Development and validation of vehicle wheel load scaling method for formation of durability testing loading cycle

A I Bokarev, V A Kulagin and I A Nazarkov

Federal State Unitary Enterprise “Central Scientific Research Automobile and Automotive Engines Institute”, Moscow, Russia

E-mail: aleksandr.bokarev@nami.ru

Abstract. For a vehicle durability study, performed with simulation or laboratory testing, impact on the object should be described as set of time-domain load signals, which reflects accelerated service modes of a vehicle operation. For a newly developed vehicle these load signals can be based on previously prepared loading cycle of a similar vehicle with use of the load scaling theory. Importance and certainty of the load scaling theory is approved by active use among foreign car makers and engineering centers. However, mathematical description of the theory and scaling procedure are strictly classified, despite of being based on fundamentals of vehicle dynamics. In this paper the method of scaling of time-domain wheel and driveline loads is suggested; the method is based on proportions of size and mass properties of the original and the new vehicles. Assumed that previously recorded loading cycle corresponds to the service life of a vehicle. It is worth noting that level of loading and damage accumulation, shown with scaled loading cycle and after first prototypes testing on a proving ground, will be different because of impact of chassis stiffness and damping properties. Suggested scaling theory is being studied in FSUE “NAMI” and is on the validation and verification stage. First completed experiments showed that a multiaxial test rig could perform the vehicle suspension loading cycle, obtained with the theory, with a satisfactory accuracy. This makes utilization of the theory limited to early stages of vehicle development.

1. Introduction

Ensuring failure-free service life of a vehicle is a relevant engineering problem. Implementation of state-of-the-art methods and hi-tech laboratory equipment allows avoiding numerous iterations of component geometry optimization and saving time on research and development to create a design, which meets requirements of demanded durability and safety use of a vehicle.

Improvement of cutting-edge simulation and testing approaches of safety and durability assessment stimulates car makers to develop representative load spectra for use at early stages of a new vehicle development.

For a newly developed vehicle, such signals can be derived from road load data, previously recorded for a similar vehicle, or from a set of artificial (synthetic) load signals, which ensure the equivalent damage. Derivation of new set of load signals is possible to be performed by scaling, according to a developed scaling theory. Driving cycles for an original and a new vehicle are assumed to be identical. The result of scaling is adaptation of the original vehicle wheel loads to demanded loads for the new vehicle.
It is important to notice that scaled loading cycle is not applicable for testing of vehicle design prepared for production or at final stages of design verification. The load signal scaling theory, described in the article, is prepared only for durability assessment of vehicle components at the early stage of development. The cycle of load factors, determined for durability testing by scaling, may seriously differ from loads, which a production vehicle suffers during the same driving cycle performed on proving ground. Load distribution and damage accumulation depend on a set of parameters, even with different chassis setups for the same vehicle. Nevertheless, similar methods of time-domain wheel load signal scaling have been developed and are actively used by leading European engineering centers, such as “Porsche Engineering”, “Magna Engineering” and “Fraunhofer LBF” Institute for Structural Durability [1 – 3].

Scaling methods, prepared by car makers and engineering centers, are not in common use, so the problem of developing our own skills and knowledge in terms of loading cycle formation for newly developed vehicles becomes relevant. The core of the problem is formulation of our own time-domain load signal scaling theory. Load spectra derivation for a new vehicle is possible with use of this theory. It is assumed that driving cycles of the original vehicle (which load signals are recorded) and the new vehicle (which needs to have its own load spectra) are identical.

2. Derivation of wheel load signals scaling coefficients

The main principle of scaling is based on scaling coefficients derivation with use of preliminary determined wheel loads and vehicle size and inertia properties. Firstly, theoretical values of wheel normal forces shall be determined according to equations (1 – 4):

\[ R_{zFL} = M_a \cdot K_{lat} \cdot \left( K_{long} \cdot g - \frac{j_x \cdot h}{L} + \frac{j_y \cdot h}{B} \right) \]  
\[ R_{zFR} = M_a \cdot (1 - K_{lat}) \cdot \left( K_{long} \cdot g - \frac{j_x \cdot h}{L} - \frac{j_y \cdot h}{B} \right) \]  
\[ R_{zRL} = M_a \cdot K_{lat} \cdot \left( 1 - K_{long} \right) \cdot g + \frac{j_x \cdot h}{L} + \frac{j_y \cdot h}{B} \]  
\[ R_{zRR} = M_a \cdot (1 - K_{lat}) \cdot \left( 1 - K_{long} \right) \cdot g + \frac{j_x \cdot h}{L} - \frac{j_y \cdot h}{B} \]  

Where:
- \( R_{zFL}, R_{zFR}, R_{zRL} \) and \( R_{zRR} \) are normal forces at front left, front right, rear left and rear right wheels, [N];
- \( M_a \) – vehicle mass, [kg];
- \( K_{lat} \) – lateral weight distribution, [-];
- \( K_{long} \) – longitudinal weight distribution, [-];
- \( g \) – gravity acceleration, [m/s²];
- \( j_x \) – longitudinal acceleration/deceleration, [m/s²];
- \( j_y \) – lateral acceleration, [m/s²];
- \( h \) – center of mass height, [m],
- \( B \) – wheel track, [m];
- \( L \) – wheelbase, [m].

It is important to notice that \( j_x > 0 \) for longitudinal acceleration, \( j_y > 0 \) for right turn.

Influence of vehicle inertia from additional parts for wheel normal forces is described by formula (5) in longitudinal direction of inertia force and by formula (6) in lateral direction:

\[ R_{jx} = \frac{M_a \cdot j_x \cdot h}{L} \]  
\[ R_{jy} = \frac{M_a \cdot j_y \cdot h}{B} \]
Scaling coefficients can be determined according to the following:
1) Scaling coefficient $K_{Fzi}$ for normal forces at $i$-th wheel is determined with equation (7):

$$K_{Fzi} = \frac{F_{ziN}}{F_{ziO}}$$  \hspace{1cm} (7)

Where $F_{ziN}$ and $F_{ziO}$ are $i$-th wheel normal forces for the new and the original vehicles, [N].

2) Scaling coefficient $K_{FxF}$ for longitudinal forces at front wheels is determined with equation (8):

$$K_{FxF} = \frac{R_{jxN} \cdot k_{longN}}{R_{jxO} \cdot k_{longO}}$$  \hspace{1cm} (8)

Where:
- $R_{jxN}$ and $R_{jxO}$ are additional normal wheel forces caused by inertia force in longitudinal direction for the new and the original vehicles, [N];
- $k_{longN}$ and $k_{longO}$ are longitudinal weight distribution coefficients, [-].

3) Scaling coefficient $K_{Fxr}$ for longitudinal forces at rear wheels is determined with equation (9):

$$K_{Fxr} = \frac{R_{jxN} \cdot (1 - k_{longN})}{R_{jxO} \cdot (1 - k_{longO})}$$  \hspace{1cm} (9)

4) Scaling coefficient for lateral forces at front wheels are determined for each wheel:

$$K_{FYFL} = \frac{R_{jyN} \cdot k_{longN} \cdot k_{latN}}{R_{jyO} \cdot k_{longO} \cdot k_{latO}}$$  \hspace{1cm} (10)

Figure 1. Schemes of determination of wheel normal forces additional parts.
\[
K_{FyFR} = \frac{R_{jyn} \cdot k_{longN} \cdot (1 - \kappa_{latN})}{R_{jyo} \cdot k_{longO} \cdot (1 - \kappa_{latO})}
\]  
(11)

Where:
- \( R_{jyn} \) and \( R_{jyo} \) are additional normal wheel forces caused by inertia force in lateral direction for the new and the original vehicles, [N];
- \( \kappa_{latN} \) and \( \kappa_{latO} \) are lateral weight distribution coefficients.

5) Scaling coefficient for lateral forces at rear wheels are determined for each wheel:

\[
K_{FyR} = \frac{R_{jyn} \cdot (1 - \kappa_{longN}) \cdot k_{latN}}{R_{jyo} \cdot (1 - \kappa_{longO}) \cdot k_{latO}}
\]  
(12)

\[
K_{FyR} = \frac{R_{jyn} \cdot (1 - \kappa_{longN}) \cdot k_{latN}}{R_{jyo} \cdot (1 - \kappa_{longO}) \cdot (1 - \kappa_{latO})}
\]  
(13)

6) Overturning torque \( M_X \) and aligning torque \( M_Z \), applied to wheel center, mostly depend on values of lateral force \( F_Y \), so scaling coefficients for these loading factors are equal to lateral force scaling coefficients:

\[
K_{MxF} = K_{MzF} = K_{FyF} = \frac{R_{jyn} \cdot k_{longN} \cdot k_{latN}}{R_{jyo} \cdot k_{longO} \cdot k_{latO}}
\]  
(14)

\[
K_{MxR} = K_{MzR} = K_{FyR} = \frac{R_{jyn} \cdot (1 - \kappa_{longN}) \cdot (1 - \kappa_{latN})}{R_{jyo} \cdot (1 - \kappa_{longO}) \cdot (1 - \kappa_{latO})}
\]  
(15)

\[
K_{MxF} = K_{MzF} = K_{FyF} = \frac{R_{jyn} \cdot k_{longN} \cdot k_{latN}}{R_{jyo} \cdot k_{longO} \cdot k_{latO}}
\]  
(16)

\[
K_{MxR} = K_{MzR} = K_{FyR} = \frac{R_{jyn} \cdot (1 - \kappa_{longN}) \cdot (1 - \kappa_{latN})}{R_{jyo} \cdot (1 - \kappa_{longO}) \cdot (1 - \kappa_{latO})}
\]  
(17)

7) Scaling coefficients \( K_{MyF} \) and \( K_{MyR} \) for braking torques at front and rear wheels are determined with equations (18) and (19):

\[
K_{MyF} = \frac{R_{jxn} \cdot k_{longN} \cdot r_{dfN}}{R_{jxo} \cdot k_{longO} \cdot r_{dfO}}
\]  
(18)

\[
K_{MyR} = \frac{R_{jxn} \cdot (1 - \kappa_{longN}) \cdot r_{drN}}{R_{jxo} \cdot (1 - \kappa_{longO}) \cdot r_{drO}}
\]  
(19)

Where:
- \( r_{dfN} \) and \( r_{dfO} \) are dynamic front wheel radii of the new and the original vehicles, [m];
- \( r_{drN} \) and \( r_{drO} \) are dynamic rear wheel radii of the new and the original vehicles, [m].

Load factors of the original vehicle shall be multiplied by scaling coefficients, described above, to determine time-domain values of load factors for the new vehicle.

3. Derivation of driveline torque signals scaling coefficients

For driveline torque scaling, simplified algorithm of driving wheels torque derivation is used, which excludes consideration of rotating masses. Driving mode for the new vehicle and the original vehicle are assumed identical, so driveline torque scaling coefficients can be derived by summands of the vehicle dynamic equation [4, 5]:

\[
F_T = F_k + F_a + F_w + F_j
\]  
(20)

Where:
- \( F_T \) is the total traction force, [N];
- \( F_k = G_a \cdot f_k \cdot \cos(\alpha) \) is the rolling resistance force, [N];
- \( G_a \) is the vehicle weight, [N];
- \( f_k \) is the rolling resistance coefficient, [-];
- \( a \) is the angle of climb, \([°]\);
- \( F_a = G_a \cdot \sin(a) \) is the climb resistance force, [N];
- \( F_j = \delta \cdot M_a \cdot j_x \) is the inertia force;
- \( \delta \) is the rotating mass coefficient, [-];
- \( F_w = 0.5 \cdot C_x \cdot \rho \cdot A_F \cdot V_a^2 \) is the drag resistance force, [N];
- \( C_x \) is the drag resistance coefficient, [-];
- \( \rho \) is the air density, [kg/m\(^3\)];
- \( A_F \) is the vehicle frontal area, [m\(^2\)];
- \( V_a \) is the vehicle speed, [m/s].

Inertia, rolling resistance and climb resistance forces are dependent on constant properties, so it is reasonable to define scaling coefficients relatively to these the equation summands. Formula (21) describes the desirable form of the equation:

\[
F_T - F_w = F_k + F_a + F_j
\]  

(21)

Thus, formula (21) allows getting the equation of proportion ratio (22) between the new vehicle and the original vehicle, as the vehicle mass is a common multiplier included in all summands.

\[
\frac{(g \cdot f_{kO} \cdot \cos(a) + g \cdot \sin(a) + \delta_o \cdot j_x)}{(g \cdot f_{kN} \cdot \cos(a) + g \cdot \sin(a) + \delta_N \cdot j_x)} = \frac{M_{aN}}{M_{aN}}
\]  

(22)

With the expression (22), it is possible to define the total traction force of the new vehicle and then to define traction forces for all driven wheels. The load signal scaling theory assumes number of particular cases of total traction force distribution at driven wheels:

A) Both axles of the new and the original vehicles are driving. It is reasonable to keep traction forces distribution derived for the original vehicle;

B) The original vehicle has one driving axle; the new vehicle has both driving axles. It is reasonable to keep traction forces distribution between sides derived for the original vehicle with the new vehicle torque distribution between axles taken into account;

C) The original vehicle has both driving axles; the new vehicle has one driving axle. It is reasonable to keep traction forces distribution between sides derived for the original vehicle with pulling forces arising in center of driven wheels taken into account.

4. Validation and verification of the time-domain load signals scaling theory

The above-described load signal scaling theory is on the testing and verification stage. It will become a noticeable addition to wide range of testing and simulation capabilities in terms of vehicle durability properties research. The theory is being used for such tasks as:

- Synthesis of combined loading cycle for suspension assembly research performed via CAE simulations or laboratory testing on multiaxial test rigs;
- Synthesis of combined loading cycle for suspension components (wheel hub assembly, driveshaft, suspension links, subframe, spherical joints, bushings and mounts) research performed via CAE simulations or laboratory testing;

Validation and verification stage has already achieved positive reliable results and has approved adequacy of using it for design safety and durability analysis of first versions of a new vehicle design. Figure 2 presents an actual process of the suspension assembly testing on a multiaxial test rig during validation and verification procedure. The study object in Figure 2, mounted on the test rig, is deliberately hidden for reasons of leakage of classified information.
Figure 2. Validation testing of a suspension assembly on a multiaxial test rig in FSUE “NAMI”.

Set of load signals was determined according to the time-domain load signal scaling theory; the loading cycle ensures simultaneous complex loading by forces $F_X$, $F_Y$, $F_Z$ and torques $M_X$, $M_Y$, $M_Z$ applied to both sides with braking system operation and driveline torques taken into account. Synthesized loading cycle is a combination of service driving modes, located with a preset sequence and repetition frequency:
- Braking with deceleration up to 1g;
- Acceleration up to 1g;
- Driving at the Mountain Road with longitudinal and lateral accelerations up to 1g;
- Driving at “Profiled Cobblestone” and “Belgian Pavement” roads at constant speed;
- “Eight” maneuver.

Driving modes repetition frequency defines equivalence of laboratory testing duration to vehicle service mileage. The repetition frequency is based on road testing performed for the original vehicle. Driving modes combination ensures necessary level of complex loading of a suspension assembly and proper primary assessment of components safety and durability.

It has to be noted that the study object was deliberately chosen as quite different from the original vehicle, for which load signals were recorded on proving ground. The level of difference is up to 40%, and it contains changed properties, such as:
- Vehicle size and mass properties;
- Elastokinematic, stiffness and damping suspension properties;
- Range of wheel vertical travel.

Figure 3 presents results of signal repeatability analysis, performed for the rear suspension assembly of the newly developed vehicle, in form of comparison of extreme wheel loads, targeted by determined loading cycle and achieved during the testing.
Adequacy of suspension assembly performance with scaled loading cycle is analyzed by comparison of achieved pseudo damage level by particular load factors response, recorded by strain-gauge hubs of test rig. Results of accumulated pseudo damage are obtained from this testing and from previously performed validation testing. According to them, pseudo damage level is insufficient for all wheel load signals. This can be explained by test rig control difficulties and small amount of performed iterations of testing.

Adequacy of suspension assembly performance with scaled loading cycle is analyzed by comparison of achieved pseudo damage level by particular load factors response, recorded by strain-gauge hubs of test rig. Results of accumulated pseudo damage are obtained from this testing and from previously performed validation testing. According to them, pseudo damage level is insufficient for all wheel load signals. This can be explained by test rig control difficulties and small amount of performed iterations of testing.

Figure 3. Extreme wheel loads comparison. $F_X$ is longitudinal force, [N]; $F_Y$ is lateral force, [N]; $F_Z$ is normal force, [N]; $M_X$ is overturning torque, [Nm]; $M_Y$ is braking/traction torque, [Nm]; $M_Z$ is aligning torque, [Nm]; MG_torque is main gear input torque, [Nm]; index “le” means “left wheel”, index “ri” means “right wheel”.

Figure 4. Achieved pseudo damage level comparison. $F_X$ is longitudinal force, [N]; $F_Y$ is lateral force, [N]; $F_Z$ is normal force, [N]; $M_X$ is overturning torque, [Nm]; $M_Y$ is braking/traction torque, [Nm]; $M_Z$ is aligning torque, [Nm]; MG_torque is main gear input torque, [Nm]; index “le” means “left wheel”, index “ri” means “right wheel”.

Comparative study of testing objects performance on a test rig allowed preparation of an action list to improve testing procedure and to correct critical errors, correct test rig settings have been found for carrying out sufficient volume of iterative testing. That shall allow achievement of better convergence of target and actual wheel load signals.

Figure 5 shows results of iterative testing of target wheel load signal reachability on a test rig, performed for “Mountain Road” driving mode. Results are presented as a graph of wheel load signals convergence (mean-square derivation of responded load signal from input load signal), which is a key criteria of analysis. Good convergence is limited by error value of 10-15%, so it is quite reachable.

![Figure 5](image.png)

**Figure 5.** Results of iterative testing performed for “Mountain Road” driving mode.

### 5. Conclusion

The time-domain load signal scaling theory is an important tool for safety and durability analysis of newly developed vehicle components. Use of available set of initial data and uncomplicated formulae allows the theory to be relatively easy to be used and at the same time to ensure satisfactory results, enough for early stages of vehicle development.

The developed scaling theory finds its application in projects, performed by FSUE “NAMI”, for the following tasks:

- Suspension assembly simulation and testing;
- Wheel hub assembly simulation and testing;
- Drive shafts simulation and testing;
- Spherical joints simulation and testing;
- Subframe simulation and testing;
- Bushings, rubber mounts and hydromounts simulation and testing.

The developed scaling theory is on the validation and verification stage. First results of theory application show reliable and positive quality, which can contribute to personnel skill progress, simulation and testing method improvement. Works on the wheel load signal scaling theory shall be continued in terms of improvement of experimental results accuracy and reliability.

References
[1] Numata F and Gattringer O 2018 JSAE Annual Congress (Spring) 409 20185409
[2] Müller A et al 1993 Baut.: Rech. und Ver. 19 65-21
[3] Grubisic V 1994 Int. J. of Veh. Des. 15 8-26
[4] Husainov A Sh and Selifonov V V 2008 Automobile Theory. Compendium of Lectures (Ulyanovsk: UlGTU) p 121
[5] Petrushov V A 2008 Automobiles and Trailer Trucks. New Technologies of Research of Rolling and Drag Resistance (Moscow: Torus Press) p 352