American Journal of Engineering Research (AJER)
e-ISSN: 2320-0847 p-ISSN: 2320-0936
Volume-15, Issue-2, pp-08-16
www.ajer.org
Research Paper Open Access

Methodology for Dynamic Modeling and Shift Control Development of a Hybrid Dual-Clutch Transmission

Sid Pasumarthi
Automotive Professional and Independent Researcher

Abstract
Dual Clutch Transmission (DCT) is widely used in conventional and hybrid automobiles as they provide fast gear shifts with high mechanical efficiency and minimal torque interruption. Despite these advantages, achieving smooth and repeatable shift quality remains a key engineering challenge and depends on a robust dynamic model of the transmission and effective clutch control strategies. Challenges are further amplified in hybrid DCT architectures, where torque transfer from the Internal Combustion Engine (ICE) and Electric Motor should be transferred to the wheels through the DCT, ensuring smooth shifts and minimal torque interruptions. A systematic approach for the dynamic modeling, analysis, and shift control development of a hybrid P2.5 DCT using an analytical approach and MATLAB-based simulation is proposed. The proposal captures the torque flow through the transmission and driveline during normal operating conditions. Clutch models represent the clutch engagement, slip, and torque transfer behavior during gear shifts and demonstrate a clutch-to-clutch control, illustrating torque phase and inertia phase behavior under power-on conditions. Simulation results show the influence of the clutch pressure control and torque transfer strategies have the effect on the shift duration, torque transfer, and drivability. The proposed framework methodology provides a guideline for control strategy development, calibration, evaluation of shift quality, and serves as a foundation for future work in adaptive shift optimization for hybrid DCT applications.
Keywords: *Power-on Upshift, Clutch-to-Clutch Control, Inertia Phase, Torque Phase*

Date of Submission: 24-01-2026 Date of acceptance: 06-02-2026

I. Introduction

The P2.5 hybrid powertrain architecture enables the electric motor integrated with the transmission by connecting to the even gear set thereby providing the direct supply of torque into the transmission, assists the engine during acceleration, and compensates for torque interruptions during gear changes, and enables regenerative braking.

Dual-clutch transmission utilizes two independently controlled clutches, one clutch (C1) for odd-numbered gears and a second clutch (C2) for even-numbered gears. The arrangement enables the power-on shifts by allowing one clutch to transmit torque through the currently engaged gear while the next gear is preselected on the parallel shaft. During a gear change, torque is smoothly transferred from the off-going clutch to the on-coming clutch without interrupting power delivery to the transmission output shaft, thereby to the wheels. The clutch-to-clutch shift mechanism results in significantly reduced shift times providing the higher mechanical efficiency

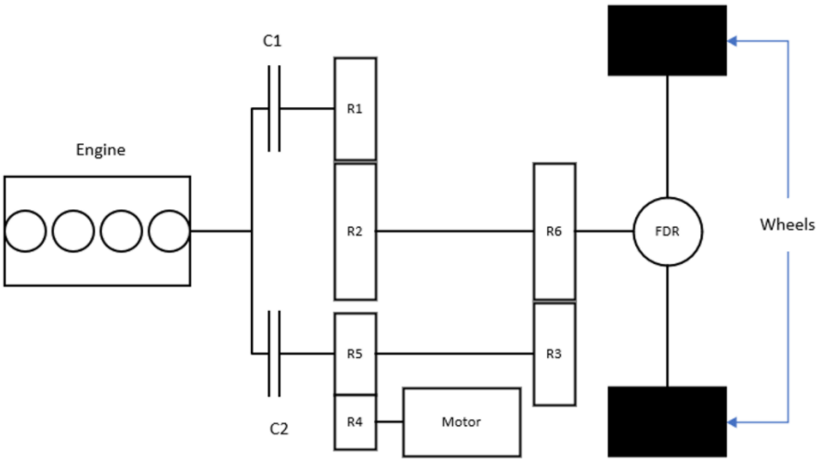

Fig.1 Architecture of the 2.5 DCT Hybrid Transmission

P2.5 hybrid DCT configuration has improved shift performance by the presence of the electric motor. The motor can actively add or subtract torque during the torque phase and inertia phase of a shift, assisting clutch torque handover thus reducing the driveline disturbances.
Additionally, the hybrid system supports synchronization of shaft speeds during both upshifts and downshifts, improving shift smoothness and minimizing jerk. P2.5 hybrid architecture paired with the DCT architecture delivers quick shift responses enhancing the driving experience while maintaining optimal fuel efficiency.

II. Dynamic Modeling of P2.5 Dual-Clutch Transmission

Need for Dynamic Modeling: During the automobile's design transmissions are designed to meet the demands of shift quality and drivability KPI targets which have direct impact on the customer perception of the automobile. Dynamic modeling plays a key role in ensuring that transmission is integrated with the vehicle to meet the KPI metrics and establishes a baseline for controls and calibration development. Dual-clutch transmissions have elements connecting torque providers with frictional elements, and rotating inertias that interact with each other during gear shifts. Static or kinematic models are unable to capture the transient events, such as clutch slip, torque transfer during the oncoming and off-going clutch engagements that would impact the vehicle's acceleration and jerk; all of these events impact the shift quality and drivability perception. A physics-based dynamic model estimates the transient response and provides a systematic baseline for the design and evaluation of shift control and calibration strategies. Key performance metrics such as shift quality and drivability, can be governed with these modeling exercises and can be assessed using simulation prior to hardware implementation. Dynamic modeling capability reduces calibration effort, improves the control development, and overall transmission performance.

Approach to Dynamic Modeling: The dynamic model developed for the Hybrid Dual-Clutch Transmission (DCT) includes the engine/input shaft, odd-gear shaft, even-gear shaft, and output shaft, along with kinematic gear constraints and external torques from the clutches and road load.

Simulation of the Hybrid DCT needs assumptions to be stated for the parameters such as Gear Ratios, Driveline layout, Inertias, Final Drive Ratio etc.

Category	Assumption
Gear Ratios	R_1 = 3.50, R_2 = 2.10, R_3 = 1.40, R_4 = 1.00, R_5 = 0.80, R_6 = 0.60 (typical DCT ratio spread)
Driveline Layout	Two-shaft DCT architecture with C1 operating odd gears and C2 operating even gears
Inertias	Engine $J_i = 0.25$ kg·m², Motor $J_m = 0.05$ kg·m², Shaft inertias $J_1 = 2 \times 10^3$, $J_2 = 3 \times^2$ kg·m²
Final Drive	Fixed final drive ratio (FDR) = 3.73 applied to reflect wheel inertia
Clutch Modeling	$T_c = T_{cap} \cdot \tanh(\text{slip}/0.001)$; no thermal, hysteresis, or wear effects considered
Clutch Initial Conditions	C1 torque capacity = 1000 Nm (engaged), C2 = 0 Nm (disengaged)
	Slip defined as shaft-speed difference; no compliance or backlash modeled
Shift Trigger	Upshift initiated at input speed ≈ 2500 RPM
Torque-Phase Timing	Target torque-phase duration $d\iota_{TP}$ = 0.15 s
Torque Limits	Engine torque limited to 300 Nm; motor torque lin:100 Nm
Road Load	Vehicle mass = 1500 kg; zero road grade; no tire slip or compliance
Simulation Scope	Single 1 → 2 upshift simulated from launch; no multi-shift sequences

Fig.2 Assumptions of DCT Parameters used for MATLAB Simulation

Generalized Coordinates and States: Modeled four rotational degrees of freedom for the system that are defining the Input shaft angle, Odd Gear path & Even Gear path angles and Output shaft angles, and the angular speeds are defined corresponding to the rotational system angles.

$$\boldsymbol{\theta}(t) = \begin{bmatrix} \theta_i(t) \\ \theta_1(t) \\ \theta_2(t) \\ \theta_o(t) \end{bmatrix}$$

where:

- θ_i: engine / input shaft angle
- θ_1: odd-gear shaft (C1 path) angle
- θ_2: even-gear shaft (C2 path) angle
- θ_o: output shaft / final drive angle

Angular speeds are:

$$\boldsymbol{\omega}(t) = \dot{\boldsymbol{\theta}}(t) = \begin{bmatrix} \omega_i \\ \omega_1 \\ \omega_2 \\ \omega_o \end{bmatrix}$$

Fig.3 Equations of Rotational Degree of Freedom and Angular speed for Rotating Components

Kinetic Energy and Lagrangian: The total kinetic energy of the rotating components is represented from the equations derived from the Rotational Inertia of the Input shaft, Output shaft and DCT Components of odd & even gear shaft path.

$$T = \frac{1}{2}J_i\omega_i^2 + \frac{1}{2}J_1\omega_1^2 + \frac{1}{2}J_2\omega_2^2 + \frac{1}{2}J_o\omega_o^2$$

where:

- J_i: rotational inertia of engine/input shaft
- J_1: inertia of odd-gear shaft
- J_2: inertia of even-gear shaft
- J_o: equivalent inertia of vehicle and wheels reflected to the output shaft

Potential energy is neglected $(V \approx 0)$, so the Lagrangian is

$$L = T - V \approx T$$

Fig.4 Equations of Rotational Inertia of the Rotating Components

Gear Kinematic Constraints: Gear meshes impose the speed relationships between the intermediate shafts and the output shaft.

$$G_1(\boldsymbol{\omega}) = R_1\omega_1 - R_2\omega_o = 0$$

$$G_2(\boldsymbol{\omega}) = R_3\omega_2 - R_6\omega_o = 0$$

Fig.5 Equations for Gear Meshing

where R1,R2,R3,R6 are the effective gear ratios for the odd and even gear paths.

Equations of Motion with External Torques External torques acting on the system are proposed, applying the Lagrange's equations that derive the equations of motion.

- T_i: engine torque
- T_m: motor torque
- T_{c1}: clutch C1 torque
- T_{c2}: clutch C2 torque
- T_o: output torque due to road load (aerodynamic drag, rolling resistance, grade)

Using Lagrange's equations with constraints, the equations of motion can be written compactly as:

$$M\,\mathbf{a} = \mathbf{B}$$

where:

- $\mathbf{a} = [\dot\omega_i,\ \dot\omega_1,\ \dot\omega_2,\ \dot\omega_o,\ \lambda_1,\ \lambda_2]^T$
- M is the 6×6 mass/constraint matrix
- $\mathbf{B}$ is the external torque/right-hand side vector

The inertia matrix block is:

$$J = \begin{bmatrix} J_i & 0 & 0 & 0 \\ 0 & J_1 & 0 & 0 \\ 0 & 0 & J_2 & 0 \\ 0 & 0 & 0 & J_o \end{bmatrix}$$

and the constraint matrix is:

$$\Omega = \begin{bmatrix} 0 & R_1 & 0 & -R_2 \\ 0 & 0 & R_3 & -R_6 \end{bmatrix}$$

The full mass/constraint matrix is then:

$$M = \begin{bmatrix} J & \Omega^T \\ \Omega & 0 \end{bmatrix}$$

and the external torque vector is:

$$\mathbf{B} = \begin{bmatrix} T_i + T_m - T_{c1} - T_{c2} \\ T_{c1} \\ T_{c2} \\ -T_o \\ 0 \\ 0 \end{bmatrix}$$

Fig.6 Torque acting on the system derived from the Lagrange's Equations

III. Shift Control Strategy

Hybrid Dual Clutch Transmission shift control strategy is employed by the simulation of a Power On "1 → 2" upshift using the "2 Phase" Technique.

Phase I would be the "Torque Phase" which would move the drive torque from Clutch C1 to Clutch C2 with the target of keeping the output torque as smooth as possible.

Phase II would be the "Inertia Phase" which requires slowing down the input shaft until it reaches the same rotational speed as the 2nd gear after the clutch C2 is engaged. During this Inertia shift the output shaft speed disturbances shall be as minimal as possible

Torque Phase: The torque phase begins when a shift trigger condition is satisfied. In this exercise, the upshift is initiated when the transmission input speed exceeds a defined threshold as below

$$\omega_i \geq \omega_{\text{shft,th}} \quad \Rightarrow \quad \text{enter torque phase}$$

Fig.7 Upshift Initiation Threshold Condition

During the beginning of the torque phase, C1 Clutch is fully applied and carries nearly all drive torque, while C2 Clutch has near-zero capacity. The objective is to ramp down C1 capacity and ramp up C2 capacity such that the sum of drive torque remains constant. The clutch capacities are commanded as:

$$T_{\text{cap},1}(t) = T_{\text{cap},1}^{\text{start}} \left(1 - \frac{t - t_{TP,0}}{\Delta t_{TP}}\right)$$

$$T_{\text{cap},2}(t) = T_{\text{cap},2}^{\text{final}} \left(\frac{t - t_{TP,0}}{\Delta t_{TP}}\right)$$

for $t_{TP,0} \leq t \leq t_{TP,0} + \Delta t_{TP}$, where:

- $T_{\text{cap},1}^{\text{start}}$: initial C1 torque capacity at shift start
- $T_{\text{cap},2}^{\text{final}}$: target C2 capacity at end of torque phase
- Δt_{TP}: desired torque-phase duration

The actual clutch torques are computed using a smooth nonlinear clutch model:

$$T_{c1} = T_{\text{cap},1}(t)\,\tanh\left(\frac{\omega_i - i_1\omega_o}{\epsilon}\right), \quad T_{c2} = T_{\text{cap},2}(t)\,\tanh\left(\frac{\omega_i - i_2\omega_o}{\epsilon}\right)$$

where i_1 and i_2 are the effective gear ratios, ω_o is output shaft speed, and ϵ is a small regularization parameter.

The end of the torque phase is detected when the C1 torque drops below a small threshold:

$$|T_{c1}| \leq T_{c1,\text{th}} \quad \Rightarrow \quad \text{enter inertia phase}$$

Fig.8 Equations to Command Clutch Torque Capability for C1 and C2 Clutches

Inertia Phase: After C1Clutch is unloaded and opened, the shift enters the inertia phase. C2 remains applied, and the target is to decelerate the input shaft from its 1st-gear speed and engage into the 2nd-gear speed synchronously.

$$\omega_{i,\text{sync},2} = i_2\,\omega_o$$

$$\dot{\omega}_{i,\text{target}} = \frac{\omega_{i,\text{sync},2} - \omega_{i,TP}}{\Delta t_{IP}}$$

The simplified input shaft equation of motion during the inertia phase is:

$$J_i\dot{\omega}_i = T_i + T_m - T_{c2}$$

where T_i is engine torque, T_m is motor torque, and T_{c2} is C2 torque. The control objective is to choose $T_i(t)$, $T_m(t)$, and maintain $T_{\text{cap},2}(t)$ such that $\dot{\omega}_i$ tracks $\dot{\omega}_{i,\text{target}}$. A simple proportional shaping can be used:

$$T_m(t) = T_{m,\text{base}} + K_m\left(\dot{\omega}_{i,\text{target}} - \dot{\omega}_i\right)$$

with engine torque either held approximately constant or slightly reduced to avoid excessive acceleration spikes.

Fig.9 Equations to Maintain the Incoming and Outgoing Clutch Speeds

IV. Overview of the MATLAB Simulation

Time-domain simulation was developed in MATLAB to analyze the dynamic behavior of a dual-clutch transmission and derive clutch control strategy for the gear shifting. The Powertrain elements are modeled as a multi-degree-of-freedom rotational system representing the engine, electric motor, clutches, gear sets, along with the vehicle loads. The kinetic energy was formulated using the rotational inertias of the shafts, and the equations of motion were derived using Lagrange's method. External torques from the engine, electric motor, clutches, and road load were incorporated into the governing equations. Dynamic modeling enables calculation of the clutch torque required to transmit power and synchronize input and output shaft speeds during gear shifting. The torque demand is converted into a time-varying clutch pressure command established through a torque–pressure relation. Two phase control strategy is employed with torque and inertia phases, using the shaft speed feedback used to control the clutch engagement

```
if (omega_in_rpm >= p.shft_input_spd_rpm) && (Outputs.TorquePhase == 0)
    Outputs.TorquePhase = 1;
    Outputs.TP_tstart   = t;
end

if Outputs.TorquePhase == 1 && Outputs.InertiaPhase == 0
    tau = (t - Outputs.TP_tstart)/p.dt_TP;
    tau = min(max(tau,0),1);
    T_cap_1 = (1 - tau)*p.C1_max;
    T_cap_2 = tau * Outputs.Tc2_target;
end
```

Fig.10 MATLAB Simulation Code for the Command Clutch Torque Capability for C1 and C2 Clutches

Dynamic modeling enables the calculation of clutch torque capacity needed for the transmission of power and synchronizing shaft speeds during the gear shifting. Simulation results provide insight into shift time, torque transfer, slip behavior, and vehicle acceleration response, supporting the physics-based clutch control strategy development.

```
if Outputs.InertiaPhase == 1
    omega_sync_2 = p.iG2 * omega_out;
    omega_target_dot = (omega_sync_2 - Outputs.omega_i_TP) / p.dt_IP;

    err = omega_target_dot - dwdt(1);  % input shaft accel error
    Tm  = p.Tm_base + p.Kmot * err;    % motor assist
    T_cap_2 = p.C2_hold;               % keep C2 applied
end
```

Fig.11 MATLAB Simulation Code for Synchronous Speed handover of Incoming and Outgoing Clutch

V. Results and Discussion

The simulation results indicate that the two-phase shift control system for hybrid dual-clutch transmissions will successfully perform a 1→2 upshift. Clearly, the shift event is divided into two distinct phases of operation: the torque phase and the inertia phase, with expectation from a physical standpoint being met by the transmissions response.

During the torque phase, the departing clutch (C1) is unloaded smoothly, while during this same period of the sequencing, the engaging clutch (C2) ramps up to provide maximum torque capacity. Have successfully simulated the duration of the torque phase, and the absence of any discontinuity in the speed of the output shaft during this phase, will guarantee continued torque availability to the wheels. [Reference - Fig.12]

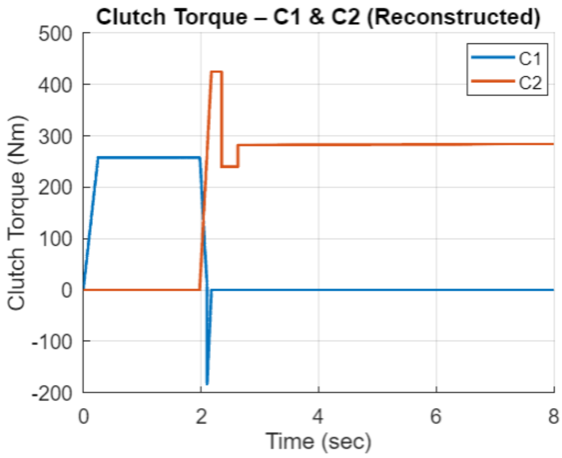

Fig.12 Clutch Torque changes during the Shift Control Strategy

Although there is a momentary slight disturbance to the vehicle acceleration during this timeframe due to torque relaying from the clutch, the magnitude of this disturbance is small enough to consider it minimal jerk and to provide acceptable shifting comfort. [Reference - Fig.13]

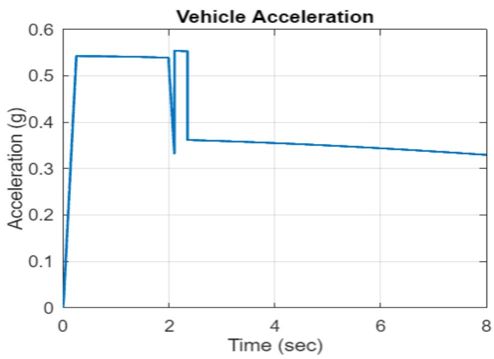

Fig.13 Vehicle Acceleration

In the inertia phase of design, the input shaft will decelerate down to the torque gear's synchronous speed as the engaging clutch (C2) remains engaged, and we have closely matched target value for the inertia phase duration. Near zero slip of the C2 clutch indicates that the system has achieved speed synchronization. [Reference - Fig.14]

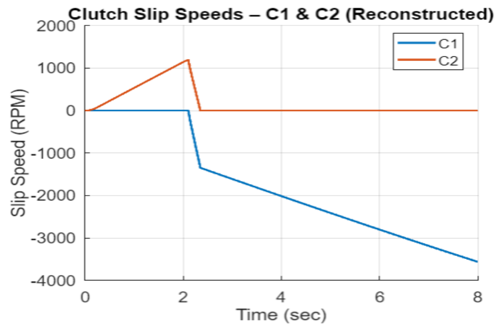

Fig.14 Clutch Slip Speeds during the Shifting

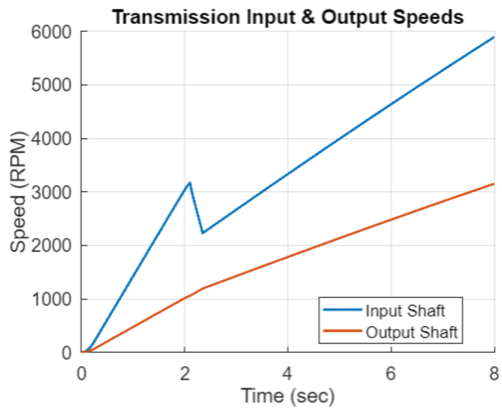

Fig.15 Input and Output Transmission Shaft Speed

The inertia of the driveline is being controlled, the system experienced a short duration of reduced vehicle acceleration. The engine's torques, clutch improved torques, and electric induction motor torques have all been maintained within feasible operating limits during the course of this shifting sequence.

It has been established through variation in the model's two primary input parameters; shift time and shift initiation speed, that this approach to shifting will perform adequately through the mid- to high ranges of parametric variance and still provide a completed shift with controllable levels of acceleration disturbance.

In summary, the findings presented demonstrate that the control strategy and model utilized in this work are capable of accurately reproducing the primary dynamic characteristics of hybrid DCT shift behaviour.

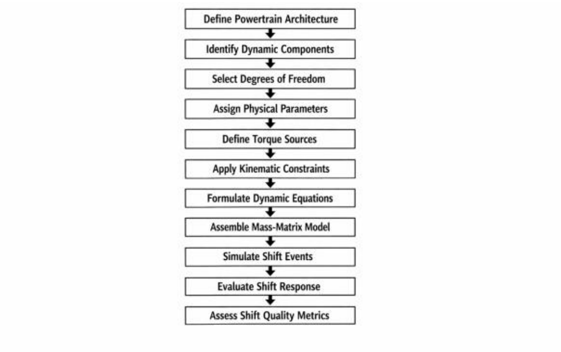

Fig. 16 Summary of the Methodology for Dynamic modeling and Clutch Control Strategy

VI. Conclusion

Two-phase shift control strategy, integrated with an dynamic simulation model for a Hybrid Dual Clutch Transmission (DCT) is capable of creating the vehicle's response to an Power On 1→2 upshift event. The mass-matrix model used in this simulation has succeeded in simulating all aspects of transmission dynamics, including clutch engagement and vehicle performance, both during steady state operations and shifting. The simulations show torque smoothly transferring, speed matching appropriately at the beginning of each shift, and acceleration occurring realistically. Finally, this approach has demonstrated that there is a clear benefit to applying torque-phase and inertia-phase control within hybrid DCTs and establishes a platform for future scope related to closed-loop slip control that would improve coordination between Engine and Electric motor torque with multiple gear-shifts.

REFERENCES

[1]. Naunheimer, H., Bertsche, B., Ryborz, J., & Novak, W., *Automotive Transmissions: Fundamentals, Selection, Design and Application*, Springer.
[2]. Gao, B., Chen, H., & Sanada, K., "Shift Control of Dual Clutch Transmissions," *IEEE/ASME Transactions on Mechatronics*.
[3]. Guzzella, L., & Sciarretta, A., *Vehicle Propulsion Systems*, Springer.
[4]. MathWorks, *MATLAB ODE Suite and Dynamic System Modeling Documentation*.