Increase ride wheeled vehicles due to the continuous damping control in the suspension system

M Zhileykin¹ ³ and I Fedotov²

1 Bauman Moscow State Technical University
2 FSUE «NAMI» State Research Center of the Russian Federation (2 Avtomotornaya St., Moscow, Russia, 125438),

³E-mail: jileykin_m@mail.ru

Annotation. One of the promising ways to improve the wheeled vehicles smooth running is the development of dynamic semi-active springing systems, as well as the laws of such systems control. Dynamic control systems in this paper are understood as systems operating at a real-time rate with the current (instantaneous) phase coordinates values. The aim of the work is to develop the laws of optimal control of the wheeled vehicles suspension, providing an increase in the driver and passengers comfort in a wide range of operating conditions. The of the optimal control system algorithm of the level of damping shock absorber suspension wheel machine to improve of the driver and passengers comfort. The proposed algorithm efficiency is proved by simulation methods. It is established that at rectilinear movement on different roads with different speeds, the level of the movement comfort is increased by 8 to 41% when the frequency controlling effect of 15 Hz and 13...59% at the frequency control action of 50 Hz compared to uncontrolled suspension.

Introduction
Creation of effective means of protection against external dynamic influences has always been one of the important problems of modern technology. Particularly acute issues of ensuring acceptable levels of vibration are raised in the creation of modern vehicles: aircraft, wheeled vehicles (WV), ships. With increasing speeds of these vehicles, the intensity of dynamic impacts usually increases, so the development of vehicles is accompanied by a continuous increase in the requirements for vibration protection systems.

Traction and speed properties of the car have a major impact on the formation of maximum and average speeds. Other operational properties, in particular, smooth running, act as factors limiting the further increase in operating speeds [1]. Currently, the design solutions of components and assemblies of the car, affecting the indicators of stability, handling, braking properties, turnability, patency have reached a high degree of perfection. However, despite these successes, the system of the primary cushioning inhibits further growth speeds.

One of the promising ways to improve the smoothness of the WV is the development of dynamic semi-active springing systems, as well as the laws of such systems control [2–20]. Dynamic control systems in this paper are understood as systems operating at a real-time rate with the current (instantaneous) values of phase coordinates. The aim of the work is to develop the laws of optimal control WV suspension, providing increased comfort driver and passengers in a wide range of operating conditions.

Calculation of control actions to improve the stroke smoothness
Consider a motion linearized model of the vehicle body, with respect to only three phase coordinates (fig. 1):

- relative to the vertical $Z$ axis of the associated coordinate system;
- rotation of the body relative to the main axis of inertia $X$ related coordinate system;
- rotation of the body relative to the main axis of inertia $Y$ of the associated coordinate system.

![Design scheme for multiple-seated oscillatory system WV](image)

**Fig. 1.** A design scheme for multiple-seated oscillatory system WV:

$C$ — vehicle body center of gravity; $X, Y, Z$ — the coordinate system associated with the vehicle body center of gravity; $\phi, \psi$ — angles of longitudinal and transverse inclination of the body, respectively; $P_i$ — force in $i$-th suspension; $m_i$ — mass $i$-th wheel; $n$ — number of axis; $Q_i$ — force in $i$-th tyre; $q_i$ — unevenness height of the support base under the $i$-th wheel.

Let's consider the body is absolutely solid, suspension wheels made by "McPherson" scheme.

As a result, we obtain the following equations of state for the body of WV:

\[
\begin{align*}
\ddot{z} & = -U_1 = \frac{1}{M} \sum_{i=1}^{2n} \Delta P_{li} \\
\phi & = -U_2 = \frac{1}{J_x} \sum_{i=1}^{2n} \Delta P_{li} \\
\psi & = -U_3 = \frac{1}{J_y} \sum_{i=1}^{2n} \Delta P_{li}
\end{align*}
\]

(1)

where $\ddot{\phi}$ — body angular acceleration relative to the axis $X$; $\ddot{\psi}$ — body angular acceleration relative to the axis $Y$; $\ddot{z}$ — vertical acceleration of the vehicle body center of gravity; $J_x, J_y$ — sprung mass moments of inertia WV relative an axis $X$ and $Y$ respectively; $M$ — the wheel vehicle sprung weight; $l_i, B_i$ — longitudinal and transverse coordinates (respectively) of fastening $i$-th suspension to the WV body in the coordinate system associated with the machine body; $\Delta P_{li}$ — $i$-th control action (force change in $i$-th the damping suspension member); $n$ — wheeled vehicle number of axles; $U_1$ — controlling the main vector of forces created by the suspension damping elements in the vertical direction; $U_2$ — control torque generated by the suspension damping elements relative to the axis $Y$. 
(applied to the vehicle body center of gravity); $U_3$— control torque generated by the suspension damping elements relative to the axis $X$ (applied to the vehicle body center of gravity).

The initial system of equations of state (1) is a linear system described by a vector differential equation

$$\frac{dX(t)}{dt} = A(t)X(t) + R(t)U(t),$$

(2)

where $X(t)$— state vector; $A(t)$, $R(t)$— matrix functions of time according to orders $2n \times 2n$; $2n \times 3$; $U(t)$— $2n$- dimensional control vector.

Note that in the system of equations (2) all elements $a_{ij}$ matrix $A(t)$ equal to zero.

Vector control $U(t) = [U_1(t), U_2(t), U_3(t)]^T$ for obvious physical reasons is bounded, i.e. belonging to a closed set

$$\|U_i(t)\| \leq U_{\text{max}}, i = 1, 2, 3$$

Control forces:

$$\Delta P_i = P_{\text{id}} - P_{\text{ir}}$$

where $P_{\text{id}}$— actual current force value in the suspension damping element; $P_{\text{ir}}$—the required force value in the damping element.

To determine values $\Delta P_i$ we formulate the criterion of optimality, on the basis of which we will determine the required variables values. The most appropriate is to obtain such values $P_{\text{ir}}$, which would be the least different from the actual current forces values in the damping elements $P_{\text{id}}$, that is, it is necessary to strive to ensure that

$$\Delta P_i = P_{\text{id}} - P_{\text{ir}} \rightarrow 0, i = 1, 2, ..., 2n$$

We formulate the quadratic optimality criterion as follows:

$$f(\Delta P) = \sum_{i=1}^{2N}(P_{\text{ir}} - P_{\text{id}})^2 = 0$$

(3)

The resulting problem is an optimization problem in the presence of constraints in the form of equality. Let's introduce auxiliary functions

$$g_1 = U_1 + \frac{1}{M} \sum_{i=1}^{2n} \Delta P_i = 0$$

$$g_2 = U_2 + \frac{1}{J_y} \sum_{i=1}^{2n} \Delta P_i = 0$$

$$g_3 = U_3 + \frac{1}{J_x} \sum_{i=1}^{2n} \Delta P_i = 0$$

The necessary minimum conditions of the function (3) are written as follows [9]:

$$\left\{ \begin{array}{l}
d\frac{df}{d\Delta P_i} + \sum_{i=1}^{3} \lambda_i \frac{\partial g_i}{\partial \Delta P} = 0 \quad j = 1, 2, ..., 2n \\
g_i(\Delta P) = 0 \quad i = 1, 2, 3
\end{array} \right.$$  

(4)

where: $\lambda$— indeterminate Lagrange multipliers.

In our case, equations (4) will take the following form
System (5) is a linear system containing $2n+3$ algebraic equations with respect to $2n+3$ unknown $\Delta P_i, \lambda_1, \lambda_2, \lambda_3$, from which are easily determined $\Delta P_i$.

After calculating the control actions $\Delta P_i$ you must convert these values to damping factors $k_i$.

**Limitations on control action**

The regulator output must have limited amplitude for at least two reasons. First, due to the fact that in case of exceeding the maximum pressure value, the safety valve is triggered. Thus, the upper limit of permissible control actions in the equations system (5) is always determined. There is also a lower limit of control actions associated with the presence of a viscous friction minimum value in the damping device, as well as the presence of "dry" friction in the seals. The upper and lower bounds form the area of acceptable control values. We assume for certainty that the upper and lower bounds of the permissible values of forces $P_i$, developed shock absorber, differ from the nominal level twice.

**Operation algorithm of the continuous optimal control system of springing system damping**

1. For the current time $t_j$ according to the vertical acceleration sensor installed in the vehicle body center of gravity of the body, determine the value of $\ddot{z}_j$ and differentiating the readings of the angular velocity sensors of the housing to determine the values $\dot{\phi}_j$ and $\dot{\psi}_j$.

2. For the current time $t_j$ sensor readings differentiation of suspensions deflections to determine the speed of deflection of $\dot{h}_{ij}, i=1,2,\ldots,2n$.

3. For the current time $t_j$ to determine the current power $P_{ijT}, i=1,2,\ldots,2n$ in each shock absorber based on the readings of pressure sensors installed in their upper and lower cavities:

$$P_{ijT} = p_{up\_ij}S_{up\_i} - p_{low\_ij}S_{low\_i}, i=1,2,\ldots,2n,$$
where \( p_{up,ij} \), \( p_{low,ij} \) — pressure values in the upper and lower cavities \( i \)-th shock absorber, respectively; 
\( S_{up,i} \), \( S_{low,i} \) — shock absorber piston area in the upper and lower cavities \( i \)-th shock absorber, respectively.

4. For the current time \( t_j \) solve the system (5) and determine the required force value 
\( P_{imp} = P_{ijT} + \Delta P_j, i = 1, 2, \ldots, 2n \) in each shock absorber.

5. For the current time \( t_j \) according to pre-programmed in the memory of the on-Board computer load characteristics of the shock absorber on the basis of the values obtained in paragraph 1 \( \dot{h}_j \) to define a nominal current strength in the shock 
\( P_{ij0}, i = 1, 2, \ldots, 2n \) (curve 1 on fig. 2).

6. For the current time \( t_j \) define the upper 
\( P_{ij0}, i = 1, 2, \ldots, 2n \) (curve 2 on fig. 2) and lower 
\( P_{ij max} = 0.5 P_{ij0}, i = 1, 2, \ldots, 2n \) (curve 3 on fig. 2) the limits of the shock absorber force variation to the obtained values \( \dot{h}_j \).

7. For the current time \( t_j \) determine the values of scaling factors \( \Delta k_{ij}, i = 1, 2, \ldots, 2n \), which will allow to proportionally change the values of the forces \( P_{imp} \), to done the inequality 
\( P_{ij max} \leq P_{imp} \leq P_{ij max}, i = 1, 2, \ldots, 2n. \)

7.1. if \( P_{ij max} > 0 \) and \( P_{imp} > P_{ij max} \), then 
\( \Delta k_{ij} = \frac{P_{ij max}}{P_{imp}}, i = 1, 2, \ldots, 2n, \)

7.2. if \( P_{ij max} < 0 \) and \( P_{imp} < P_{ij max} \), then 
\( \Delta k_{ij} = \frac{P_{ij max}}{P_{imp}}, i = 1, 2, \ldots, 2n, \)

7.3. if \( P_{ij max} > 0 \) and \( P_{imp} < P_{ij max} \), then 
\( \Delta k_{ij} = \frac{P_{imp}}{P_{ij max}}, i = 1, 2, \ldots, 2n, \)

7.4. if \( P_{imp} < 0 \) and \( P_{imp} > P_{ij max} \), then 
\( \Delta k_{ij} = \frac{P_{ij max}}{P_{imp}}, i = 1, 2, \ldots, 2n. \)

8. For the current time \( t_j \) of all \( \Delta k_{ij} \) to choose the minimum value 
\( \Delta k_{j\min} = \min[\Delta k_{ij}, i = 1, 2, \ldots, 2n]. \)

9. For the current time \( t_j \) adjust the values of the required force 
\( P_{imp} = P_{ijT} \Delta k_{j\min}, i = 1, 2, \ldots, 2n. \)

10. For the current time \( t_j \) calculate damping coefficients for each shock absorber \( k_{ij} \)

10.1. if \( \text{sign}(\dot{h}_j, \Delta P_j) > 0 \), then 
\( k_{ij} = \min \left[ \frac{P_{ij max}}{P_{ij max}} \right], i = 1, 2, \ldots, 2n, \)

10.2. if \( \text{sign}(\dot{h}_j, \Delta P_j) < 0 \), then 
\( k_{ij} = \max \left[ \frac{P_{ij max}}{P_{ij max}} \right], i = 1, 2, \ldots, 2n, \)

10.3. if \( \text{sign}(\dot{h}_j, \Delta P_j) = 0 \), then 
\( k_{ij} = 1, i = 1, 2, \ldots, 2n. \)

11. For the current time \( t_j \) to normalize the values of the coefficients \( k_{ij} \) for the range \([0.5–1.0]\)
\( k_{j max} = \max[k_{ij}], i = 1, 2, \ldots, 2n, \)
\( k_{ij} = 0.5 + \frac{k_{j max}}{2k_{ij}}, i = 1, 2, \ldots, 2n. \)
12. Further, the steps 1 — 11 again.

**Efficiency investigation and algorithm efficiency of continuous optimal control system damping springing system**

The controlled suspension efficiency according to the above algorithm was tested by simulation methods in comparison with the unmanaged suspension, elastic and damping characteristics of which had a nominal damping and static deflection. The main technical characteristics of the two-axle wheeled machine are given in Table 1.

**Table 1. Specifications two-axle wheeled machine**

| Parameter                                           | Value       |
|-----------------------------------------------------|-------------|
| Number of axis                                      | 2           |
| Vehicle body mass, kg                               | 2400        |
| Wheel base, m                                       | 3,3         |
| Wheel track, m                                      | 1,7         |
| Wheel weight, kg                                     | 80          |
| Free wheel radius, m                                | 0,4         |
| The maximum suspension deflection, relative to the wheel, m | 0,24        |
| The vehicle body inertia of the relative to the longitudinal axis passing through the center of gravity, kg×m² | 1450        |
| The vehicle body inertia of the relative to the lateral axis passing through the center of gravity, kg×m² | 8000        |
| Distance from body center of gravity to first axis, m | 1,56        |
| Distance from body center of gravity to second, m    | 1,74        |
| Springingsystem type                                | independent suspension of all wheels |
| Elastic suspension element type                     | pneumatic elastic element |
| Damping suspension member type                      | controlled twin-tube shock absorber |

**Table 2. Study results of the suspension control system effectiveness of a two-axle wheeled vehicle**

| Type of road surface                      | Movement speed, kph | Frequency regulation, Hz | The value of the criterion "Comfort", m/s² |
|------------------------------------------|---------------------|--------------------------|------------------------------------------|
|                                          | controlled suspension | uncontrolled suspension  |
| Road with asphalt concrete pavement      | 35, 15              | 3,78                     | 4,55                                     |
|                                          | 35, 50              | 3,71                     |                                          |
|                                          | 80, 15              | 7,73                     | 8,89                                     |
|                                          | 80, 50              | 8,19                     |                                          |
|                                          | 20, 15              | 6,14                     |                                          |
| Dirt road of satisfactory quality        | 20, 50              | 7,55                     | 8,25                                     |
|                                          | 35, 15              | 24,90                    | 60,66                                    |
|                                          | 35, 50              | 35,60                    |                                          |
"Comfort" criterion was used as a criterion of the wheel suspension control system efficiency [21]. The analysis of this criterion was carried out in the frequency range of 0.7–22.4 Hz.

The simulation of wheeled vehicle rectilinear motion with different speeds on three types of road surface was carried out:

1. road with asphalt concrete pavement;
2. dirt road of satisfactory quality.

To assess the control frequency influence on the control efficiency, the continuous damping control system simulation was carried out for two frequency values: 15 Hz and 50 Hz. The simulation results are presented in table. 2.

Analysis of the results given in table. 2 shows that in the case of moving two-axle wheeled vehicle with controlled suspension on various quality roads cover level of comfort movements increased by 8 to 41% when the frequency controlling effect of 15 Hz and 13...59% at the frequency control action of 50 Hz compared to uncontrolled suspension.

Conclusions
1. The continuous changes system algorithm in the level of damping shock absorber suspension wheel machine to improve the driver and passengers comfort.
2. The proposed algorithm efficiency is proved by simulation methods. It is established that at rectilinear movement on different roads with different speeds, the comfort level of the movement is increased by 8 to 41% when the frequency controlling effect of 15 Hz and 13...59% at the frequency control action of 50 Hz compared to uncontrolled suspension.

References
[1] Novikov, V.V., Pozdeev, A.V., Diakov, A.S. Research and testing complex for analysis of vehicle suspension units (2015) Procedia Engineering, 129, pp. 465–470.
[2] Du H, Sze K Y and Lam J 2005 Semi-active H∞ control of vehicle suspension with magneto-rheological dampers J. Sound Vib. 283 981–96.
[3] Poussot-Vassal C, Šename O, Dugard L, Gaspar P, Szabo ZandBokor J 2008 A new semi-active suspension control strategy through LPV technique Control Eng. Pract.161519–34.
[4] Fallah M S, Bhat R and Xie W F 2009 New model and simulation of Macpherson suspension system for ride control applications Veh. Syst.Dyn.47195–220.
[5] Laws S M 2010 An active camber concept for extreme maneuverability: mechatronic suspension design, tire modeling, and prototype development PhD Diss. StanfordUniversity.
[6] Zhileykin, M.M., Kotiev, G.O., Nagatsev, M.V. Comparative analysis of the operation efficiency of the continuous and relay control systems of a multi-axle wheeled vehicle suspension (2018) IOP Conference Series: Materials Science and Engineering, 315 (1), article № 012030.
[7] Zhileykin, M.M., Kotiev, G.O., Nagatsev, M.V. Synthesis of the adaptive continuous system for the multi-axle wheeled vehicle body oscillation damping (2018) IOP Conference Series: Materials Science and Engineering, 315 (1), article № 012031.
[8] Belousov, B.N., Merkulov, I.V., Fedotov, I.V. Controlled suspensions of automobiles(2004) Avtomobil'nayaPromyshlennost, (1), pp. 23–25.
[9] Belousov, B.N., Merkulov, I.V., Fedotov, I.V. Synthesis of dynamic control system for active mountings of multiaxis ATM (2004) Avtomobil'nayaPromyshlennost, (4), pp. 15–17.
[10] Belousov, B., Ksenevich, T.I., Naumov, S. Automated system to control steering and wheel springing parameters in vehicle locomotion module(2015) SAE Technical Papers.
[11] Belousov, B., Demik, V., Kozlova, A., Ksenevich, T., Naumov, S., Medvedev, E., Lyushnin, S., Kuzminkov, K. The schematic diagrams of actuators of an mechatronic wheel steering system (2018) FISITA World Automotive Congress 2018.
[12] Belousov, B., Ksenevich, T.I., Vantsevich, V., Naumov, S. An active long-travel, two performance loop control suspension of an open-link locomotion module for off-road applications (2014) SAE Technical Papers, 2014-January.

[13] Sarach, E., Kotiev, G., Beketov, S. Methods for road microprofile statistical data transformation (2018) MATEC Web of Conferences, 224, article № 04009.

[14] Alanoly J and Sankar S 1987 A new concept in semi-active vibration isolation J. Mech. Des. 109242–7.

[15] Sohn H-C, Hong K-S and Hedrick J K 2000 ‘Semi-active control of the Macpherson suspension system: hardware-in-the-loop simulations.’ 2000. Proc. of the 2000 IEEE Int. Conf. on Control Applications, pp982–7.

[16] Karnopp D, Crosby M J and Harwood. R A 1974 Vibration control using semi-active force generators J. Manufact. Sci. Eng. 96619–26.

[17] Choi S B, Choi Y T, Chang E G, Han S J and Kim. C S 1998 Control characteristics of a continuously variable ER damper Mechatronics 8143–61.

[18] Carlson J D, Catanzarite D M and Clair K A St 1996 Commercial magneto-rheological fluid devices Int. J. Mod. Phys. B 102857–65.

[19] Jalili N 2002 A comparative study and analysis of semi-active vibration-control systems J. Vib. Acoust. 124593–605.

[20] Vasiliev O. V. Optimization methods. World Federation Publishers Company, ICN, Atlanta, USA, 1996.

[21] GOST 31191.1-2004 (ISO 2631-1:1997) «Vibration and shock. Measurement of total vibration and assessment of its impact on humans. Part 1. General requirements». — M.: Publishingstandards 2004. — 45 p.