

Torque Gap Filler for Automated Manual Transmissions: Principles for the Development of the Control Algorithm

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ABSTRACT

One of the most significant tasks in automotive design is related to the implementation of gearboxes capable of reducing the torque gap during the gearshift process and, at the same time, not decreasing vehicle performance from the point of view of driveline efficiency. Automated gearboxes based on torque converters ([1], [2]) satisfy the first requirement but not the second. On the other hand, manual automated gearboxes ([3], [4], [5], [6]) satisfy the requirements in terms of consumption, due to the absence of the dissipations caused by the torque converter. In fact, they consist of the basic layout of a manual transmission with hydraulic or electro-mechanical actuators which are adopted for the clutch and the synchronizers. However, automated manual transmissions cannot guarantee optimal longitudinal dynamics of the vehicle due to the discontinuity in torque transmission when the clutch is disengaged. New solutions are based on the adoption of Dual Clutch systems ([7], [8], [9]) characterized by a significant mechanical complexity. This paper presents, from the points of view of basic concepts and vehicle dynamics simulation, the Torque Gap Filler, a new system patented by the author ([10], [11]) capable of joining the advantages of the systems described in [1] - [9] without their disadvantages. First of all, the basic concepts of the system are described in detail. Then the basic equations necessary for the development of the control algorithm of the system are presented, and finally a discussion about the possible benefits of the system for two different mechanical layouts are discussed.

SYSTEM LAYOUT

Torque Gap Filler (TGF) is a system which can be added to the conventional layout of a manual gearbox, without a significant variation to the external size of the system. From a practical point of view, it could be implemented by adopting the two distinct physical configurations presented in Figures 1 and 2. The basic concept of TGF consists of connecting, during gearshift, the engine shaft and the secondary shaft of a manual gearbox through an epicyclical gearbox. The three elements of the epicyclical group which are connected with the rest of the driveline are the sun gear, the ring gear and the carrier. One of these elements is connected to the engine flywheel, another one to the secondary shaft of the gearbox and the third one to a hydraulic brake, which (if activated) permits the transmission of the torque to the secondary shaft of the gearbox. In configuration 1 (Figure 1), the flywheel is connected to the sun gear and the brake to the ring gear. In configuration 2 (Figure 2), the flywheel is connected to the ring gear and the brake is connected to the sun gear.

When the driveline is not performing a gearshift, the brake is released and the components of the epicyclical gearbox transmit zero torque (with the exception of the friction losses), depending on the inserted gear ratio. The brake has to be properly activated when the vehicle performs an upshift. A possible target for the control algorithm could be to achieve, during the gearshift, the same level of longitudinal acceleration (perhaps after a short transition) compared to a vehicle with the new gear ratio inserted with the conditions of pedal throttle position currently imposed by the driver. On a vehicle equipped with an automated manual transmission, the driver does not have to release the throttle pedal during the gearshift.

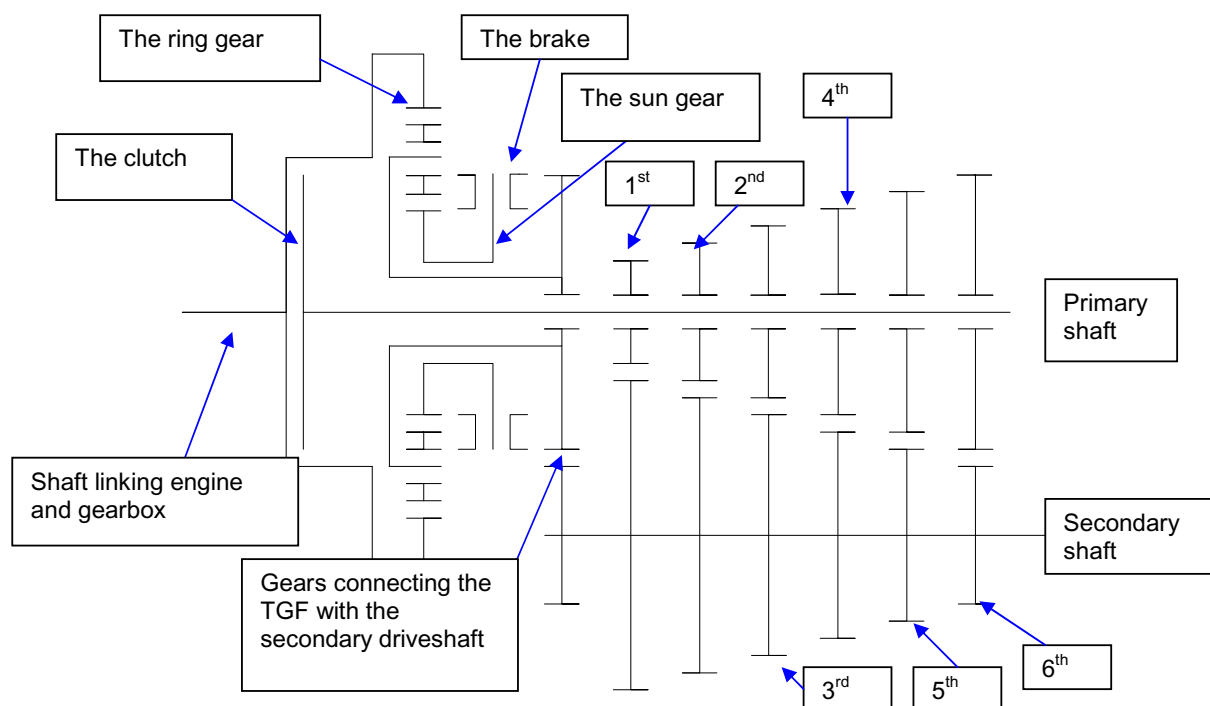
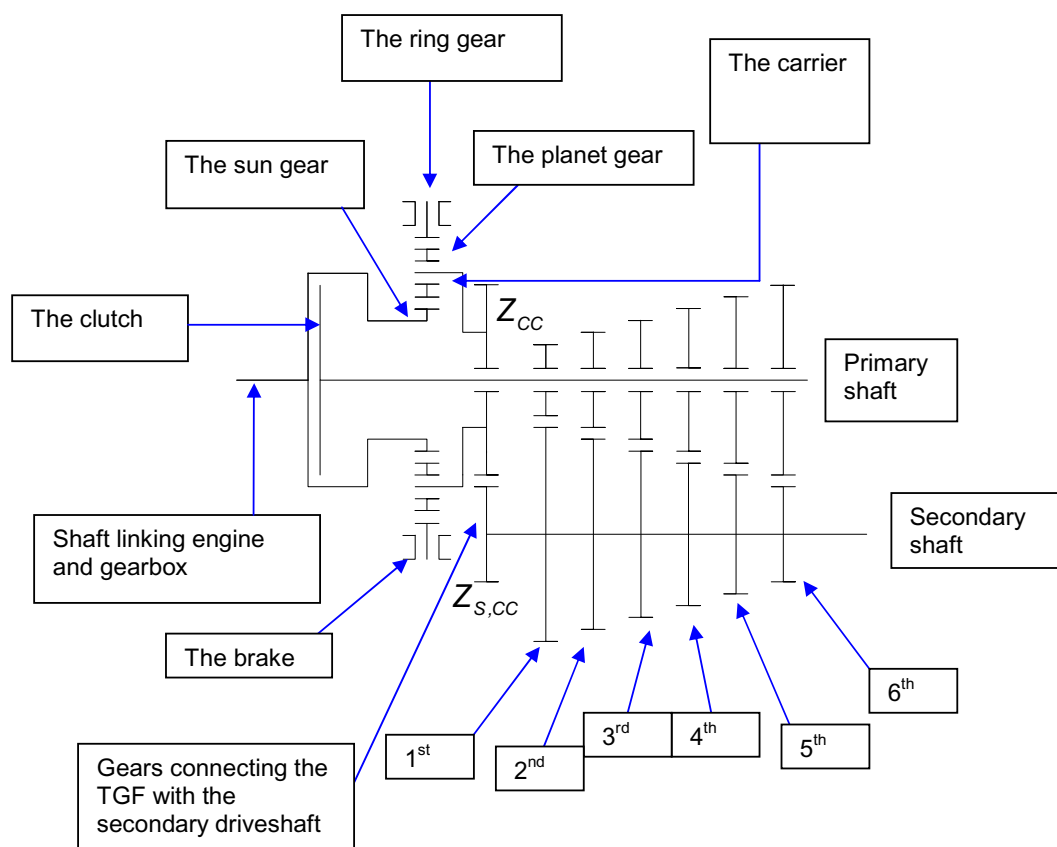


Figure 2 – Second possible layout for the implementation of the Torque Gap Filler (TGF Type II)

The control algorithm of the TGF has three main outputs, the first one in terms of reference brake torque, the second one in terms of required engine torque and the third one in terms of clutch disc axial displacement (a devoted sensor is usually adopted in the case of an automated manual transmission).

BASIC EQUATIONS GOVERNING THE SYSTEM

In this section the basic equations for the system are presented under the hypothesis of configuration 1. These equations can be adopted as the basis for the design of the control algorithm. In a very first approximation, if driveline dynamics are not considered, vehicle inertia can be supposed as if it were directly linked to the secondary shaft of the gearbox. The effect of the clutch damper and half shafts is not considered for this first approximation model devoted to the development of the control algorithm. Under this hypothesis and without considering synchronization dynamics, the power-train of a vehicle equipped with the TGF can work in two different states:

- slipping clutch;
- engaged clutch.

At this step, TGF dynamics are taken into account by considering the inertia J_3 of the brake disc.

Slipping Clutch

In this condition, the system can be represented by the following equations:

$$T_{engine} - J_{eng} \frac{d\omega_{eng}}{dt} = F_1 R_1 + T_{clutch} \quad (1)$$

where T_{engine} is engine torque, J_{eng} is engine inertia, $\frac{d\omega_{eng}}{dt}$ is engine angular acceleration, F_1 is the force between the teeth of the gears of the epicyclic gearbox (under the hypothesis of unit efficiency), R_1 is sun gear radius, T_{clutch} is the torque transmitted by the clutch. Vehicle longitudinal dynamics are given by:

$$\begin{aligned} T_{out,TGF} \frac{Z_{S,CC}}{Z_{CC}} \rho_P - T_{res} + T_{clutch} \rho_{Cx} \rho_P &= \\ &= J_{veh} \frac{d\omega_{veh}}{dt} \end{aligned} \quad (2)$$

where $T_{out,TGF}$ is the output torque from the carrier of the epicyclic gearbox, $Z_{S,CC}$ is the number of the teeth of the gear on the secondary shaft of the gearbox (Figure 1), Z_{CC} is the number of the teeth of the gear on the carrier (Figure 1), ρ_P is the final gear ratio at the differential, T_{res} is the torque due to rolling resistance of

the tires, aerodynamic drag and longitudinal slope of the road, ρ_{Cx} is the current gear ratio, J_{veh} is the equivalent inertia of the vehicle and $\frac{d\omega_{veh}}{dt}$ is wheel angular acceleration. In a first approximation, $T_{out,TGF}$ can be expressed by:

$$T_{out,TGF} = F_1 (R_1 + R_3) \quad (3)$$

where R_3 is the radius of the ring gear of the TGF. The dynamics of the brake disc and the ring gear is given by:

$$-F_1 R_3 + T_{BRAKE} = J_3 \frac{d\omega_3}{dt} \quad (4)$$

where T_{BRAKE} is the applied brake torque (according to the control algorithm of the TGF), J_3 is brake disc and

ring gear inertia, $\frac{d\omega_3}{dt}$ is ring gear angular acceleration.

The Willis formula for epicyclic gearboxes can be applied to this system (where Ω is the angular velocity of the carrier):

$$\Omega = \frac{\omega_1 R_1 + \omega_3 R_3}{R_1 + R_3} \quad (5)$$

By combining equations (1)-(5), it is possible to find out the two equations which govern the entire system. The equation which permits the computation of the dynamics of the brake is:

$$\begin{aligned} T_{BRAKE} \left(\frac{\rho_P Z_{S,CC}}{Z_{CC}} \frac{Z_1 + Z_3}{Z_3} + \frac{J_{veh} Z_{CC} Z_1^2}{Z_{S,CC} J_3 \rho_P Z_1 (Z_1 + Z_3)} \right) + \\ - T_{engine} \frac{J_{veh} Z_{CC} Z_1}{Z_{S,CC} J_1 \rho_P (Z_1 + Z_3)} + \\ - T_{res} + T_{clutch} \left(\rho_{Cx} \rho_P + \frac{J_{veh} Z_{CC} Z_1}{J_1 Z_{S,CC} \rho_P (Z_1 + Z_3)} \right) = \\ = \frac{d\omega_3}{dt} \left(\frac{J_3 Z_{S,CC} \rho_P (Z_1 + Z_3)}{Z_3 Z_{CC}} + \frac{J_{veh} Z_{CC}}{\rho_P Z_{S,CC}} \right) \end{aligned} \quad (6)$$

Where Z_1 and Z_3 are the number of teeth which are proportional to the radii R_1 and R_3 .

Equation (7) is the dynamic equilibrium of engine driveshaft:

$$\begin{aligned} \frac{T_{engine} - T_{clutch}}{J_{eng}} - T_{BRAKE} \frac{Z_1}{J_{eng} Z_3} + \frac{d\omega_3}{dt} \left(\frac{J_3 Z_1}{J_{eng} Z_3} \right) = \\ = \frac{d\omega_{eng}}{dt} \end{aligned} \quad (7)$$