Vehicle’s transmission as a vibro-impact system

George Korendyasev and Konstantin Salamanda
Mechanical Engineering Research Institute of the Russian Academy of Sciences (IMASH RAN), Maliy Kharitonievsky pereulok, 4, Moscow 101990, Russia
ksalamanda@yandex.ru

Abstract. The article discusses a dynamic model of a dual-clutch transmission. The absence of a torque converter and the short duration of gear shifting (0.2 - 0.5 sec) in such transmissions allows to present the shifting process as impact interaction. As a result, the gear shift excites the oscillations of the gearbox output shaft. Thus, the transmission of the vehicle can be considered as a vibro-impact self-oscillating system.

1. Introduction

Requirements for increasing efficiency and reducing emissions of harmful substances from vehicle engines have led, in particular, to the fact that gear shifting takes 0.2 - 0.5 seconds in modern automatic transmissions [1, 2]. The gearshift process occurs without interrupting the power flow - by simultaneously switching off one friction clutch and switching on another clutch. A quick shifting process even with a small difference in the velocities of the engine and the output shaft of the gearbox leads to a sharp change in inertial moments, which in turn can impair the passengers’ comfort, reduce the service life and reliability of the transmission.

A torque converter is installed between the engine output shaft and gearbox in automatic transmissions consisting of planetary mechanisms [1-3]. The rotation of the turbine wheel connected to the output shaft of the engine is transmitted to the pump wheel connected to the input shaft of the gearbox in the torque converter using oil. The absence of a rigid connection between the engine and the gearbox allows the torque converter to damp a sharp change in inertial moments caused by short-term gear changes. Low efficiency is the main disadvantage of the torque converter.

The dual-clutch transmissions [4, 5] is widely used currently, in which there is no torque converter, and the output shaft of the engine and the input shaft of the gearbox are connected by multi-plate friction clutches controlled by servomotors. The use of clutches instead of a torque converter allows to increase efficiency, but in view of the short duration of the gear shifting process, it leads to impact interaction of transmission parts.

The kinematic diagram of a 6-speed dual-clutch gearbox [6] is shown in figure 1. In the diagram in figure 1: I - input shaft; O1, O2 - gears meshed with the output shaft of the gearbox (not shown in the diagram); c1, c2 - multi-plate friction clutches; s31, s24, s5 and sR6 are synchronizers, engaging the corresponding transmission.

The branched arrangement of dual-clutch transmissions allows gear shifting in an unloaded branch, that is, the next stage can be preselected in advance before it is directly switched on. Conventionally, such a transmission can be presented in the form of two gearboxes. The corresponding multi-plate friction clutch located at the input of the gearbox is responsible for alternately connecting them to the input shaft. Structurally, both clutches are made in a single unit. For example, if the vehicle moves on
the fourth gear and continues to accelerate, the synchronizer responsible for the fifth gear can be preselected in advance and, as soon as the necessary gear shifting speed to the next gear is reached, the clutch c2 switched off and the clutch c1 switch on.

**Figure 1.** Kinematic diagram of a 6-speed gearbox with two clutches.

Currently, dynamic analysis of the gear shifting process is carried out sequentially by dividing it into stages [1, 2, 7]. The differential equations are compiling whose solutions determine the general picture of changes in velocities and torques for each stage. This approach is fully justified for the dynamic analysis of gear shifting processes in manual gearboxes, in which the gearshift time is seconds and all phases of the process can be described correctly.

A sequential dynamic analysis of rapid gearshift processes in modern automatic transmissions leads to useful and interesting results, but is associated with a number of assumptions about the interactions of the gearbox clutches. It is often assumed that the moment of inertia of the rotating links of the vehicle after the gearbox is disproportionately large compared to the moment of inertia of the engine links. Such an assumption halves the order of the equations being solved, but introduces errors in the analysis results. In addition, it is quite difficult to divide a short gearshift time even into two stages for sequential analysis. It is also obvious that, the gear shift really acquires the properties of impact interaction at small values of the gear shift duration. The inertial moments caused by a sharp change in the velocities of the transmission elements in this case significantly exceed the engine moments and the reduced moments of the resistance to movement.

It should be noted that in the first half of the 20th century, to determine the duration of the gear shifting process in the gearboxes and the influence of inertial parameters and link velocities on impact loads, the theorem on the change in angular momentum have been used [8]. The accuracy of the quantitative assessment of the impact moment for a manual gearbox was not large, since manual gear shifting is calculated in seconds, and unaccounted for moving torques and moments of resistance forces to the movement are almost of the same order as the moments of inertia of the rotating links. In view of the short duration of the gear shifting process in modern gearboxes, for the dynamic analysis of this process, in this paper it is proposed to return to using the theorem on the change in the angular momentum of a mechanical system upon impact.
2. Transmission dynamic model

Consider the simplified transmission structure in figure 2, comprising an engine, a two-speed gearbox with two clutches (c1, c2). The output shaft of the gearbox is connected to the differential by an elastic link with a stiffness coefficient c and damping b. The differential distributes the rotation to the drive wheels of the vehicle. The rotation is transmitted to the intermediate shaft through the gear wheel of the first gear with the gear ratio \(i_1\) when the clutch c1 is switched on and to the intermediate shaft through the gear wheel of the second gear with the gear ratio \(i_2\) when the clutch c2 is switched on. The rotation is transmitted with the gear ratio \(i\) from the intermediate shaft to the output shaft.

The engine moving parts’ moment of inertia reduced to the housing of the clutch is denoted by \(J_I\); the gearbox moving parts’ moment of inertia before the link with maximum flexibility reduced to the output shaft is denoted by \(J_O\); \(J_V\) element is the reduced to the output shaft of the gearbox moment of inertia of the transmission links after the link with maximum flexibility, including the differential, half shafts, drive wheels, and the vehicle body. Thus, the obtained dynamic model contains three inertial elements: \(J_I\) and \(J_O\) are rigidly connected using clutches c1 or c2, and the element \(J_V\) is connected to the element \(J_O\) by an elastic bond. We show below that the resulting model is a self-oscillating vibro-shock system.

Assume the movement of the model in figure 2 on first gear and consider shifting to the second gear. The driving torque of the engine and the moment of the resistance forces have been excluded from consideration because they do not significantly affect the changes in the velocities of the inertial links of the model for a short period of time during the engagement of the clutch c2. We also accept that the pressure relief in the hydraulic actuator of the switched-off clutch c1 occurs instantly, and the residual torque transmitted by it is negligible and therefore also not taken into account. Thus, only the clutch c2 is involved in the gear shift process under consideration.

There is a difference in the rotational velocities of the inertial elements \(J_I\) and \(J_O\) at the initial time of the process of engaging clutch c2. The velocities of the clutch housing and its disks are equalized when the clutch c2 is switched on for a short time. The moment of frictional forces between the clutch disks and the housing, which depends on the compression force and the friction coefficient, eliminates slipping between the disks in this case and are internal, applied to both parts of the model with different signs.
The model under consideration in view of the foregoing in a short time interval is a closed mechanical system for which the theorem on the change in the angular momentum upon impact is applicable.

The vehicle velocity sensor is located on the gearbox output shaft. Let us denote the values of the velocities \( \omega_0 \) specified in the gearshift schedule, at which they should be switched: \( \omega_0^{(12)} \) — from the first gear to the second and \( \omega_0^{(21)} \) — from the second gear to the first. Moreover, it is necessary that the acceleration is \( \dot{\omega}_0 > 0 \) to shift from the first gear to the second, and \( \dot{\omega}_0 < 0 \) when shifting from the second gear to the first.

The rotational velocity \( \omega_i \) of the input shaft of the gearbox is related to the rotational velocity \( \omega_0 \) of the output shaft by the gear ratios: \( \omega_i^{(12)} = i_1 \omega_0^{(12)} \) when switching from first gear to second, \( \omega_i^{(21)} = i_2 \omega_0^{(21)} \) when shifting from second gear to first. The velocity of the element \( f_0 \) before gear shifting, for example, from the first gear to the second \( \omega_0^{(12)} \), and the velocity of the element \( f_1 \) at that moment \( i_1 \omega_0^{(12)} \). The moment of inertia of the element \( f_1 \) reduced to the gearbox output shaft before shifting on the second gear is equal to \( f_1 i_2^2 l^2 \).

The theorem on the change in the angular momentum for the inertial elements \( f_1 \) and \( f_0 \) before and after the second gear is engaged:

\[
J f_1 i_2^2 l^2 \frac{\omega_0^{(2)}}{i_2} + J_0 \omega_0^{(12)} = (J f_1 i_2^2 l^2 + J_0) \omega_0^{(2)}
\]

where \( \omega_0^{(2)} \) is the velocity of the output shaft after shifting to the second gear. Moreover, \( \omega_i^{(2)} = i_1 \omega_0^{(12)} \), then we get:

\[
\omega_0^{(2)} = \frac{J f_1 i_2^2 l^2 + J_0}{J f_1 i_2^2 l^2 + J_0} \omega_0^{(12)}
\]

The change in velocity \( \Delta \omega_0^{(2)} \) at the output of the gearbox after the second gear is engaged:

\[
\Delta \omega_0 = \omega_0^{(2)} - \omega_0^{(12)} = \frac{i_1 - i_2}{J f_1 i_2^2 l^2 + J_0} J f_1 i_2^2 l^2 \omega_0^{(12)}
\]

(1)

As can be seen from (1), the change in the velocity of the output shaft is positive, because \( i_1 > i_2 \) when shifting from the lowest gear to the highest. In other words, the velocity at the output of the gearbox during the shift under consideration will increase.

Similarly, the theorem on the change in angular momentum for the inertial elements \( f_1 \) and \( f_0 \) before and after the first gear is turned on:

\[
J f_1 i_2^2 l^2 \frac{\omega_0^{(1)}}{i_1} + J_0 \omega_0^{(21)} = (J f_1 i_2^2 l^2 + J_0) \omega_0^{(1)}
\]

where \( \omega_0^{(1)} \) is the velocity of the output shaft after shifting to the first gear. Moreover, \( \omega_i^{(1)} = i_2 \omega_0^{(21)} \), then we get:

\[
\omega_0^{(1)} = \frac{J f_1 i_2^2 l^2 + J_0}{J f_1 i_2^2 l^2 + J_0} \omega_0^{(21)}
\]

The change in velocity \( \Delta \omega_0^{(1)} \) at the output of the gearbox after the first gear is engaged:

\[
\Delta \omega_0 = \omega_0^{(1)} - \omega_0^{(21)} = \frac{i_2 - i_1}{J f_1 i_2^2 l^2 + J_0} J f_1 i_2^2 l^2 \omega_0^{(21)}
\]

The velocity of the output shaft of the gearbox will decrease when shifting from second to first gear accordingly.
Thus, the ratios of changes in velocities during gear shifts in the impact part of the model in figure 2 are derived.

3. Self-oscillations of the gearbox output shaft

Consider the self-oscillation of the gearbox output shaft caused by a sharp change in the velocity of the output shaft when shifting gears and the presence of elastic and dissipative bonds.

The total moment of inertia of the engine and gearbox elements \( (J_1 + J_0) \) in the intervals between gear shifting could be considered after reducing the moment of inertia \( J_1 \) to the output shaft before the link with maximum flexibility \( (j \) is the number of the gear engaged). This eliminates one of the three degrees of freedom of the model in figure 2.

The motion of the model in the \( j \)-th gear between gear shifts is described by a system of two linear differential equations:

\[
(J_1 i^2 + J_0) \omega_o^{(j)} + b \left( \omega_o^{(j)} - \omega_v^{(j)} \right) + c \left( \varphi_o^{(j)} - \varphi_v^{(j)} \right) = iM_t
\]

\[
J_v \dot{\omega}_v^{(j)} + b \left( \omega_v^{(j)} - \omega_o^{(j)} \right) + c \left( \varphi_v^{(j)} - \varphi_o^{(j)} \right) = -M_R
\]

In system (2), \( \varphi_o \) are the angular coordinates of the inertial element \( (J_1 i^2 + J_0) \); \( \varphi_v \) - the angular coordinate of the output shaft after the link with maximum flexibility.

Free vibration frequencies of the model in the absence of damping: \( k_1 = 0 \); \( k_2^{(j)} = \sqrt{c(J_1 i^2 + J_0 + J_{VT})/(J_1 i^2 + J_0)J_v} \). Frequency of damped oscillations: \( \|k_2^{(j)} = \sqrt{k_2^{(j)} - h^{(j)}}, \)

where \( h^{(j)} = b(J_1 i^2 + J_0 + J_v)/2(J_1 i^2 + J_0)J_v \).

The movement of model elements between gear shifts is a torsional vibration of inertial elements \( (J_1 i^2 + J_0) \) and \( J_v \) relative to the center of mass, rotating under the action of the angular momentum and the moments of resistance forces \( M_t = \text{const} \neq 0, M_R = \text{const} \neq 0 \). The motion of the center of mass of the model is the figurative movement of the model. The relative motion of the elements \( (J_1 i^2 + J_0) \) and \( J_v \) are their synchronous torsional vibrations in antiphase, and the magnitudes of the oscillation amplitudes are inversely proportional to the moments of inertia [9]. The general solution of the obtained equations (2) has the form:

\[
\varphi_o^{(j)} = A^{(j)} e^{-h^{(j)}t} \sin(k_2^{(j)} t + \beta^{(j)}) + C_1^{(j)} + C_2^{(j)} t + a^{(j)} t^2/2
\]

\[
\varphi_v^{(j)} = \mu^{(j)} A^{(j)} e^{-h^{(j)}t} \sin(k_2^{(j)} t + \beta^{(j)}) + C_1^{(j)} + C_2^{(j)} t + a^{(j)} t^2/2
\]

where \( \mu^{(j)} = A_v^{(j)} / A^{(j)} = -(J_1 i^2 + J_0)/J_v \), \( a^{(j)} = (iM_t - M_R)/(J_1 i^2 + J_0 + J_v) \).

The constants \( A, \beta, C_1 \) and \( C_2 \) are determined through the initial angular coordinates and velocities of the inertial elements of the model. In (3), parameters \( C_1, C_2 \) and \( \alpha \) - characterize the motion of the center of mass of the model, parameters \( A \) and \( \beta \) - characteristics of the relative motion.

A general form of the equations of velocity of masses on the \( j \)-th gear between gear shifts can be obtained by differentiating equations (3):

\[
\omega_o^{(j)} = A^{(j)} e^{-h^{(j)}t} \left( k_2^{(j)} \cos(k_2^{(j)} t + \beta^{(j)}) - h^{(j)} \sin(k_2^{(j)} t + \beta^{(j)}) \right) + C_1^{(j)} + a^{(j)} t
\]

\[
\omega_v^{(j)} = \mu^{(j)} A_v^{(j)} e^{-h^{(j)}t} \left( k_2^{(j)} \cos(k_2^{(j)} t + \beta^{(j)}) - h^{(j)} \sin(k_2^{(j)} t + \beta^{(j)}) \right) + C_2^{(j)} + a^{(j)} t
\]
\( \mathcal{C}_2^{(j)} \) is the initial velocity of the center of mass of the model at the \( j \)-th step. The expressions of mass velocities are transformed to the form after substituting \( \sin \alpha^{(j)} = h^{(j)}/\sqrt{h^{(j)}^2 + k_2^{(j)} \gamma^2} \) and \( \cos \alpha^{(j)} = k_2^{(j)} / \sqrt{h^{(j)}^2 + k_2^{(j)} \gamma^2} \):

\[
\omega_0^{(j)} = A^{(j)} e^{-h^{(j)} t} \sqrt{h^{(j)}^2 + k_2^{(j)} \gamma^2} \cos \left( k_2^{(j)} t + \beta^{(j)} + \alpha^{(j)} \right) + C_2^{(j)} + a^{(j)} t
\]

\[
\omega_v^{(j)} = \mu^{(j)} A^{(j)} e^{-h^{(j)} t} \sqrt{h^{(j)}^2 + k_2^{(j)} \gamma^2} \cos \left( k_2^{(j)} t + \beta^{(j)} + \alpha^{(j)} \right) + C_2^{(j)} + a^{(j)} t
\]

4. Conclusion
The short duration of gear shifting in modern automatic transmissions allows to consider this process as impact interaction. A transmission of a vehicle in a short time interval is like a closed mechanical system for which the theorem on the change in the angular momentum upon impact is applicable. The dependences of the velocities of transmission elements before and after switching are obtained with its help.

It was shown as a result of the transmission dynamic model structure description that it corresponds to a vibro-impact self-oscillating system in which the impact caused by the gear shift leads to oscillations in the rotation angle of the output shaft of the gearbox. Modes of movement of the system are determined by gearshift velocities. The functions of the angular position of the output shaft of the gearbox and the function of its velocity are continuous over the entire segment of movement. The function of the velocity of the input shaft of the gearbox are continuous on intervals between gear shifts and loses continuity and has finite jumps between intervals after gear shift.

Acknowledgments
The research was supported by Russian Science Foundation (Project No. 19-19-00065).

References
[1] Fischer R, Küçük F, Jürgens G, Najork R and Pollak B 2015 The Automotive Transmission Book (Cham: Springer) p 372
[2] Naunheimer H, Bertsche B, Ryborz J and Novak W 2011 Automotive Transmissions (Berlin: Springer) p 740
[3] Förster H J 1991 Automatische Fahrzeuggetriebe (Berlin: Springer) p 551
[4] Matthes B 2005 Dual Clutch Transmissions - Lessons Learned and Future Potential SAE Tech. P. 2005-01-1021
[5] Wheals J C, Turner A, Ramsay K, O’Neil A, Bennet J and Fang H 2007 Double Clutch Transmission (DCT) using Multiplexed Linear Actuation Technology and Dry Clutches for High Efficiency and Low Cost SAE Tech. P. 2007-01-1096
[6] Schreiber W and Becker V 1999 Doppelkupplungsgetriebe Patent DE 19821164
[7] Bai Sh, Maguire J and Peng H 2013 Dynamic analysis and control system design of automatic transmission (Warrendale: SAE International)
[8] Chudakov E A 1933 Calculation of the car. Power train (Moscow-Leningrad: Gosmashmetizdat) p 309
[9] Panovko Ya G 1991 Introduction to the Theory of Mechanical Vibrations: Textbook 3rd Ed. (Moscow: Nauka) p 256