness of an open clutch for receiving the engine torque from the current closed clutch and simultaneously transmitting it to the wheel through new gear ratio. Each clutch is only possible to be engaged with just one gear at a time. Furthermore, only one of the dual clutches may be engaged to the engine in order to avoid serious gearbox damage. In most DCTs, gear preselect is automatically done before the gear shift by DCT system without any inputs from the driver. It possibly will cause some dilemmas in manual gear shift mode when the preselected gear that automatically done by DCT system is different with the next gear which is desired by the driver resulting in a longer gear shift time. To overcome this dilemma, some DCT using so-called manual gear preselect in manual gear shift mode that manually selected by the driver before the gear shift to guarantee the right next gear has been preselected before the gear shift, this method however will increase the driver task load.

To handle a high engine torque, DCT is using a wet clutch instead of a dry clutch with the aim of dissipating more heat generated during clutch engaging process and minimizing the clutch pad wear to extend the gearbox lifetime. However, wet clutch DCT also has some drawbacks compared to the dry clutch DCT, e.g., it needs more energy to pump the clutch lubrication oil which also produce a viscous drag in case of a slip between clutch pad set.

Fig. 1 shows the measured results of the rotational speeds of engine, odd clutch, and even clutch for two different DCT driving modes. From this characterization, it is clearly shown that the gear shift of the sport mode has a higher engine rpm and a shorter shifting duration than that of the comfort mode. The different gear shifting durations are intended to be used in different purposes. The shorter gear shifting duration is well suited to the sport mode for having a quick response of the gear shifting. Whereas, the comfort mode is likely to have a longer shifting time in order to reduce the gear shift jerk. Moreover, the open clutch will stick and rotate together with clutch pack before gear preselect taking place in the comfort driving mode as shown in Fig. 1(b). Nevertheless, after the gear preselect has been done, the open clutch will rapidly change its rotation speeds to be either higher or lower than that of the clutch pack to prepare for the upcoming shift, i.e., downshift or upshift. The different gear preselect strategies for both sport and comfort driving modes are also depicted in

Fig. 1. In the sport mode, gear preselect is performed only after the gear shift has been finished to shorten the next gear shift time. This mechanism is opposite to that of the comfort mode where the gear preselect is executed only before the gear shift taking place. By comparing those two methods, the sport driving mode obviously exhibits higher drag losses in terms of the strategy for gear preselect compared to the comfort driving mode due to the extended slip clutch time.

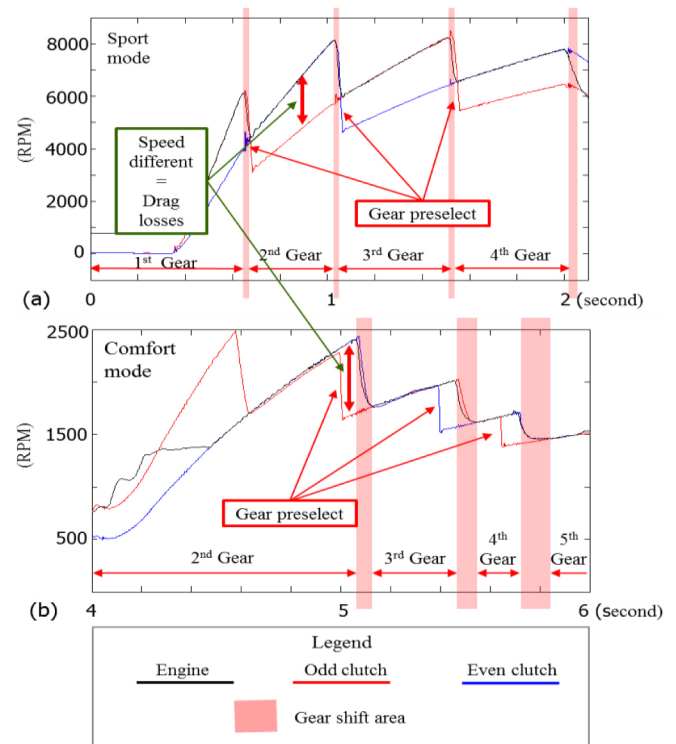


Figure 1. Measured rotational speeds of engine, odd clutch, and even clutch for (a) sport driving mode and (b) comfort driving mode.

As an alternative for gear preselect systems, a new seamless gear preselect is proposed in this study. The working principle of this gear preselect strategy is similar with that of the comfort mode. However, instead of postponing the clutch to clutch power shift after gear preselect is completely done, this seamless strategy is achieved by almost simultaneously activating the synchronizer engage action with clutch to clutch power shift in order to reduce the gap time between gear preselect and the gear shift as shown in Fig. 2. The action sequences of the seamless gear preselect method can be described as the synchronizer engagement to preselect the wanted next gear, followed by the clutch to clutch power shift and finally ended by synchronizer disengaging from the previous gear.

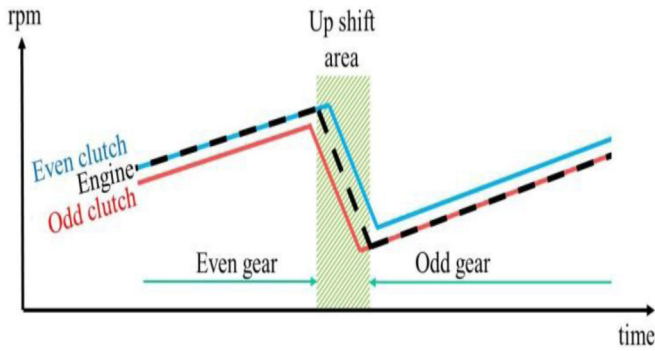


Figure 2. The illustration of the rotational speed of engine, odd clutch and even clutch for seamless gear preselect strategy in upshift from even to odd gear.

The seamless gear preselect strategy is beneficial, particularly for manual driving mode because the gear preselect is not able to be further performed automatically by DCT system before the clutch to clutch power shift. This seamless gear preselect can only be executed after the driver gear shift command, i.e., downshift or upshift, is existed and this will overcome the dilemma of gear preselect in manual gear shift mode as mentioned before. Furthermore, the gear shift response time of the seamless gear preselect will exhibit a slightly longer time in the manual driving mode compared to the manual sport mode. Nevertheless, for wet DCT, it will gain the other advantages, e.g., less clutch drag torque and reduced work load of DCT system for preparing the next wanted gear by its preselection.

3. DRIVETRAIN MODEL

The vehicle studied in this paper is BMW M3 E92 model with 7 speed DCT designed by Getrag. The general technical data of the vehicle engine and transmission are shown in Table 1. The construction of the DCT comprises four types of shafts (i.e., counter, hollow, solid, and output shafts), two types of clutches (i.e., even and odd clutches), and synchronizers as illustrated in Fig. 3. The solid input shaft connects the first larger diameter clutch to the set of odd gears while the hollow input shaft connects the second smaller diameter clutch to the set of even gears. Two double cone synchronizer systems are positioned in the counter shaft for the 1st, 2nd, 3rd and reverse gears, while one single cone synchronizer system is positioned in the solid input shaft for 5th and 7th gears and one single cone synchronizer positioned in hollow input shaft for 4th and 6th gears. By using this configuration setup, the engine torque can flow from one of the two input shafts to the output shaft through the counter shaft during driving, with the exception for 7th gear which is direct torque flow from solid input shaft to the output shaft with a 1:1 speed ratio.

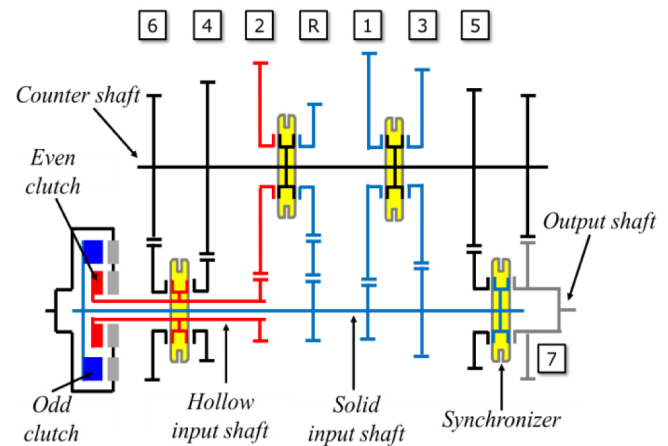


Figure 3. The structure of DCT Getrag Powershift Transmission 7DCI600

Table 1. Technical data of BMW M3 Engine and DCT Getrag Powershift Transmission 7DCI600

BMW M3 E93 Engine Specification	
Engine type	S65B40 naturally aspirated
Capacity/configuration	4.0 liter/V8
Max power	414 bhp @ 8300 rpm
Max torque	400 Nm @ 3900 rpm
Weight (dry)	202 kg
DCT Getrag 7DCI600 Specification	
Maximum input speed	9200 rpm
Maximum torque	600 Nm
Weight (dry)	79 kg
Installation length	660 mm
Max. vehicle/trailer mass	2500 kg/4500 kg
Gear spread ratio	4.8
Clutch type	Wet coaxial normally open
Actuator	Electrohydraulic system
Synchronizer System	
1 st , 2 nd , 3 rd & reverse gears	Double cone synchronizer
4 th , 5 th , 6 th & 7 th gears	Single cone synchronizer

3.1. Drivetrain Lumped Inertia Model

The diagram of powertrain model developed for this study is shown in Fig. 4. This 13 degree of freedom lumped inertia model is a simplification from the real powertrain construction without sacrificing the important parameters, particularly for gear shifting study [1, 2]. All model parameters of the lumped inertia are listed in the Appendix 1,2,3 including all derived equations.

The dynamic drivetrain model consists of two major form equations, i.e., the lumped inertia and spring-damper models. In Fig. 4, the lumped inertia model for all mass moments of the component inertia and the spring-damper model for all shafts are illustrated as blocks and spring dampers, respectively. All of these sets of equations form a chain of drivetrain dynamic including the lock-release process of double clutch. However, the synchronizer system equations

are described separately from [Appendix 1](#) for better explanation. Engine torque T_m used in this study is simply modeled as a static torque in a lookup table as a function of engine speed ω_m and throttle opening percentage T_A as shown in [Fig. 5](#).

The model parameters presented in [Appendix 1](#) are mostly based on the measurement data and the published literatures [1, 2]. Nevertheless, in order to complete the needed parameters in this model, some estimation values obtained from parameter estimation tool box in MATLAB Simulink are also used during verification of the model simulation results to the measurement data.

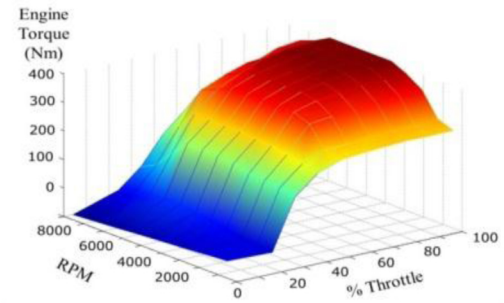


Figure 5. The static engine torque as a look up table

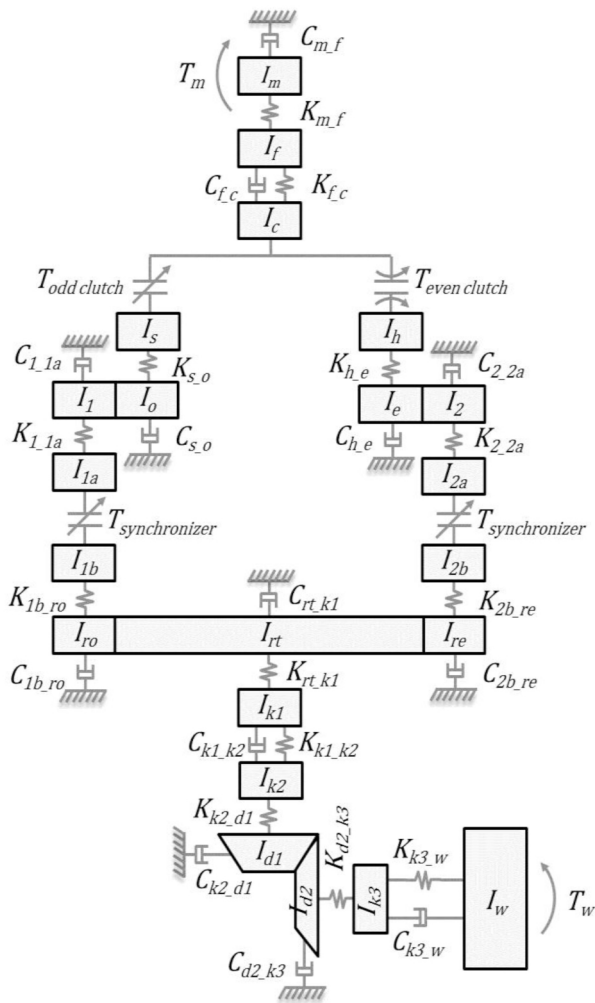


Figure 4. The 13 degree of freedom lumped inertia drivetrain model. Blocks represent the component inertia and springs represent the shaft stiffness. This diagram shows a car driving with the 1st gear where the 2nd gear is already preselected as an example. See [Appendix 1](#) for complete model parameter and all the equations.

3.2. Synchronizer System Model

In this design, synchronizer system is used to connect two separate rotating parts with positive interlocking made by meshing between splines of several components as shown in [Fig. 6\(a\)](#). The hub (green color) has inner and outer splines that are meshed with the shaft and sleeve splines, respectively. The sleeve can move in shaft axial direction from its neutral position toward gear hub spline. The strut with a notch is pushed by the helical spring to the outside radial direction to plug the sleeve detent groove and hold the sleeve in neutral position while all sleeve spline lengths are in contact only with the hub spline. The ring has an inner cone friction surface which rubs the outer cone friction surface of the gear hub and acts as a synchronizer cone clutch. The ring has a narrow tab that fills the wider slot in the hub. Thus, the ring can rotate together with the hub and also spin relatively to it with a limited rotating degree depending on the cone clutch torque direction as shown in [Fig. 7](#) (phase 2). This limited ring rotation at its maximum locking position will block the sliding path of the sleeve spline.

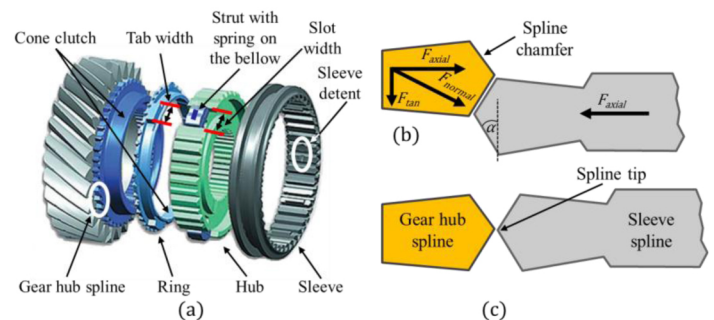


Figure 6. Synchronizer system with its (a) main components, (b) forces involved in its splines, and (c) tip to tip spline blocking.

The synchronizer model is derived from five phases of the engagement processes as a function of sleeve axial displacement from its neutral position toward the gear hub as depicted in [Fig. 7](#). The first, second, third, fourth, and fifth phases are the sleeve neutral position breaking, the speed

synchronization, the ring unblocking, the hub indexing and the spline locking, respectively.

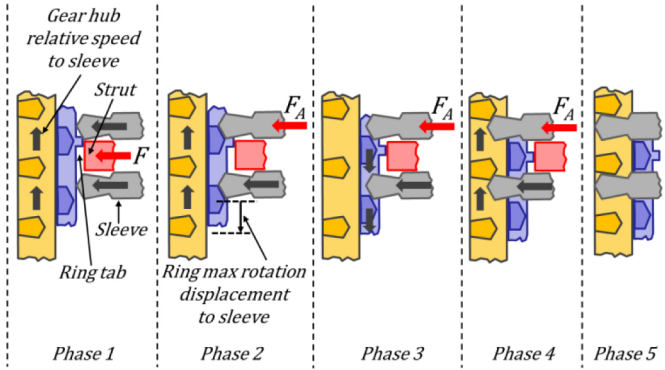


Figure 7. Five phase of synchronizer engagement process

In the first phase, the sleeve is pushed by a hydraulic gear lever actuator with an axial force F_A toward the gear hub. This initial force rises until the break through load (BTL) value is exceeded in order to overcome the resistance of strut spring force that holds the sleeve on its neutral position. The BTL can be described as [3]

$$BTL = 3 \cdot F_R \cdot \left(\frac{\mu_{str} + \tan \theta}{1 - \mu_{str} \tan \theta} \right) \quad (1)$$

where F_R is the reaction force of strut spring, μ_{str} is the friction coefficient between strut and sleeve detent groove, and θ is the sleeve detent ramp angle. Before the axial force F_A value reaches the BTL, the sleeve brings the strut (red block in Fig. 7) to push the ring tab and start squeezing the oil film in cone clutch friction surface [4] and creating a cone torque T_C that can be described as

$$T_C = 4 \cdot \pi \cdot R_C^3 \cdot b \cdot \left(\frac{\omega_S}{h} \right) \quad \text{for } x_S < X_0 \quad (2)$$

where R_C is the cone mean radius, b is the width of the cone contact surface, ω_S is the cone slip speed, h is the film thickness, X_S is the sleeve displacement and X_0 is the minimum sleeve displacement for cone contact.

In the second phase, the sleeve is sliding toward the gear hub after the axial force F_A exceeds the BTL. This movement will be stopped when the sleeve sliding movement is blocked by ring spline chamfer. The tangential force F_{tan} in ring spline which is a reaction of the axial force F_A (Fig. 6(b)) creates a chamfer torque T_B that can be derived as

$$T_B = F_A \cdot R_I \cdot \left(\frac{1 - \mu_I \tan \beta}{\mu_I + \tan \beta} \right) \quad (3)$$

where R_I is the chamfer pitch radius, μ_I is the friction coefficient of the chamfer and β is the chamfer angle. At the

same time, the axial force F_A will also create a cone torque T_C synchronizing the targeted gear speed that can be calculated as

$$T_C = \frac{\mu_C \cdot F_A \cdot R_C}{\sin \alpha} \quad \text{for } x_S \geq X_0 \text{ and } \omega_S \neq 0 \quad (4)$$

where μ_C and α are the cone friction coefficient and angle, respectively.

In the third phase, the ring unblocking is started when the chamfer torque T_B exceeds the cone torque T_C . The axial force F_A through the sleeve spline chamfer rotates and pushes the ring as a cone clutch for finishing the speed synchronization process during ring unblocking. The cone torque T_C at the end of the speed synchronization is

$$T_C = T_D + I_{FW} \cdot \alpha_{FW} \quad \text{for } x_S \geq X_0 \text{ and } \omega_S = 0 \quad (5)$$

where T_D is the drag torque, I_{FW} is the freewheeling inertia and α_{FW} is the angular acceleration of freewheeling component. The drag torque T_D is a total resistance torque comprises all resistances from the bearings, seals, gears and friction which can be written as

$$T_D = -\text{sign}(\omega_G) \cdot (T_W + T_F + T_B) \cdot -\text{sign}(\Delta\omega_{CL}) \cdot (T_{SH} + T_B + T_{CL}) \quad (6)$$

where ω_G is the gear speed, T_W is the gear windage torque, T_F is the gear tooth friction torque, T_{SF} is the shaft drag torque, T_{CL} is the clutch drag torque and $\Delta\omega_{CL}$ is the clutch slip speed.

In the fourth phase, the sleeve continues to slide towards the gear hub until it is blocked again by the gear hub spline chamfer. The gear hub splines are randomly aligned to the sleeve splines as shown in Fig. 8. Therefore, there are four possible schemes that can be obtained in this step. The first possibility is when the drag torque has the same direction with chamfer torque (Fig. 8(a)). In this case, the need of axial force to finish the spline engagement will significantly be reduced. Meanwhile, the second possibility occurs when the drag torque suffers the axial force to unblocking and finishes the spline engagement (Fig. 8(b)). The phenomenon of this axial force sudden rising is known as a second bump. Furthermore, the third possibility shows the best alignment without any chamfer torques (Fig. 8(c)). The worst possibility however is achieved with tip-on-tip spline alignment. In this case, the synchronizer will block out or fail to lock the gear hub splines (Fig. 8(d)). The spline tip is designed as sharp as possible to avoid a surface contact when tip-on-tip spline alignment occurs. Thus, the sleeve still can continue to slide and unblock the sliding way. There are four indexing torque T_I equations regarding to the gear hub spline alignments which can be described in Eq. (7). The sleeve is then fully

locked with the gear hub spline in the fifth phase after passing the gear hub splines blocking.

$$T_I = \begin{cases} F_A \cdot R_I \cdot \left(\frac{1 - \mu_I \tan \beta}{\mu_I + \tan \beta} \right) & \text{for } +T_D, +T_I \text{ direction} \\ F_A \cdot R_I \cdot \left(\frac{1 + \mu_I \tan \beta}{\mu_I - \tan \beta} \right) & \text{for } +T_D, -T_I \text{ direction} \\ -F_A \cdot R_I \cdot \left(\frac{1 + \mu_I \tan \beta}{\mu_I - \tan \beta} \right) & \text{for } -T_D, +T_I \text{ direction} \\ -F_A \cdot R_I \cdot \left(\frac{1 - \mu_I \tan \beta}{\mu_I + \tan \beta} \right) & \text{for } -T_D, -T_I \text{ direction} \end{cases} \quad (7)$$

The spline chamfer angle (α) is designed to create enough friction force on the cone friction surface to stop the slip between the ring and the gear hub before the sleeve continues to slide again toward the gear hub. Increasing the spline chamfer angle will decrease the friction force on cone friction surface and reduce the effectiveness of speed synchronization. On the other hand, lowering the spline chamfer angle will increase the axial force needed to unblock the synchronizer ring blocking. In order to increase the speed synchronization effectiveness, more friction surface areas are possible to be created using more than one cone as used in this Getrag DCT.

In this system, a double cone synchronizer system is utilized for the lower gears (i.e., the 1st, 2nd, 3rd gears) which have a relatively higher drag torque compared to the higher gears (the 4th, 5th, 6th and 7th gears).

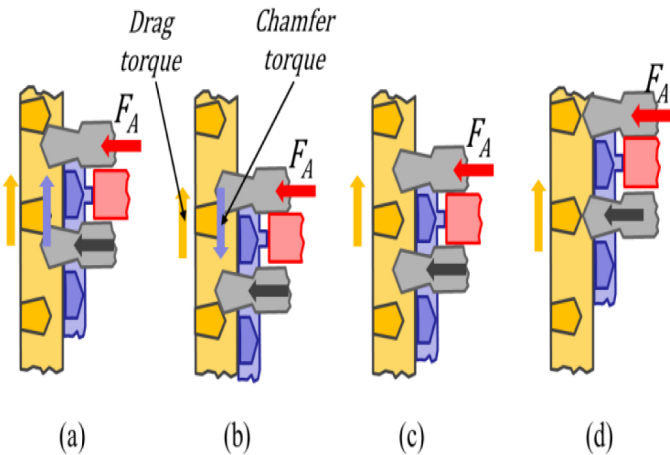


Figure 8. Four possibilities of the gear hub spline indexing

3.3. Electrohydraulic System Model

The model of the electrohydraulic valve consists of three main parts, i.e., the electromagnetic circuit, the flow equation and the spool dynamics.

3.3.1. Electrohydraulic System

The DCT electrohydraulic system comprises several mechanical components, e.g., mechanical oil pump, pressure regulator valve, normally closed control valve, clutch actuator piston and synchronizer sleeve actuator piston. Fig. 9 and Fig. 10 illustrate the DCT electrohydraulic system for clutch and synchronizer sleeve piston actuations, respectively.

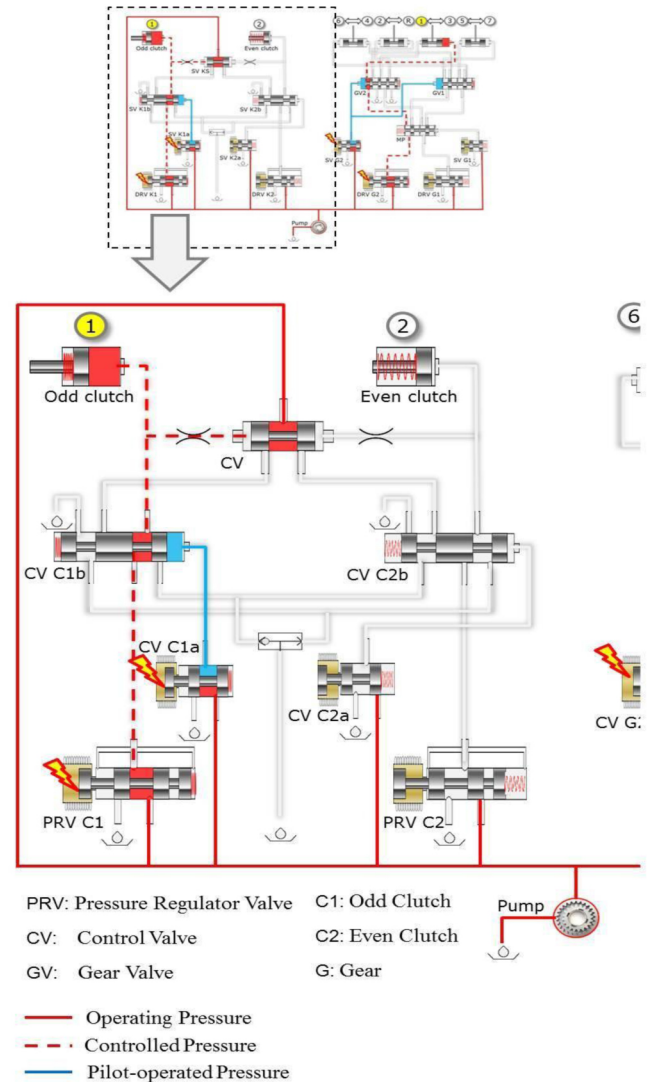


Figure 9. Layout of the electrohydraulic system for clutch piston actuation

In order to actuate the clutch piston as shown in Fig. 9, electric current is needed. This current will energize the spool in control valve CV-C1a and open the valve. Once this valve has been opened, the hydraulic fluid as pilot-operated pressure will be flown through it which is generated from the oil pump to control valve CV-C1b. The pilot-operated pressure then pushes the valve in CV-C1b to open the path for controlling the pressure yielded from the pressure regulator valve PRV-C1. The pressure in clutch piston