Indian Institute of Technology Hyderabad

Driveline of dual clutch transmission system

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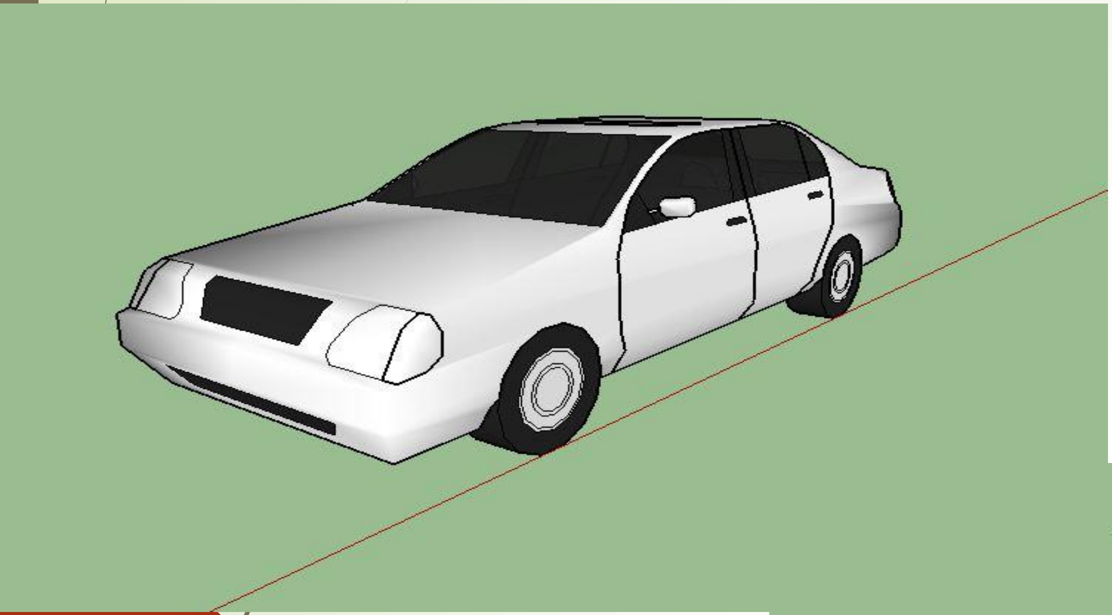
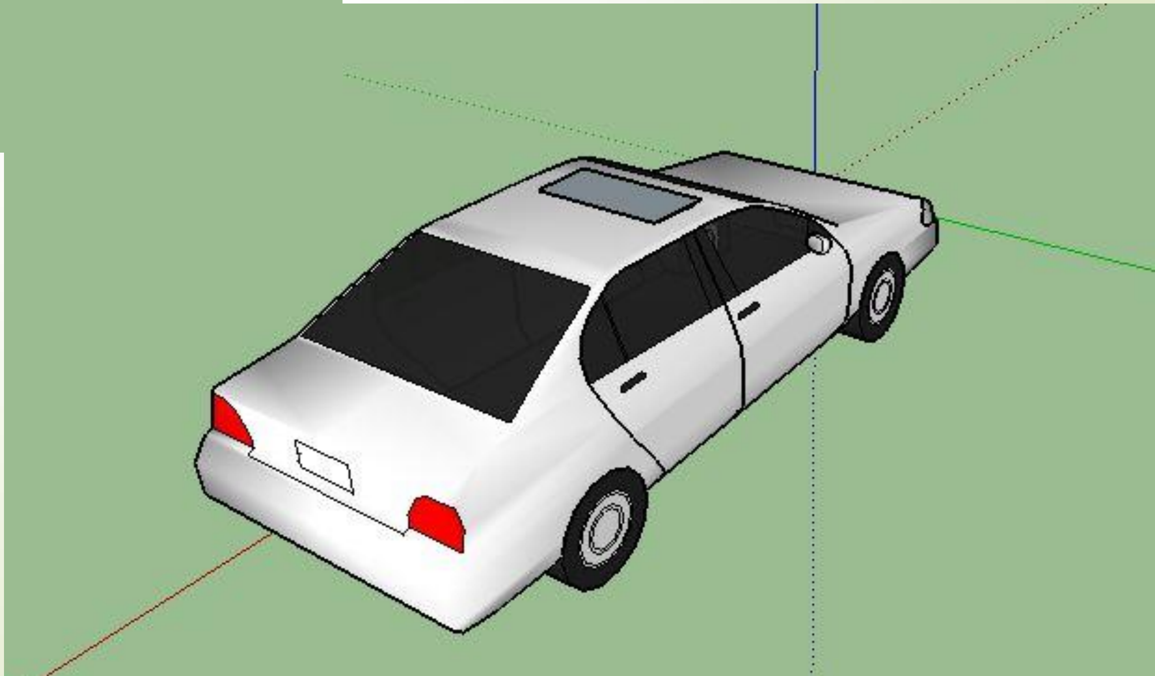
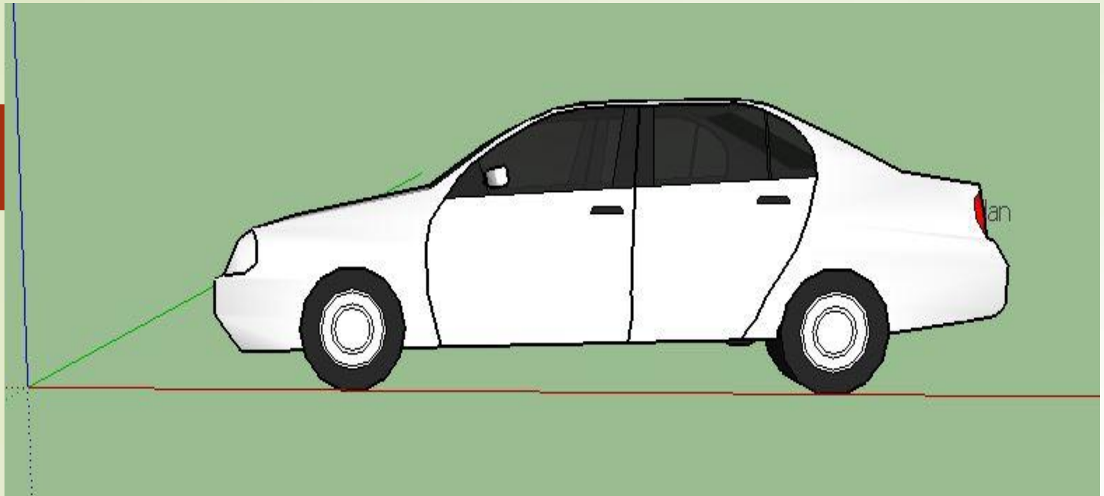
Car model





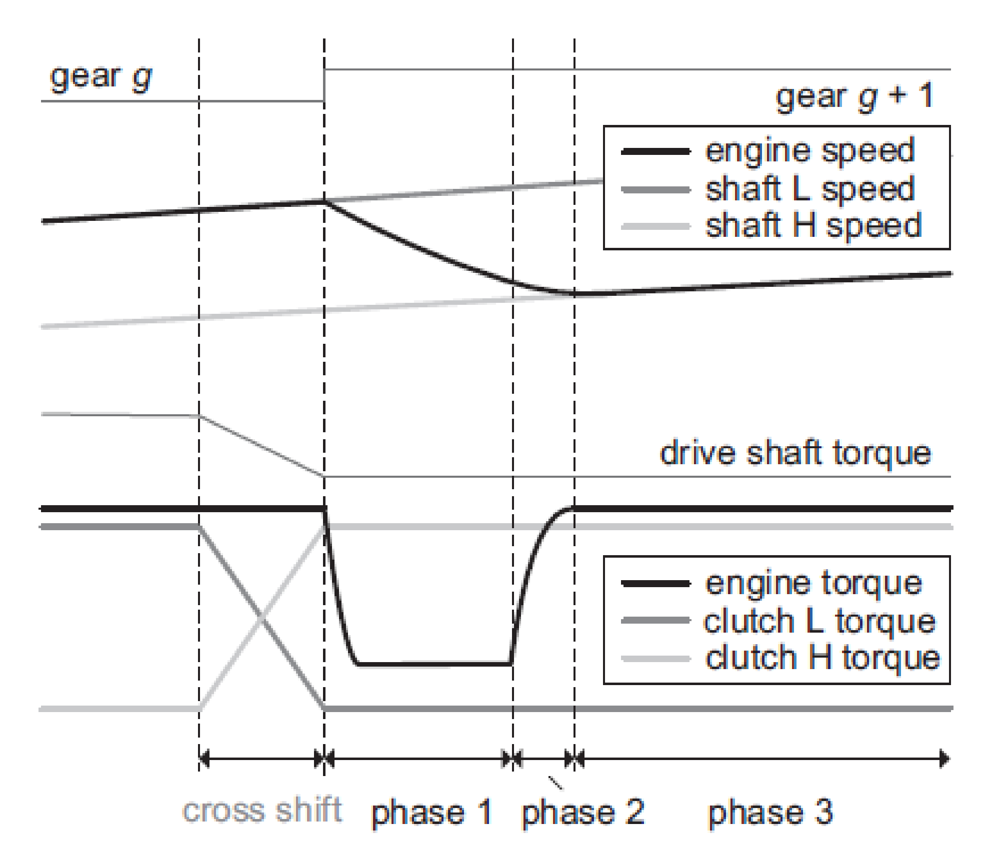


Chevrolet Corsa Sedan

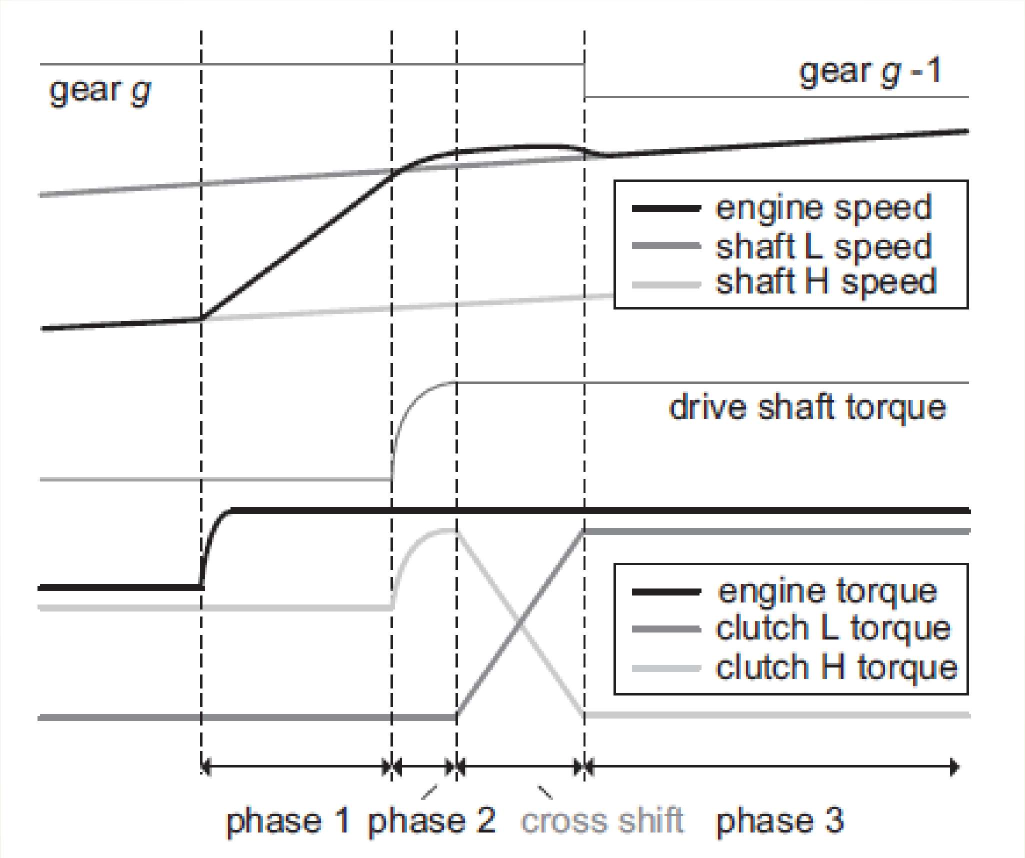
How actually dual clutch transmission system works !!

Driving mode

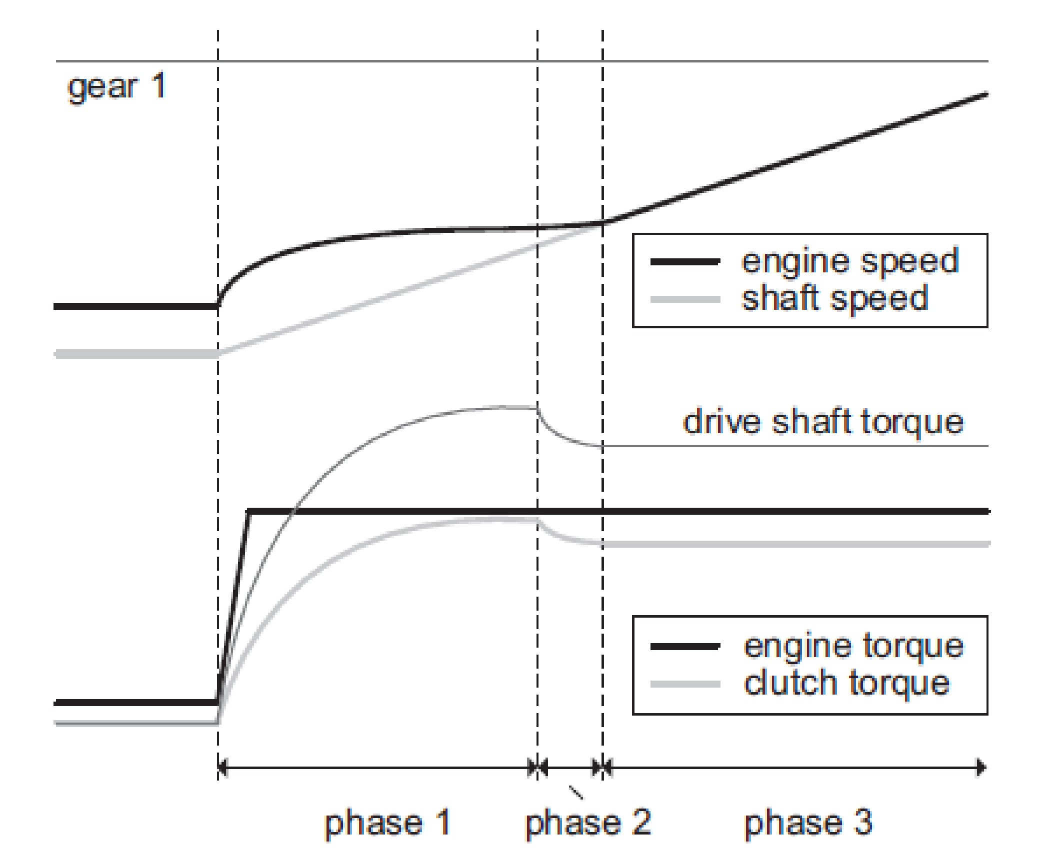
POWER UPSHIFT



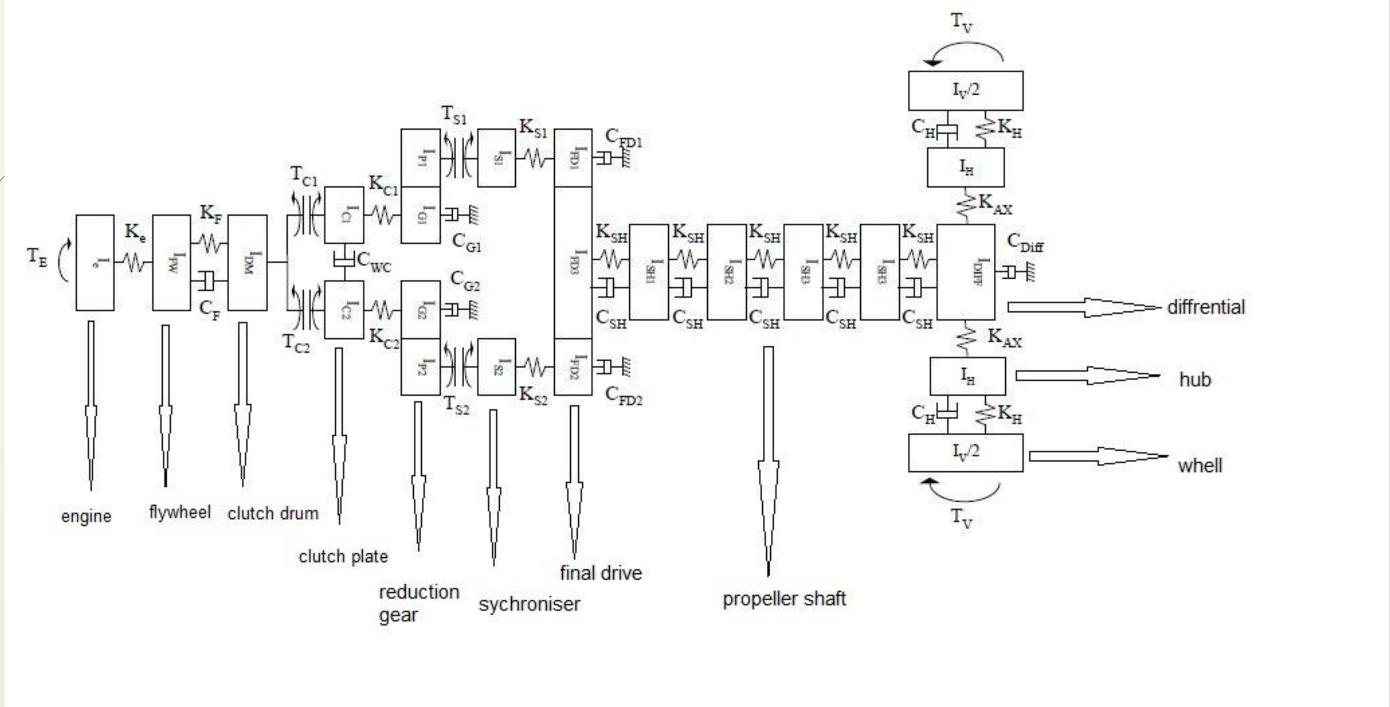
POWER DOWNSHIFT



VEHICLE LAUNCH

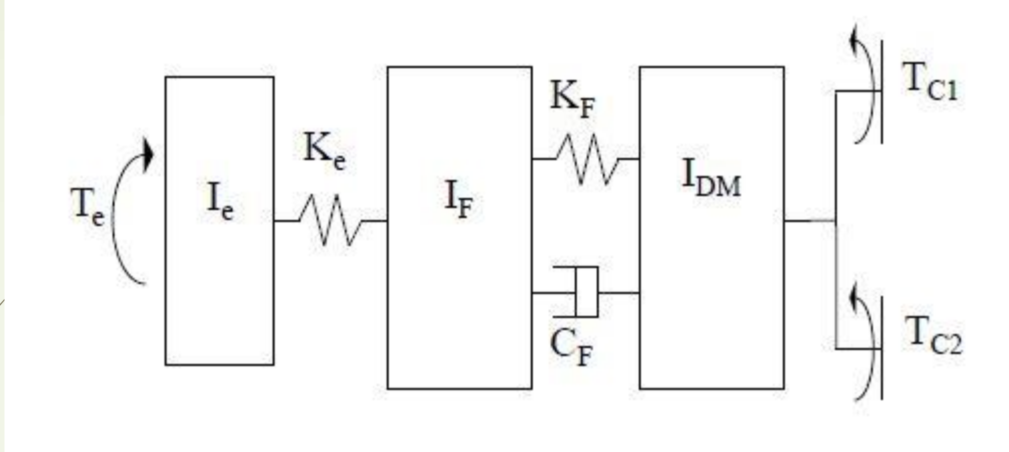


Lumped parameter model for a DCT equipped Powertrain



Fifteen degree of freedom model

Engine, Flywheel, and Clutch drum model elements



By Newton second law , we get the equation of motion for each element is as follow

Equation of motion :

$$I_e \ddot{\theta}_e - K_e(\theta_F - \theta_e) = T_e$$

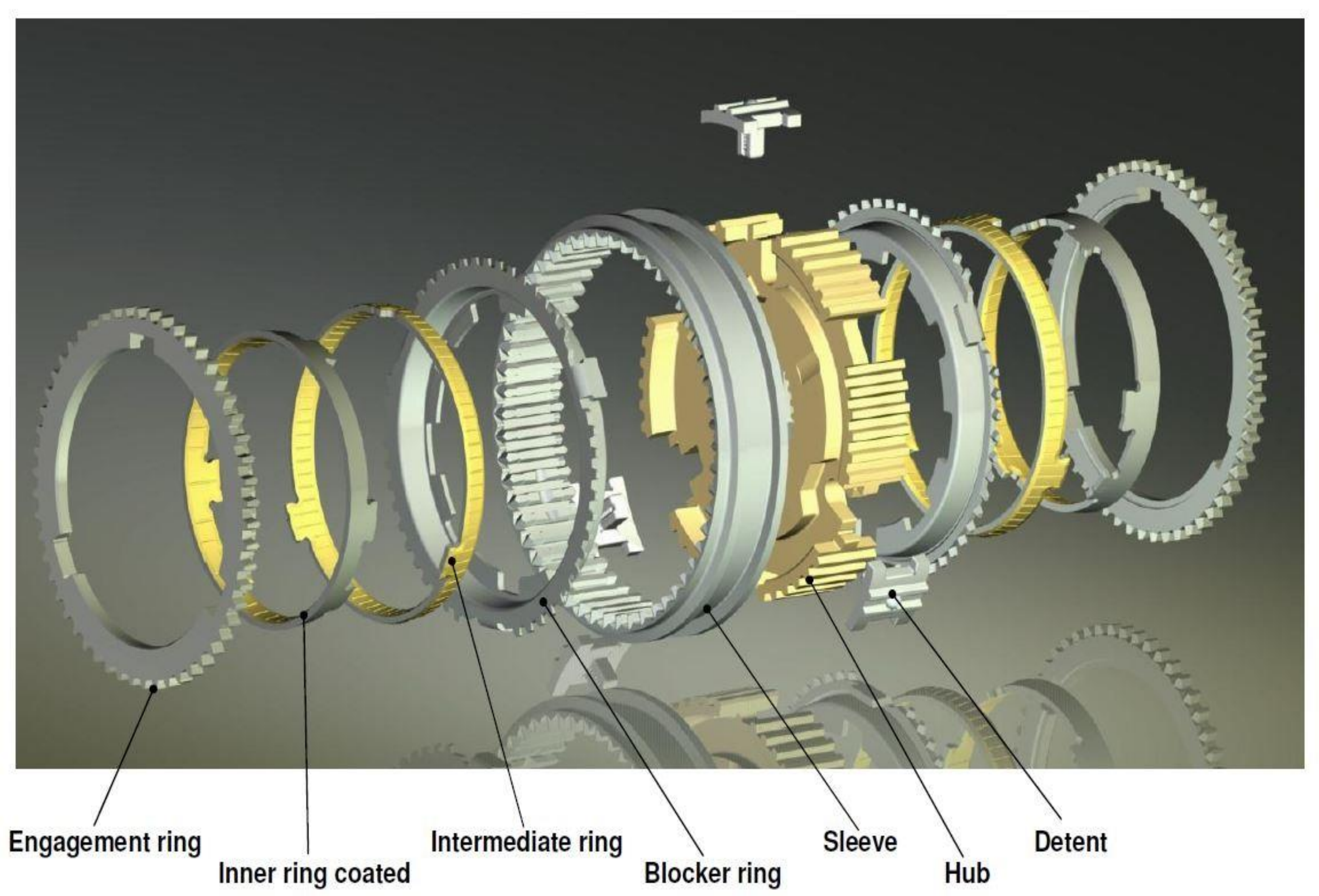
$$I_F \ddot{\theta}_F + K_e(\theta_F - \theta_e) - K_F(\theta_{DM} - \theta_F) - C_F(\dot{\theta}_{DM} - \dot{\theta}_F) = 0$$

$$I_{DM} \ddot{\theta}_{DM} + K_F(\theta_{DM} - \theta_F) + C_F(\dot{\theta}_{DM} - \dot{\theta}_F) = -(T_{C1} + T_{C2})$$

Synchronizer

- Function
- Types of Synchronizer
  - Single-cone Synchronizer
  - Dual-cone Synchronizer
  - Triple-cone Synchronizer

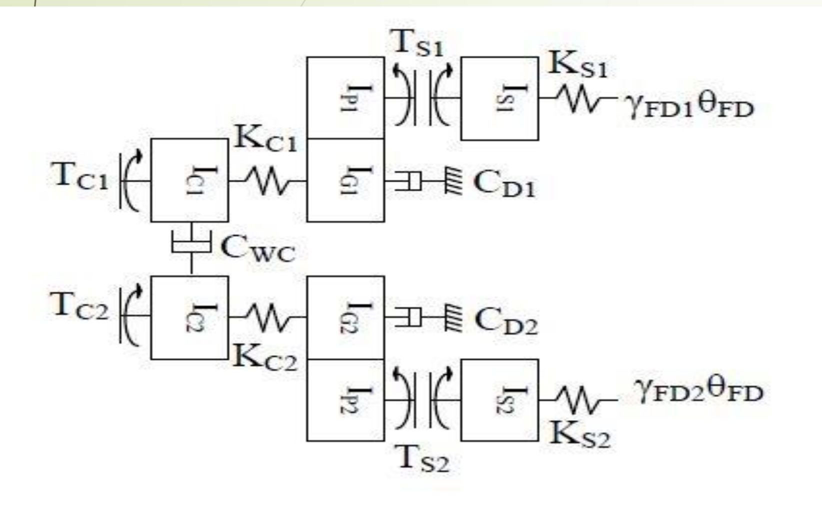
Components of Synchronizer



Different Steps of Synchronization-

- Disengagement
- Neutral
- Neutral Detent
- Pre Synchronization
- Synchronizing
- Synchronization
- Blocking Release
- Engagement Tooth contact
- Full Engagement

Clutch and simple transmission model elements



For each possible gear of the elements will vary based on the location within the transmission, and the inertia of gears and pinions will also vary depending on the gear selected.

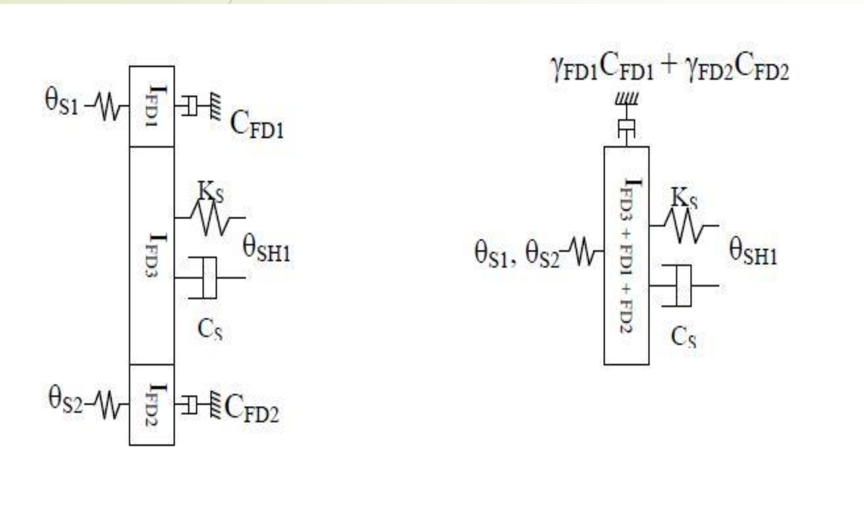
Equation of motion :

$$I_{C1}\ddot{\theta}_{C1} - K_{C1}(\theta_{G1} - \theta_{C1}) - C_{WC}(\dot{\theta}_{C2} - \dot{\theta}_{C1}) = T_{C1}$$

$$\left[\frac{I_{P1}}{\gamma_1^{2}} + I_{G1}\right]\ddot{\theta}_{G1} + K_{C1}(\theta_{G1} - \theta_{C1}) - C_{D1}\dot{\theta}_{G1} = -\frac{T_{S1}}{\gamma_1}$$

$$I_{S1}\ddot{\theta}_{S1} - K_{S1}(\gamma_{FD1}\theta_{FD3} - \theta_{S1}) - C_{SD1}\dot{\theta}_{S1} = T_{S1}$$

Final drive and reduced differential model elements



In final drive gears are used to link independent lay shafts to a single output shaft.
Thus the final drive is the integration of three inertial gear components, utilizing shaft stiffness elements to link the gear set to both the transmission and the drive shaft.

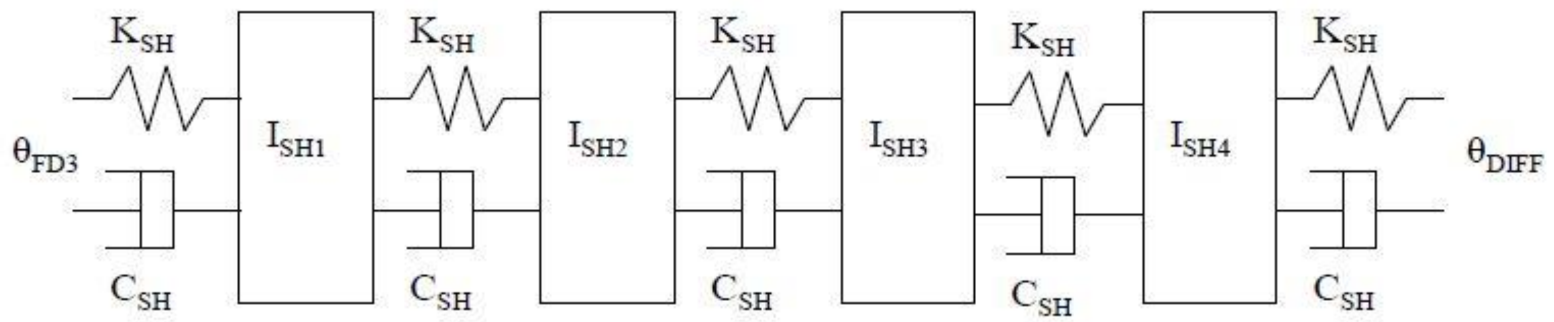
Equation of motion :

$$(\gamma_1{}^2 . I_{FD1} + \gamma_2{}^2 . I_{FD2} + I_{FD3})\ddot{\theta}_{FD3} + \gamma_{FD1} K_{S1}(\gamma_{FD1} . \theta_{FD1} - \theta_{S1})\ldots$$

$$+ \gamma_{FD2} K_{S2}(\gamma_{FD2} . \theta_{FD2} - \theta_{S2}) - K_{FD3}(\theta_{SH1} - \theta_{FD3})\ldots.$$

$$-(\gamma_{FD1} C_{FD1} + \gamma_{FD2} C_{FD2})\dot{\theta}_{FD3} - C_{SH}(\dot{\theta}_{SH1} - \dot{\theta}_{FD3}) = 0$$

Propeller shaft model



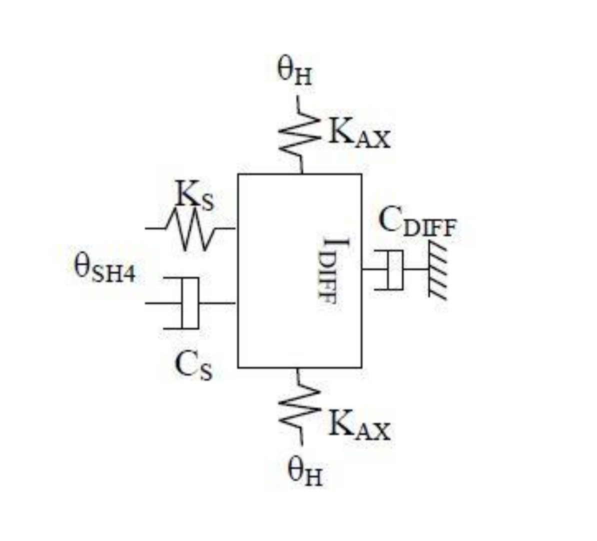
The shaft is modeled as a four degree of freedom system . Equations of motion for these shaft elements are identical as it is assumed that the shaft is of constant cross-sectional geometry.

Equation of motion :

$$I_{SH1}\ddot{\theta}_{SH1}+K_{SH}(\theta_{SH1}-\theta_{FD3})-K_{SH}(\theta_{SH2}-\theta_{SH1})\ldots$$

$$+C_{SH}(\dot{\theta}_{SH1}-\dot{\theta}_{FD3})-C_{SH}(\dot{\theta}_{SH2}-\dot{\theta}_{SH1})=0$$

Differential, axle, and wheels and tyre models



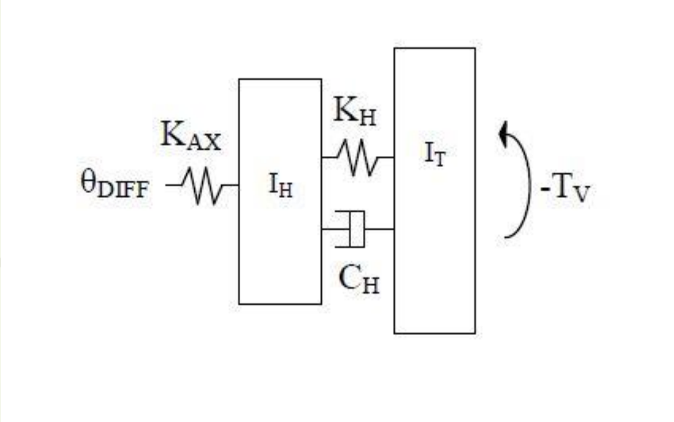
The differential and axle splits the drive torque to both rear wheels .
The differential is modeled as a lumped mass with damping to ground. Stiffness elements connect Propeller shaft and axles to the differential.

Equation of motion :

$$I_{DIFF}\,\ddot{\theta}_{DIFF} + K_{SH4}(\theta_{DIFF} - \theta_{SH4}) - 2K_{AX}(\theta_{H} - \theta_{DIFF})....$$

$$+C_{SH}(\dot{\theta}_{DIFF} - \dot{\theta}_{SH4}) - C_{DIFF}\,\dot{\theta}_{DIFF} = 0$$

Hub and Tyre model elements



The wheel model integrates the hub and tyre inertia with the flexural rigidity of the tyre wall.

Equation of motion :

$$2I_H \ddot{\theta}_H + 2K_{AX}(\theta_H - \theta_{DIFF}) - 2K_H(\theta_T - \theta_H) - 2C_H(\dot{\theta}_T - \dot{\theta}_H) = 0$$

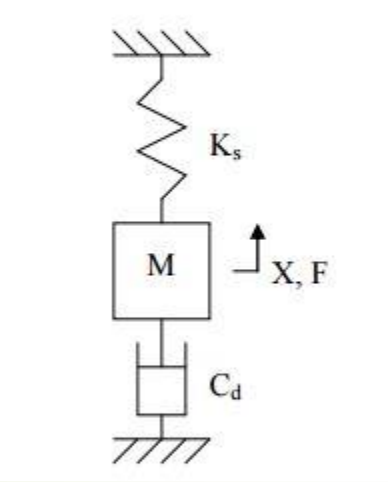
$$2I_T \ddot{\theta}_T + 2K_H(\theta_T - \theta_H) + 2C_H(\dot{\theta}_T - \dot{\theta}_H) = -T_V$$

Hydraulic Control System Modeling

- The hydraulic transmission control unit (TCU) in the dual clutch transmission is employed to perform two functions.

1) clutch-to-clutch power shifting of gears
2) engagement of the synchronizer mechanism

- Detailed mathematical models of both hydraulic systems are required for shift control.



- Consider simple 1 DOF system. Equation of motion for this system can be written as,

$$M\ddot{X} + C_d\dot{X} + K_s X = \sum F$$

- The force inputs, ΣFx, into the model are derived from several sources. This includes feedback damping where control volumes at either ends of the spool provide pressure forces that counter the motion of the spool. The other force main being the magnetic induction derived from the coil windings in the solenoid.

- The flow rates into and out of control volumes are calculated using the sharp edged orifice flow equation, defined below

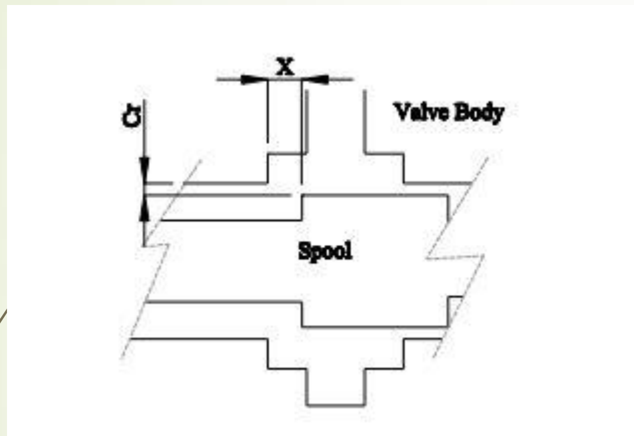
$$Q_o = C_D \pi \frac{D^2}{4} \sqrt{P_2 - P_1}$$

- The leakage flow can be calculated in a similar manner to the orifice equation

$$Q_L = C_D \pi D c_r \sqrt{P_2 - P_1}$$

Where C_D = Coefficient of discharge

c_r = Radial clearance



- For open port this is written as

$$Q_P = C_D \pi D \sqrt{(X^2 + c_r^2)(P_2 - P_1)}$$

- Particular to the feedback volumes and the clutch pack is the rate of change in volume with the spool or piston motion.

$$Q_V = A_p \dot{X}$$

- The last source of change in flow arises from the fluid compressibility.

$$Q_C = \frac{V}{\beta}\frac{dP}{dt}$$

- Above equations are combined to provide the net flow into or out of any control volume in a hydraulic system. Through mass conservation it is assumed that:

$$\sum Q = 0$$

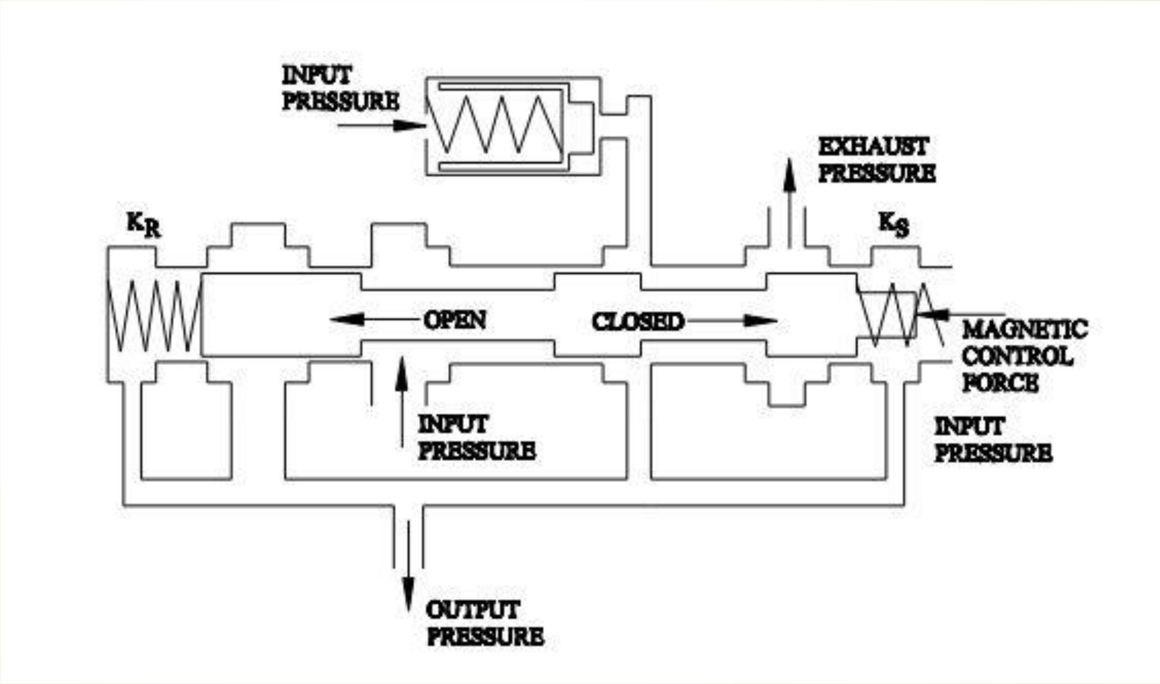
$$Q_O + Q_P + Q_L + Q_V + Q_C = 0$$

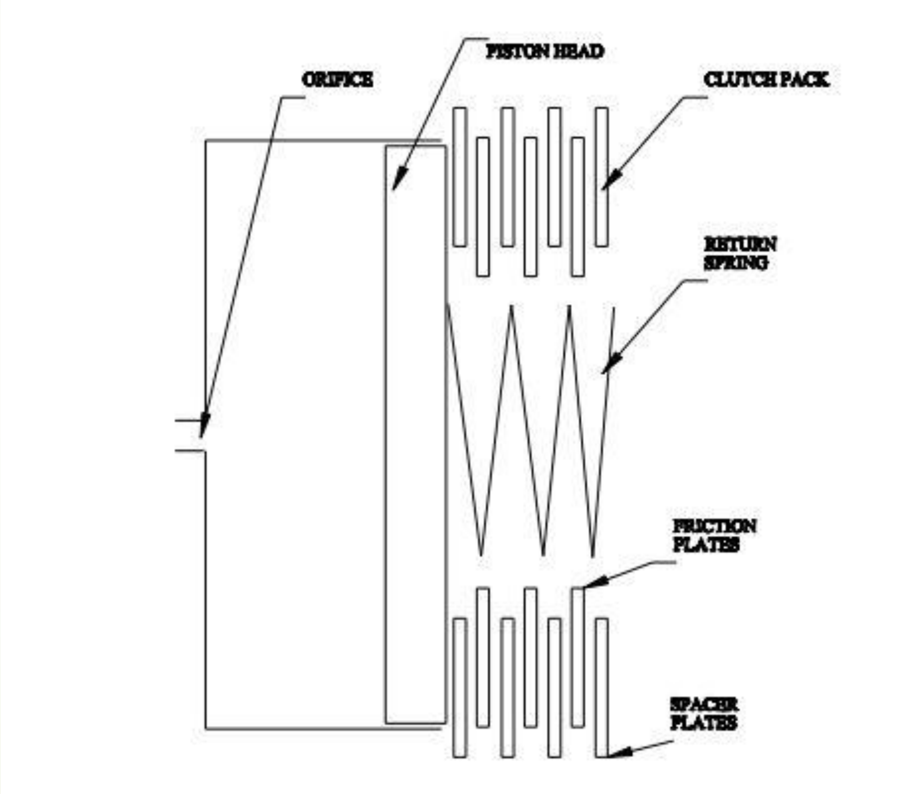
$$\frac{V}{\beta}\frac{dP}{dt} + A_P \dot{X} + C_D \pi D \sqrt{(X^2 + c_r^2)(P_2 - P_1)} + C_D \pi D c_r \sqrt{P_2 - P_1} + C_D \pi \frac{D^2}{4}\sqrt{P_2 - P_1} = 0$$

- With the inclusion of variation in the control volume pressure is calculated as:

$$P = \int \frac{\beta}{V}\left(A_P \dot{X} + C_D \pi D \sqrt{(X^2 + c_r^2)(P_2 - P_1)} + C_D \pi D c_r \sqrt{P_2 - P_1} + C_D \pi \frac{D^2}{4} \sqrt{P_2 - P_1} \right) dt$$

- By using this equation we can calculate the pressure of different control volumes in electro-hydraulic control system.
- Now we will see how this hydraulic system works:


INPUT
PRESSURE
EXHAUST
PRESSURE
K_R
K_S
OPEN
CLOSED
MAGNETIC
CONTROL
FORCE
INPUT
PRESSURE
INPUT
PRESSURE
OUTPUT
PRESSURE


PISTON HEAD
ORIFICE
CLUTCH PACK
RETURN
SPRING
FRICTION
PLATES
SPACER
PLATES

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Assistant Professor
IITH

REFERENCES:-

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2) Modelling of dual clutch transmission equipped powertrains for shift transient simulations; Paul D. Walker , Nong Zhang.

THANK

YOU