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# Increasing the Quality of Transient Processes in the Vehicle Transmission

**A. Taratorkin[1], V. Derzhankii[1,2], I. Taratorkin[1]**

[1]Federal State Budgetary Scientific Institution "Institute of Engineering Science, Ural Branch of the Russian Academy of Sciences" (Ekaterinburg, Russian Federation),
[2]Kurgan State University (Kurgan, Russian Federation)



**Abstract.** The paper shows results of an experimental and theoretical study of dynamic processes in the vehicle transmission after kinematic alignment of elements during gearshifts. The purpose of the research is increasing the quality of transient processes in the vehicle transmission. Applying an analysis of experimental results obtained through looking into dynamics of a 3-ton vehicle transmission and studying literature sources it was established that dynamic loading of the transmission after the kinematic stage of shifting (i. e. synchronizing speeds of driving and driven elements in the gearbox) is influenced by oscillations which are in the single-node mode. Solving the task of increasing transient processes is achieved by applying a method of control power redistribution. By employing simulation models a number of methods were used to regulate power redistribution. Results of computations made it possible to determine that the efficiency of power redistribution are closely related to initial conditions of the process under the study. In the progress of the research a method for identifying the initial conditions was developed. This method is based determining signatures of the torque and its derivatives. In accordance with the research results it turned out that it is appropriate to apply the ZVD (zero vibration derivation) algorithm of power redistribution for low gears (below 4th) from point of view achieving better overshoot and robustness characteristics and a satisfactory response rate level. For higher gears it is recommended that the Ramp algorithm (linear increase in the control input) be used for the cases when the response rate is not longer than period of the single-node mode of oscillations occurring in the dynamic system during a gear shift. Application of the proposed algorithms allows to bring down dynamic loading of the transmission and also to improve the comfort in vehicles.

**Keywords:** vehicle, gearshift, control, process, power redistribution

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## Повышение качества переходных процессов в трансмиссии автомобиля

**А. Тараторкин[1], В. Держанский[1,2], И. Тараторкин[1]**

[1]Федеральное государственное бюджетное учреждение науки Институт машиноведения Уральского отделения Российской академии наук (Екатеринбург, Российская Федерация),
[2]Курганский государственный университет (Курган, Российская Федерация)

**Реферат.** В статье излагаются результаты экспериментального и теоретического исследования динамических процессов при переходных режимах в трансмиссии транспортного средства на стадии после кинематического выравнивания ведущих и ведомых элементов включаемой передачи. Целью исследования является повышение качества переходных процессов в трансмиссии автомобиля. На основе анализа экспериментальных данных, полученных при испытаниях автомобиля массой три тонны, установлено, что характер динамического нагружения трансмиссии после кинематической стадии переключения (т. е. после окончания синхронизации скоростей ведущих и ведомых элементов в коробке передач) соответствует низшей, одноузловой форме колебаний. Новизна результатов исследования заключается

**Адрес для переписки**
Тараторкин Александр
Федеральное государственное бюджетное учреждение науки Институт машиноведения
Уральского отделения Российской академии наук
ул. Комсомольская, 34,
620049, г. Екатеринбург, Российская Федерация
Тел.: +7 906 778-76-98
alexandr.taratorkin@mail.ru

**Address for correspondence**
Taratorkin Alexander
Federal State Budgetary Scientific Institution
"Institute of Engineering Science,
Ural Branch of the Russian Academy of Sciences"
34 Komsomolskaya str.,
620049, Ekaterinburg, Russian Federation
Tel.: +7 906 778-76-98
alexandr.taratorkin@mail.ru

в применении метода перераспределения силового управляющего воздействия при обосновании способа идентификации начальных условий по сигнатурам крутящего момента и его производных. По результатам расчетно-экспериментального исследования установлено, что для низших передач (ниже 4-й) целесообразно применять ZVD-алгоритм (zero vibration derivation) перераспределения управляющего воздействия. Он позволяет достичь лучших характеристик перерегулирования и робастности при удовлетворительном уровне быстродействия. Для более высоких передач рекомендуется использовать алгоритм линейного изменения (линейное увеличение управляющего воздействия) для случаев, когда время достижения «уставки» равно или больше периода одноузловой формы колебаний на включаемой передаче. Применение предложенных алгоритмов позволяет снизить динамическую нагрузку в трансмиссии, а также повысить уровень комфортабельности транспортного средства.

**Ключевые слова:** управление, процесс, переключение передач, транспортное средство, перераспределение управляющего воздействия

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## Introduction

Application of modern high-torque engines results in generating intensive vibrations in the transmissions of vehicles when they starts to move and when gears are shifted. It reduces the riding comfort. Low-frequency oscillations in the transmission are quite natural; they are generated when the driving torque in a dynamic system rises sharply at its lowest natural frequency, which is in fact the first single-node oscillation mode. This phenomenon made a full-scale manifestation as a 3-ton transportation vehicle fitted with a hybrid power plant, which contained a 650 hp eight-cylinder engine, was subjected to an experimental study. When the dynamic load of the vehicle was estimated, it was established that low-frequency torsional oscillations in the transmission initiate intense longitudinal vibrations of the vehicle's body with the acceleration ranging from 2 to 4 m/s$^2$.

## Analysis of low-frequency oscillations occurring in transient processes while a vehicle is beginning to move from rest and gear are shifted

Kinematic and power parameters were measured which describe the dynamics of the power plant and of the vehicle. Fig. 1 shows an oscillogram characterizing the dynamics of an engine-transmission system in the process when the vehicle accelerates. The figure also shows variations of the gearbox input shaft rotation speed, the vehicle velocity, and the longitudinal acceleration of the vehicle's body.

Fig. 1 shows that changing gears from the first gear to the fifth one results in initiating low-frequency oscillation in the transmission. Alongside with it, the torque amplitude reaches the values of 1.3–1.5 of the maximum engine torque. At the same time, the longitudinal accelerations of the vehicle's body are 2–4 m/s$^2$. It should be noted that the parameters of the transient process are considerably influenced by variations of the natural frequency of the dynamic system which is related to the engaged gear in the transmission. When the number of the engaged gear goes up, the natural frequency increases from 1.6 to 6.9 Hz (Tab. 1, line 4). Insignificant mismatching between calculated and experimentally derived values of the natural frequency (for example, from 2.1 to 2.5 Hz at the second gear) might be caused by a nonlinearity of the dynamic system, the changing weight of the vehicle depending on its load (number of passengers, amount of fuel, oscillations of the sprung mass), temperature, pressure in the tyres, etc. At the same time, changes of the natural frequency values depending on which gear is in are related to the values of the equivalent moment of inertia of the engine to the vehicle weight. Thus, the effect of varying natural frequencies should be taken into consideration when the algorithm for oscillation damping is to be developed, and it should be estimated by making use of the robustness parameter of the dynamic system.

It is worth mentioning that according to the results of numerical modeling and experimental data equalizing the angular velocities of sliding friction elements is also accompanied by torque variations at higher frequencies but with lower amplitudes (Fig. 1 – from 37.5 to 38.8 s, from 40.6 to 41.5 s, from 43.2 to 43.8 s). It is related to the ambiguity of the structural condition of the dynamic system during the slippage of friction elements when the equivalent moment of inertia is significantly lower. These oscillations do not influence as much the dynamic load and the comfort (longitudinal accelerations) in the vehicle due to their higher frequencies and insignificant amplitudes of the torque. This type of oscillation damping needs further research.

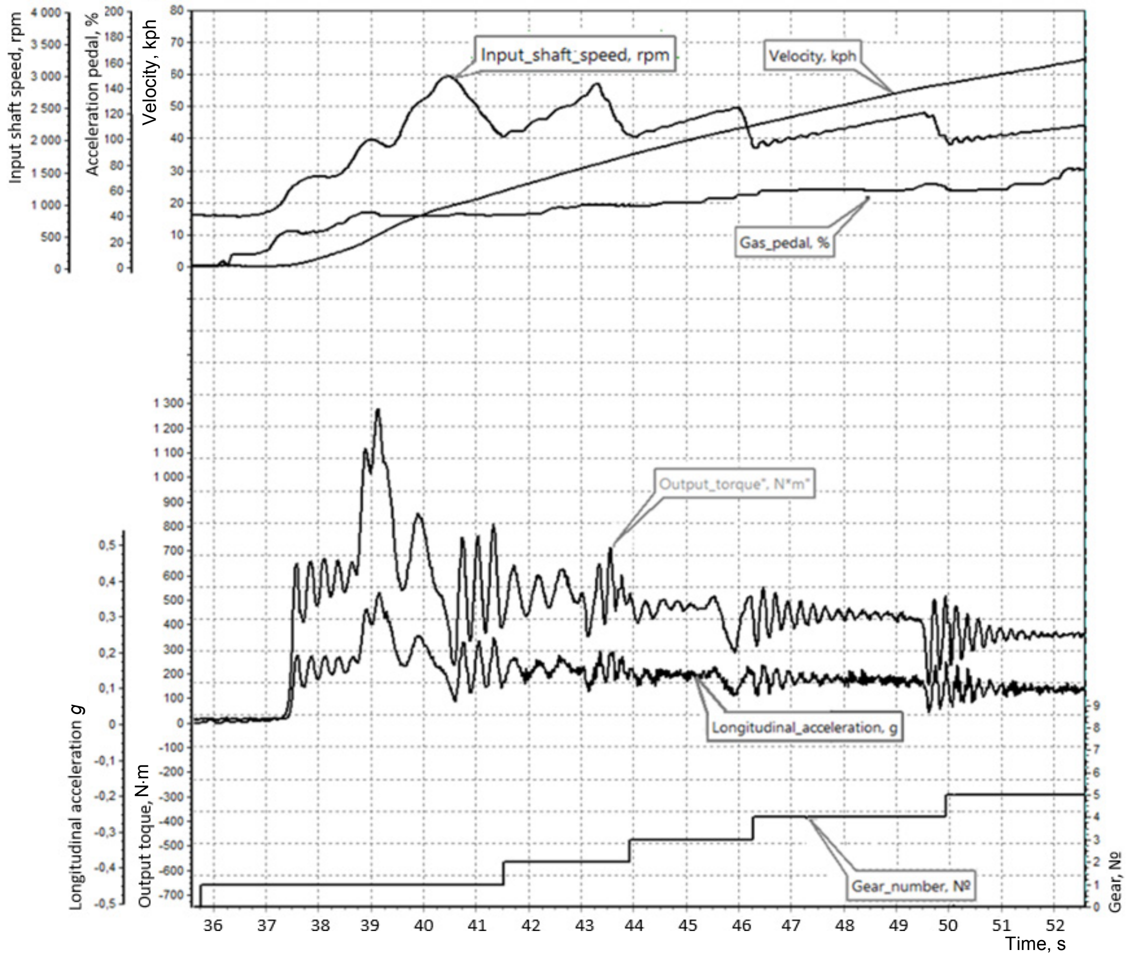



*Fig. 1.* Fragmented oscillogram illustrating the ways the movement parameters changed when the vehicle moved from rest and accelerated

*Table 1*

**Natural frequencies of the system**

| Gear | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| Gear ratio | 5.72 | 3.02 | 2.05 | 1.51 | 1.20 | 1.00 | 0.86 | 0.75 | 0.66 |
| Experimental frequency, Hz | 1.5–1.8 | 2.1–2.5 | 3.2–3.4 | 3.9–4.4 | 4.4–4.8 | 4.6–5.0 | 5.0–5.6 | 5.5–6.1 | 6.7–7.0 |
| Estimated frequency, Hz | 1.6 | 2.3 | 3.3 | 4.2 | 4.8 | 5.1 | 5.9 | 6.3 | 6.9 |

Referring to researching into the transient processes occurring at the beginning of the motion and when gears are shifted, the regulation action can be regarded as a unit function because at the time when a vehicle is accelerating after the angular velocities of control elements are equalized the required specific moving force is calculated applying the following formula

$$f_d = \frac{\dot{v}}{g\delta_j} + f_r,$$

where $f_d$, $f_r$ – specific driving force and tractive resistance respectively; $\dot{v}$ – required vehicle acceleration, $f_r = 0.02$; $g$ – gravitational acceleration ($g$ = 9.81 m/s$^2$); $\delta_j$ – rotary inertia coefficient with $j$-gear engaged.

Hence, when a vehicle accelerates in the first gear at an acceleration of 2.5 m/s$^2$, the relation is $\frac{f_d}{f} = 9.45$, in other terms, the required regulating action exceeds the rolling resistance torque by nearly ten times. The response of the system to this regulating action may be regarded as a response to a unit regulation function.

The most intensive generation of these oscillations can be observed when the fuel feed goes up quickly in the kick-down mode. This assertion is well defined in the graphs in Fig. 2.

When the vehicle is moving with the first gear in, the quick increase in fuel supply at the 78th and 82th seconds results in changing the torque at the transmission output shaft with an amplitude of up to 1760 N·m. This process goes together with longitudinal oscillations of the vehicle body at an amplitude of 4.5 m/s$^2$ its frequency being $f$ = 1.6 Hz: it corresponds to the first single-node mode. Thus, analyzing the experimental results it was discovered that dynamic and inertial loads in the required frequency range are created in correspondence with the single-node oscillation mode with the node in the axial shaft area.

This phenomenon is the Bonanza effect and is described in the book by Robert Fischer et all (see below). The energy of these oscillations is proportional to the squared torque divided by the double value of the angular rigidity of the system. As gear shifts occur (lower gears in particular) in the transmission of a transportation vehicle, the current torque is at its maximum while the equivalent angular rigidity is limited. It leads to an intensive accumulation of oscillation energy with low natural frequencies in the shifts at lower gears. The energy and amplitude of these oscillations can be reduced by varying the parameters of the system that would increase the natural frequency. Anyhow, it can hardly be achieved in real structures. The closest approach to it, regarding its technical essence and achievable results, is the method of damping oscillations detailed in [1] (Fig. 2.28, p. 78). This method implements damping of low-frequency oscillations in transportation vehicle transmissions at the stage following kinematic equalizing of velocities of driving and driven components by creating an antiphase regulating action at the natural frequency of a dynamic system. It also implies estimating the quality of transient processes, i. e. overshoot and duration based on modeling the dynamics of the system in which a unit regulating action is applied (quick application of the engine torque). In other words, the paper mentioned above proposes to alter the function of the regulating signal (in this particular case the regulating signal of the torque): to make it ramp-like (RAMP – a gradual increase of the engine torque in terms of time) or step-like, correlating the duration of the stages of the regulating function with the period (frequency) of the natural oscillation of a dynamic system.

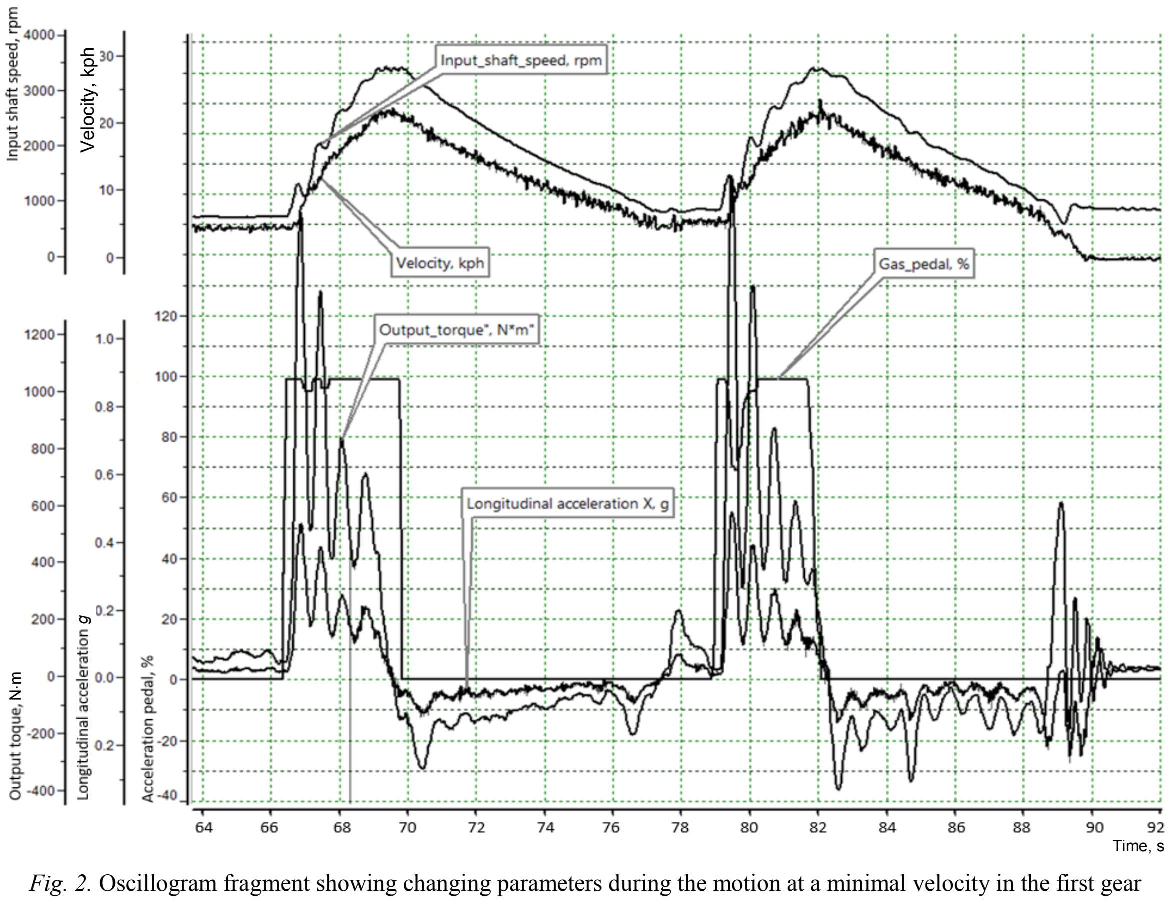



*Fig. 2.* Oscillogram fragment showing changing parameters during the motion at a minimal velocity in the first gear when the motion was periodically accelerated by quick shifts of the gas pedal from 0 to 100 %

This engineering solution, based on the idea of oscillation damping by means of generating regulating pulses that are antiphase to the oscillations, i. e. by means of redistributing the regulating input in terms of time and relating it to dynamic properties of the system, is implemented by the method which is known as Input Shapers [2–6]. This method is widely used in various commercial applications, and it has resulted in improving performances of various units: satellite systems [7], disk drives [8], cranes [9–11], coordinate measuring machines [12, 13], remotely piloted planes [14], milling machines, etc.

At the same time, if to speak about transport vehicles, these regulating algorithms have not been widely applied. To evaluate the possibility of utilizing this well-formalized method in vehicles for the purpose of improving the transient processing of shifting at the stage following the kinematic equalizing of angular velocities the researchers [15] performed a comparative estimation of how efficient different algorithms of torque redistribution are. To do it a two-mass mathematical model is used. In addition to it, these papers present a list of indicators that show the efficiency of various algorithms for torque redistribution and their applicability related to the criterion of optimizing gear shifts: acceleration intensity of vehicles, comfort when in motion, minimizing dynamic loads, etc. These indicators are in fact overshooting, rapidity of action and robustness.

**Estimating the effectiveness of the proposed method by applying the extended multi-mass model of the dynamic system**

To estimate the efficiency of input regulators in application to the research subject a more complicated simulation model was developed; it is presented below. In this model components are grouped in two major unities.

The first one is shown in Fig. 3; it describes how the following transmission elements interact: the engine, the double-clutch, the gears and shafts in the gearbox (even and odd branches, the cardan shaft and the axle shafts. It also contains a separate component simulating the vehicle body.

In the mathematical model the internal combustion engine is modeled as a 3*D* dependence of the torque on the rpm of the engine crankshaft and the fuel feed percentagewise (Fig. 4). This dependence allows modeling the engine in the traction mode employing external and partial characteristics and in the braking mode.

The second unit of the elements describes the operation of the gearbox controller and that of the engine controller. The control logic of the gearbox is implemented by two actuating mechanisms; they actuate the gearbox control unit and the doubled clutch friction regulating unit. The gearbox control unit makes use of the current values of the mechanical model of the gearbox (the gear shifted in, the selector position, the current velocity of the vehicle) to generate a request for a gearshift as per the gearshift pattern.

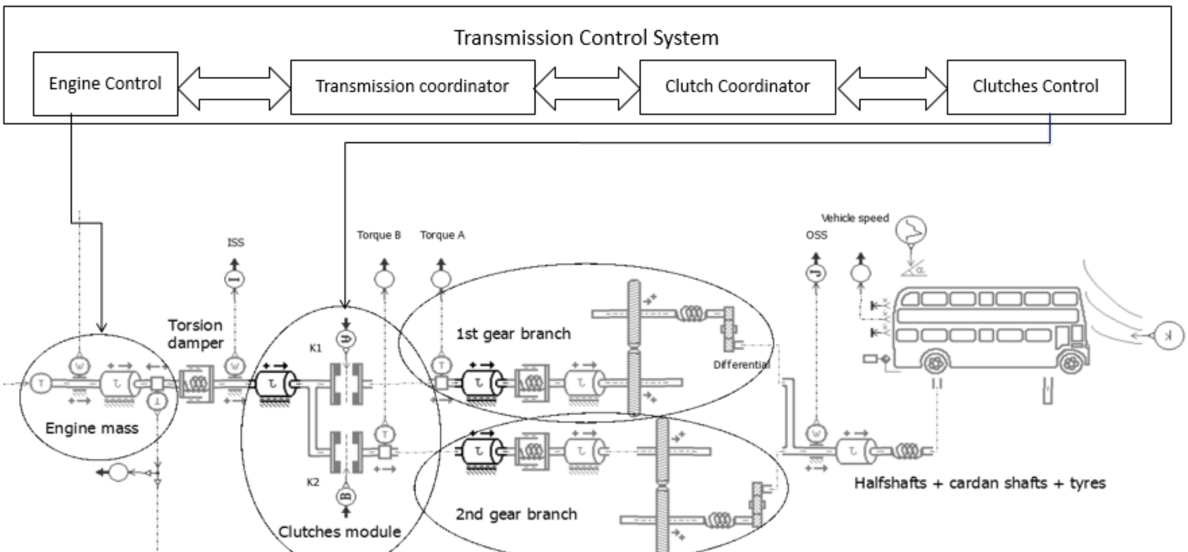



*Fig. 3.* Model elements describing mechanical components of the transmission

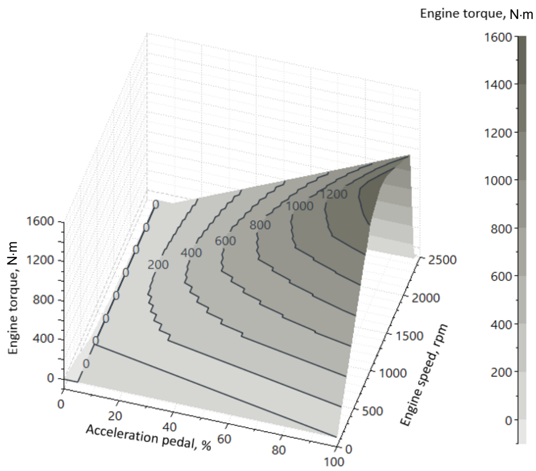



*Fig. 4.* Dependence of engine torque on crankshaft speed and fuel supply

The doubled clutch regulator defines the required condition of the friction elements for different stages of gear shifts. Five conditions of the friction elements are implemented: engaged (open), a linear increase of the friction torque at the stage of passing the rotational toque from the friction clutch of the gear being disengaged to the friction clutch of the gear being engaged (torque ramp up), a linear increase of the friction torque at the stage when there is a kinematic change in the gear ratio in the gearbox (torque ramp up), fully engaged (close), a linear decrease of the friction torque at the stage of passing the rotational toque from the friction clutch of the gear being disengaged to the friction clutch of the gear being engaged (ramp down).

Fig. 5 illustrates the way the rotational torque changes in the friction elements when gears are being shifted. The employed logic allows performing a gearshift by dividing the shift process into a force phase and a kinematic one. The logic of the engine control covers the switching of the modes of its operation comprising three conditions of the fuel feed control: the driver's mode, electronic maintenance of the torque, transferring the fuel feed control from the electronic unit to the driver. When the engine is controlled by the driver, the fuel is fed depending on the position of the accelerator pedal. The mode of maintaining the torque is actuated at the time when gears are being shifted; it time when specific conditions for a gear shift are needed. With the existing systems in operation the driver hardly feels when a short-time switchover of the control mode from the driver to the electronic unit takes place. At the same time, it helps to economically use the computational resources of the electronic gearbox control unit. The signal for switching modes is generated by the gearbox control unit when the torque is switched and when the kinematic equalizing occurs.

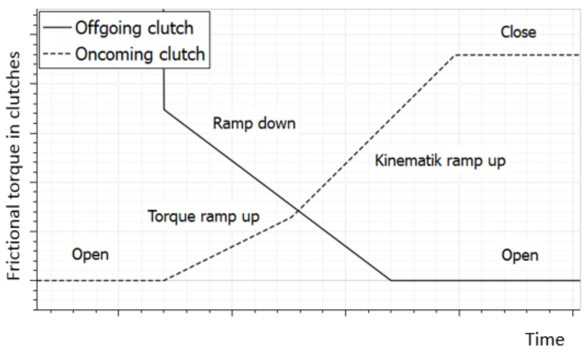



*Fig. 5.* Illustration of torque change in friction elements

To transfer the controls of the engine from the electronic unit back to the driver, there exists a transit mode which operates according to a certain law. This control law can be applied when a gearshift is completed or when the driver quickly changes the position of the acceleration pedal and when there is a substantial longitudinal acceleration of the vehicle body accompanied by oscillations of the vehicle body. To put this law into practice a ZV-shaper, a ZVD-shaper and a linear ramp of various duration equal to period fractions ($T$/2, $T$, 1,5×$T$) can be applied as is shown in Fig. 6.

A full-scale model of this system is implemented in the LMS Imagine. LAB Amesim. Software. Fig. 7 illustrates the results of modeling applying the Tip-in and Tip-out modes and utilizing the ZV-shaper and the ZVD-shaper. As is shown in the chart, the gearbox control unit makes use of the shapers if the driver has to quickly change the position of the acceleration pedal. Together with it, an electronic pedal mode is actuated; this mode follows the algorithm of redistributing power of the control torque depending on which shaper is in operation. Analyzing the results proves that the redistribution of the gearshift torque power is efficient (see the graphs in the upper part of the Fig. 7).

The graphs in the lower part of Fig. 7 show how efficient the results are. From analyzing the results of the modeling it can be derived that the ZVD-shaper manifests the highest efficiency in setting limits for the redistribution at acceptable values of response time and robustness. In general, the analysis proves that the redistribution of torque power during gearshifts is efficient.

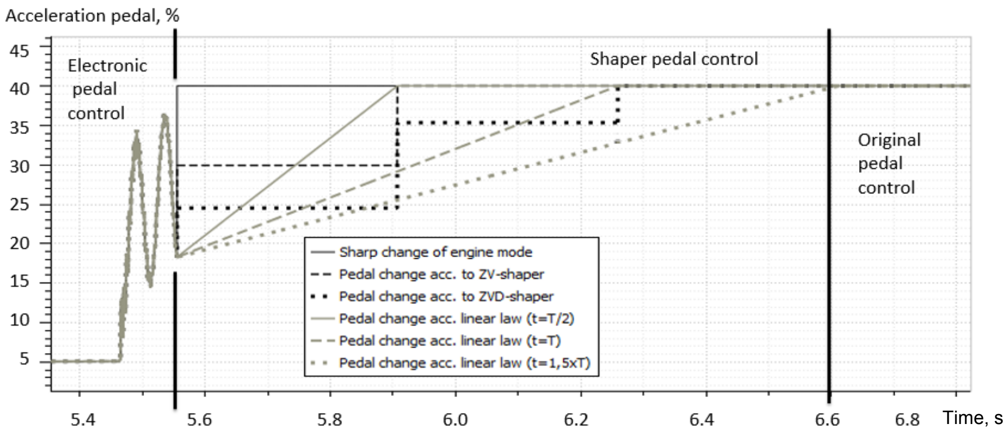



*Fig. 6.* Various control laws to change the electronic engine control mode for the driver engine control mode

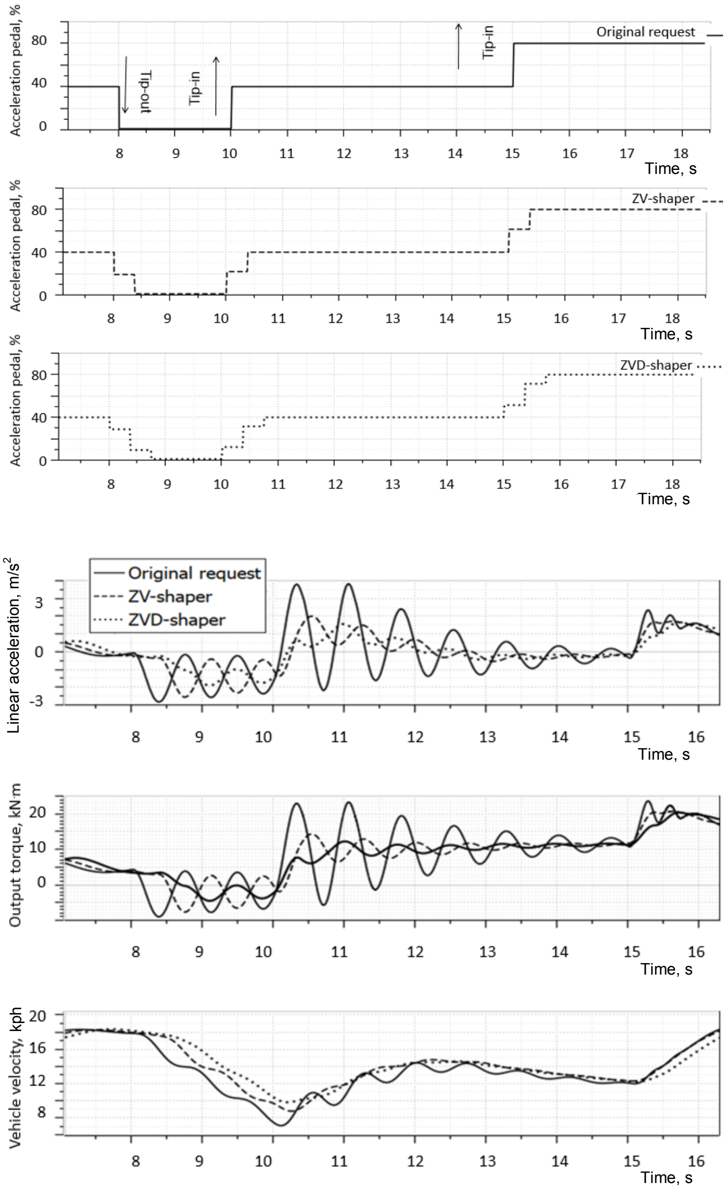
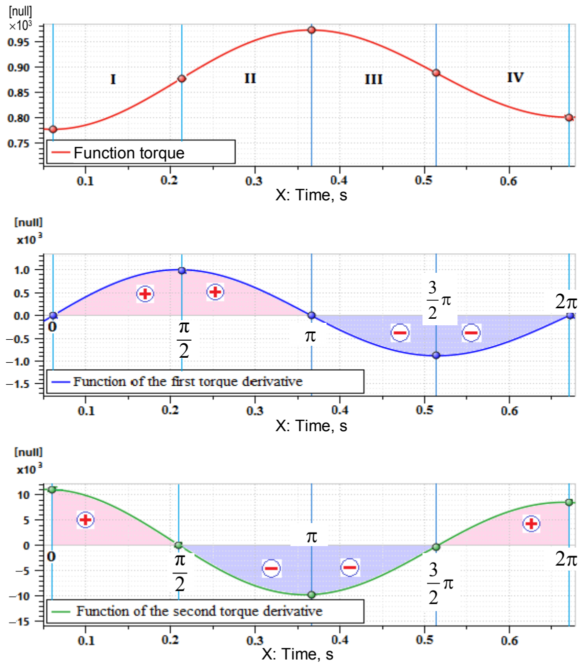



*Fig. 7.* Stability of dynamic gear shifting in a vehicle transmission by applying an input shaper

Somehow, the modeling of an extended multi-mass model of the dynamic system also showed that applying the input shapers as is to be done in compliance with the method detailed by R. Fischer is efficient only in the initial zero conditions. In real systems, which maintain control over gear-shifts in transportation vehicles, the initial conditions occurring after equalizing the speeds of the driving and driven elements are not zero conditions; they evolve at random being related to a combination of factors determined by the engine toques, tractive resistance together with oscillations of the dynamic system.

To further improve the efficiency of synthesized shapers an algorithm was developed within the framework of this research to identify the initial conditions to make use of the value of the engine torque and its derivatives as a reference. The essence of the method that is being proposed is as follows: employing the signatures of the first and second derivatives of the torque it is to be found in which of the quarter of the period of the natural oscillation (quartile) the oscillatory motion of the dynamic system takes place (see table in Fig. 8). For example, if the process occurs in the first quartile, the signature of the first derivative is positive and that of the second derivative is negative. If the process occurs in the second quartile, the sign of the first and of the second derivatives are negative. And if the process takes place, the signs are positive. Data for the correlation are given in Tab. 1 (lines 1–4). Meanwhile, the time, when the shaper is actuated, is also established by the meanings of the signatures (changes of their signs) of the corresponding derivatives of the torque (in Tab. 1, lines 5–7). For instance, when the process is in the first quartile, the time to actuate the shaper $\left(\frac{\pi}{2}\right)$ is determined by the zero value of the first derivative (the derivative's sign changes from positive to negative); in the second quartile the actuation time for the shaper $(\pi)$ is defined by the zero value of the second derivative (its sign changes from minus to plus); in the third quarter of the period the actuation time for the shaper $\left(\frac{3\pi}{2}\right)$ is determined by the zero value of the first derivative (its sign changes from minus to plus); and in the fourth quartile period the actuation time for the shaper $(2\pi)$ is determined by the zero value of the second derivative (its sign changes from plus to minus).

After determining the shaper actuation time the control system that being proposed in this paper operates following the developed algorithm. In addition to it, the control program provides for finding the natural frequency of the system as a function of the number of the gear which is engaged and an arrangement for actuating driving axles.




Determining the torgue signatures, derivatives and shaper actuation time

| Estimated parameters | | $0\leftrightarrow\frac{\pi}{2}$ | $\frac{\pi}{2}\leftrightarrow\pi$ | $\pi\leftrightarrow\frac{3\pi}{2}$ | $\frac{3\pi}{2}\leftrightarrow 2\pi$ |
|---|---|---|---|---|---|
| Signature sign | $M$ | $\pm$ | $\pm$ | $\pm$ | $\pm$ |
| | $\dot{M}$ | + | – | – | + |
| | $\ddot{M}$ | – | – | + | + |
| Shaper actuation time | | $\frac{\pi}{2}$ | $\pi$ | $\frac{3\pi}{2}$ | $2\pi$ |
| Direction of the sign change | $\dot{M}$ | $+\rightarrow-$ | | $-\rightarrow+$ | |
| | $\ddot{M}$ | | $-\rightarrow+$ | | $+\rightarrow-$ |

*Fig. 8.* Identification chart for the initial phase of the torque oscillations; the torque applies load to the transmission after the stage for kinematic equalizing of the driving and driven transmission components is completed

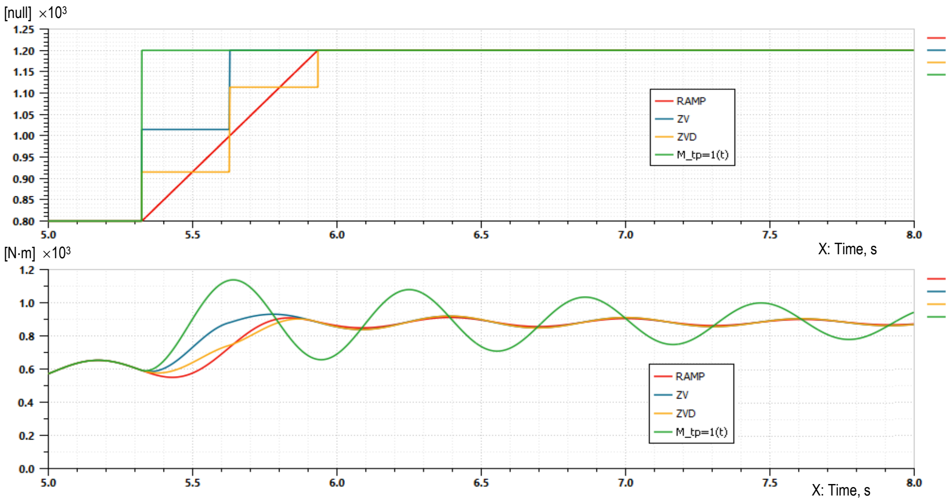
Also, the relative angular speed of the driving and driven elements and this allows identifying the completion of the kinematic equalizing stager. After the kinematic equalizing $\Delta\omega = 0$ is done, the measuring control system (MCS) of the vehicle generated an inquiry for the required engine torque $M_{tp}$, a computational torque is determined as a response to the unit regulation action. Then, a procedure is done to compare the overshoot $\sigma$ and the acceptable value. If the overshoot exceeds the acceptable value, a calculation procedure is performed to determine the parameters defining the power redistribution of the control signals $A_i$ and $t_i$ (amplitudes and their duration) for the shaper type (shaper-filter) that would produce the required value of the priority parameter which is to provide for the quality of the transient process (overshoot, rapidity of action and robustness). To find the function of the torque (measured torque) corresponding to the first single-node oscillation mode filtering is done by applying a rejection filter at the natural frequency $\omega_c$ of the system with the pass band $\pm(\Delta\%)\omega_c$. In this expression $\Delta\%$ allows for a possibility of varying the natural frequency of the system and determines the priority of the selected shaper regarding the degree of its robustness. For example, the results of the modeling show that for the first gear of a 3 t vehicle, for the ZV-shaper with the overshoot set to 10 % the pass band is $\Delta\% \approx \mp 5$, for the ZVD-shaper with the overshoot set to 10 % the pass band is $\Delta\% \approx (\pm 20)$ %, and for the for the ZV-shaper with the overshoot set to 10 % the pass band is $\Delta\% \approx \mp 5$; for the ZVD-shaper with the overshoot set to 10 % the pass band is ZV-shaper with the overshoot set to 10 % the pass band is $\Delta\% \approx \mp 5$; for the ZVDD-shaper with the overshoot set to 10 % the pass band is $\Delta\% \approx (\pm 30)$ %.

Further on, making use of the signatures of the obtained (filtered) torque, its first and second derivatives the number of the current quartile of the oscillation process and its time boundaries are found. The right boundary of the interval helps to determine the time for the control action of the synthesized shaper to be applied. After it, the parameters of the synthesized shaper are transmitted into the system controlling the fuel feed rate. At this stage the operation of the algorithm is completed.

Fig. 9 presents the results of modeling the dynamics of the system under the control of the three shapers.

The results of the modeling illustrate the efficiency of the synthesized shapers for damping low-frequency oscillations. The graphs in Fig. 9 prove that the ZVD algorithm brings about the highest efficiency in restricting the overshoot under acceptable values of rapidity and robustness. On the whole, the analysis of the results proves that the redistribution of the control torque at gearshifts is efficient.

At the same time, from the results of the modeling the dynamics of the power plant installed in the object under research it can be inferred that in the case of the first three gears it is appropriate to employ the ZVD-shaper. If a higher rapidity action is needed, usage of the ZV-shaper is possible. Regarding the engagement the fourth and higher gears, it is appropriate to utilize the RAMP algorithm. If it is so, the time of the control action ramp should match the natural frequency period of the selected gear. Application of the control action within the time which is shorter than the period duration leads to poorer indicators of overshoot and robustness.




Estimated efficiency of the algorithms under consideration

| Shaper type | Rapidity of action | Overshoot | Robustness |
|---|---|---|---|
| ZV-shaper | 1 | 3 | 3 |
| ZVD-shaper | 2–3 | 1 | 1 |
| RAMP | 2–3 | 2 | 2 |

*Fig. 9.* Results of modeling the dynamics of the system with different control versions to illustrate the efficiency of the synthesized shapers designed to damp low-frequency oscillations

Application of the control action within the time exceeding the period duration of natural oscillations results in declined acceleration dynamics, and it is unacceptable.

## CONCLUSIONS

1. Hence, in the process of performing the present research a method was devised for damping low-frequency oscillations in transmissions of transportation vehicles at the stage following kinematic equalization of speeds of driving and driven components; to do so an antiphase control action was developed that would operate at the natural frequency of dynamic systems which is in the lowest single-node oscillation mode.

2. The novelty lies in synthesizing a shaper on the basis of computing parameters for redistributing in time the control action under the following algorithms: ZV (Zero Vibration), ZVD (Zero Vibration and Derivative), ZVDD (Zero Vibration and Double Derivative), RAMP, etc. The amplitude and the duration of the stages as well as the stage when an algorithm begins to function are derived from the initial conditions of the oscillation process in dynamic systems; these conditions are identified with the help of the signatures of the first and second derivatives of the torque. The type of a regulator (shaper filter) is selected depending on the choice of the priority parameter that characterizes the quality of the transient process – overshoot, rapidity of action or robustness.

3. It was established that to damp low-frequency oscillations related to the gear shifting in the transmission of a transport vehicle the ZVD-algorithm yields the highest efficiency in restricting overshooting at acceptable values of the rapidity of action and robustness. When a more rapid action is required, the ZV-shaper can be used. It is also adequate to apply the RAMP-algorithm if the value of the natural frequency of the lowest single-node oscillation mode is equal to 4 Hz or higher.

4. On the whole, the analysis of the achieved results proves that the redistribution of the power control action at the stage after kinematic equalizing in the process of gear shifting in transmissions of transport vehicles is highly efficient.

5. The efficiency of the proposed method resides in damping oscillations in transmissions of transport vehicles when gear shifting is in progress following the stage of kinematic equalizing. The proposed method brings about better service properties such as the acceleration rate, comfort, etc.

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