THE ANALYSIS OF DOMINANT FACTORS OF LONGITUDINAL TRUCK VIBRATION WITHIN FREQUENCY RANGE 0-5 Hz

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Abstract: *Human body is especially sensitive to longitudinal vibration within frequency range up to 5Hz. The basic source of this vibration comes from the leap variation of engine torque, frequently followed by the intensive transmission vibration. The influential parameters of transmission vibration which are mostly analyzed are the following: time and flow of torque growth, frictional clutch characteristic, clearances in transmission, adhesion among wheel and ground, stiffness and damping characteristics of transmission and tires, etc. Based on the experimental investigation results of a truck of overall weight 19t, a mathematical model is developed so as to describe properly the longitudinal vehicle and driver vibration. In fact a model consists of two parts: one that includes transmission (linear model) and the other that introduces planar model for the analysis of vehicle longitudinal vibration. The paper presents in details a model that describes transmission torsional vibration of a truck wheel formula 4x4, easily transformed to a model 4x2 by use of simple mathematical constraints.*

KEYWORDS: LONGITUDINAL ACCELERATION, TORSIONAL VIBRATION

1. Introduction

The most frequently analyzed influential factors of vehicle longitudinal vibration are [2, 3]: time and torque increase flow, frictional clutch characteristic, clearances in transmission, adhesion between wheel and ground, stiffness and damping characteristics of transmission and tires, etc.

In this paper a special attention has been paid to the analysis of processes caused by the shunt variation of engine torque, being a basic cause of vibration in the frequency domain up to 5 Hz, in which humans are particularly sensitive.

2. Mathematical models for calculation of vehicle longitudinal vibration

Mechanical transmission can modelled as a large number of inertial masses tied with elasto-damping elements that represent particular transmission elements. Published papers which analyze vibration in low frequency domain mostly apply 2-5 inertial masses. The simplest models [3] are based on application of two inertial masses, one of which represents the engine with flywheel and the other represents the vehicle body. These inertial masses are linked with elasto-damping elements that apply both on transmission and tires. The most of the damping effect comes from the slip in tire-to – ground contact area. Therefore, correct modelling of dependence of adhesion coefficient and wheel slip deserves special attention.

Radial force on the wheel is directly dependant to wheel slip, i.e. difference between rotation wheel speed and vehicle speed. In references there are many formulas based on empirical data, most of them are complicated. For the analysis presented in this paper a model [2] is adopted, which defines dependence between slip and adhesion coefficient in the following way:

$$\mu = \mu_h \cdot \sin\left(2 \cdot \pi \cdot \lambda \frac{1}{4 \cdot \lambda_h}\right) \quad \text{for } |\lambda| \le \lambda_h \text{ ....(1)}$$

or

$$\mu = \frac{\mu_e - \mu_h}{\lambda_e - \lambda_h} \cdot (\lambda - \lambda_e) + \mu_h \quad \text{for } |\lambda| > \lambda_h \text{ ....(2)}$$

where μ_h -maximal value of adhesion coefficient,

μ_e -adhesion at maximal slip,

λ_h -slip at maximal adhesion,

λ_e -maximal value of slip (usually +/-1).

A suitable method for modelling of transmission clearances is so called "dead zone" model, describing processes caused by clearances in details, by establishing series of limitations in order to represent credibly processes that occur in transmission (for example, a phenomenon of pulling force in the contact area of gear in conditions of high damping in transmission elements [3] .

3. Mathematical model for calculation of longitudinal vibration of 4x4 and 4x2 wheel scheme vehicle

The choice of optimal mathematical model capable to represent process has to comply with two opposed requirements. One of them is to describe credibly the process which is subjected to the analysis, taking into consideration as many relevant factors as possible, and the other is to be applicable in case of as few degrees of freedom as possible.

On the basis of the results achieved by experimental investigation results of a truck of overall weight 19t, given in details in [4], a mathematical model has been created to describe adequately vehicle and driver longitudinal vibration caused by shunt variation of engine torque, being a result of drivers command.

The adopted mathematical model consists in fact of two parts: one that covers power train (linear model) and the other which applies plane model for analysis of vehicle longitudinal vibration. A model that describes transmission torsional vibration of a 4x4 vehicle is shown in Figure 1. and can easily be transformed into a model 4x2 by means of mathematical restrictions.

Vehicle plane model is often composed of sprung mass (body) and unsprung mass (wheels, axles) linked to each other by elasto-damping elements. In case of heavy trucks, cab is suspended flexibly and the model should be enlarged by a single sprung mass-cab, as well as driver's seat, also flexibly suspended. Since the primary goal of this paper is the analysis of factors of longitudinal vehicle vibration excited by the vehicle driving system, the further attention will be paid to a model of driving system and for review of the whole vehicle model readers are advised to consult [4].

Generalized coordinates that describe vibration according to the adopted model of vehicle driving system, shown in Figure 1. are:

q(1)- rotation angle of engine flywheel
q(2)- rotation angle of gearbox output shaft
q(3)- rotation angle of input shaft of power distributor
q(4)- rotation angle of rear axle differential gear
q(5)- rotation angle of rear wheel rim
q(6)- rotation angle of rear tire wade surface
q(7)- rotation angle of front axle differential gear
q(8)- rotation angle of front wheel rim
q(9)- rotation angle of front tire wade surface

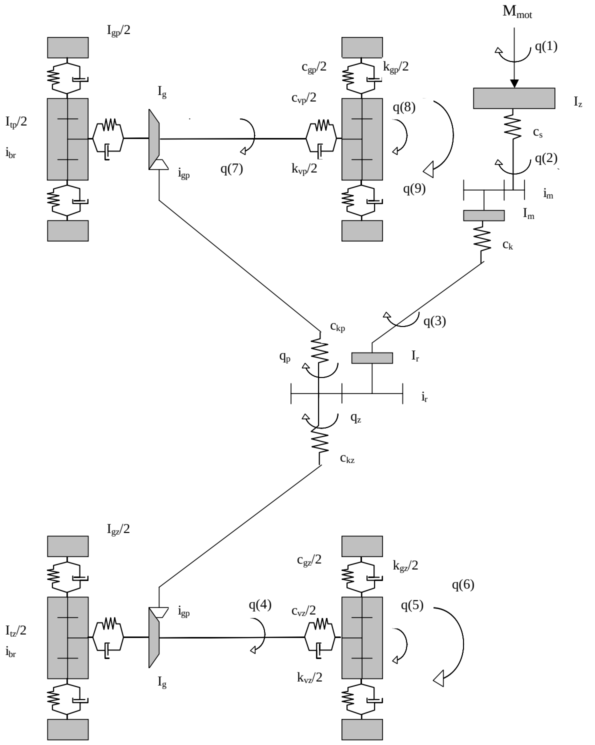

Figure 1.: Driving system vibration model

c_s – clutch stiffness
c_k – cardan shaft stiffness
c_{kz} - cardan shaft stiffness
c_{vz} – rear axle half-shaft overall stiffness
c_{tz} – rear tires overall torsional stiffness
c_{kp} - cardan shaft stiffness
c_{vp} - front axle half-shaft overall stiffness
c_{tp} – front tires overall torsional stiffness
k_{vz} - rear half-shaft damping
k_{tz} – rear tires damping
k_{vp} - front half-shaft damping
k_{tp} – front tires damping
M_{mot} - excitation originated by engine
I_z - flywheel, clutch housing and crankshaft moment of inertia
i_{m2} - gearbox ratio (second gear)
I_m - moment of inertia calculated for gearbox output shaft
I_r - moment of inertia of the elements of power distributor
i_g - rear axle gear ratio
I_{dz} - moment of inertia of main gear and differential gear
i_{br} - side gear ratio
I_{tz} - moment of inertia of rear wheel hubs, gears and rims
I_{gz} - moment of inertia of rear tire wade surface
I_{dp} - moment of inertia of main gear and differential
I_{tz} – moment of inertia of front wheel hubs, gears and rims
I_{gp} - moment of inertia of front tire wade surface

By use of D'Alambert principle [1], differential equations are written to describe small vibration around balance position[4].
Differential equations are nonlinear, with constant coefficients and are to be solved numerically, by use of Kutta-Merson method. The sample of increment is automatically changed. Initial one was specified to be h_0=0.01 and the adopted number of points was n=512, enough to enable analysis of vehicle longitudinal vibration for 5s duration time, in frequency domain 0.2-50Hz.

4. The influence of excitation originated from engine

On the basis of the experiments performed, it has been discovered that there are three basic cases of excitation coming from the engine that provoke high level of vehicle longitudinal vibration:

- Vehicle start with instant clutch release;
- Intensive acceleration after the engine brake regime;
- Instant accelerator pedal release followed by the intensive acceleration, also known as “back-out” – “tip-in” regime.

The appearance of high values of torque at vehicle start, with instant clutch release, close to the edge of engine performance, is not a phenomenon that often takes place in service conditions of such a kind of vehicle. Therefore, it does not represent a major significance for the analysis of vehicle longitudinal vibration from the aspect of vibration comfort. It also appears that the optimal case for comparison of levels of longitudinal accelerations is intensive acceleration after the engine brake regime, due to a minimal influence of driver’s behaviour.
The increase of engine torque can be defined by means of different approaches, but for this analysis the increase of torque has been specified in the form of ramp function with various duration time of engine torque increase up to the maximal value. Engine torque shunt happens in short time period starting from the initial value M_{min} up to the M_{max} according to the following expression:

$$M_{mot}(t) = \begin{cases} M_{min} & t < T_1 \\ (M_{max} - M_{min}) \cdot \dfrac{t - T_1}{T_2 - T_1} & T_1 < t < T_2 \\ M_{max} & t > T_2 \end{cases}$$

where time of torque increase is T_2-T_1 (ΔT , in the further text).
Bearing in mind that in case of presence of clearances in the system a more realistic view of relevant processes is enabled by the analyses that take engine braking as the initial state. The further text will deal with such a model.
Figure 2. shows flow of the engine torque applied as excitation for longitudinal vibration calculation, by use of the adopted mathematical model. Duration time of engine torque increase varied within the limits registered during the experimental investigation.

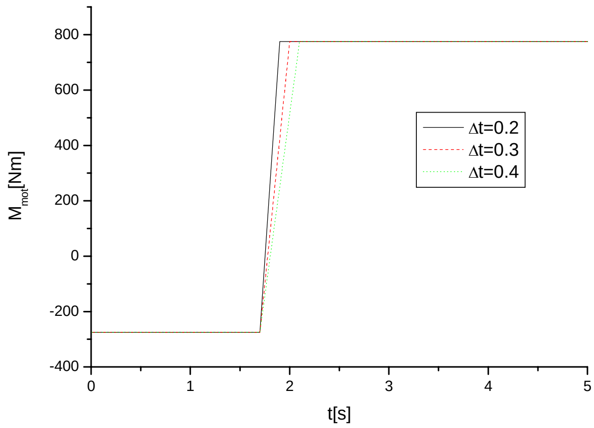

Figure 2.: Engine torque variation

By use of software “OSCILACIJE” [4], written in Pascal language, a calculation of longitudinal vibration has been performed, with the accent on determination of variables measured in the experimental part of the investigation. Figure 3. shows the cardan shaft torque variation at the gearbox output and driver’s acceleration for various time periods (ΔT) of engine torque increase, applied for unloaded vehicle of 4x4 wheel formula, accelerated in second gear, usually applied for initial vehicle start. The influence of time period duration (ΔT) is obvious, because maximal cardan shaft torque (M) varied between 3660 Nm (ΔT =0.2s), via 2930 Nm (ΔT =0.3s) up to 2260 Nm (ΔT =0.4s). Analogous behaviour is noticed for driver’s acceleration (a_{voz}).

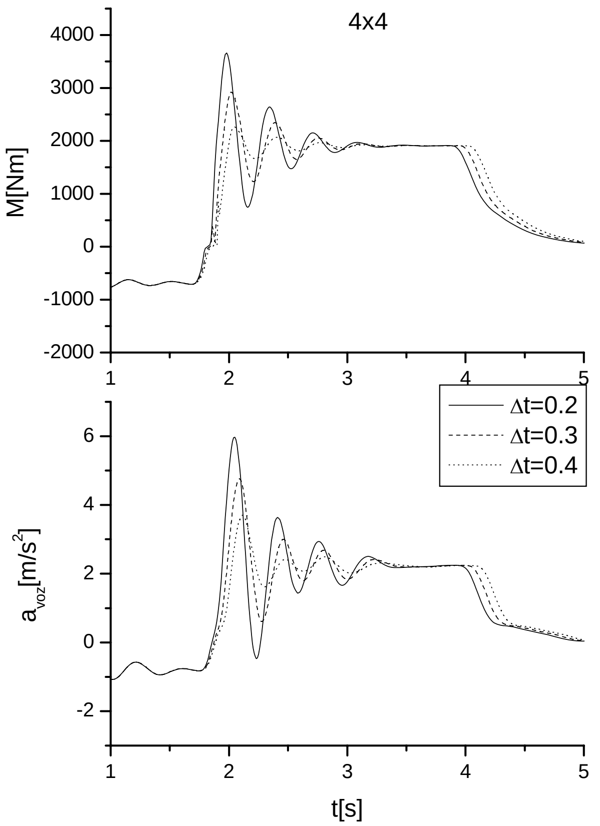

Figure 3.: Cardan shaft torque and driver's acceleration

5. The influence of clearances in driving system

Between the elements of motor vehicle transmission there are minor clearances, sum of which can reach 30-40° on crankshaft. In cases when transmission torque changes the prefix sign, it is necessary to introduce the influence of clearances in the dynamic model, by use of appropriate mathematical equations. This is contributes to a more realistic view to the processes that occur in transmission.

According to [3], the most convenient for application is modified model of "dead zone", which gives good results in conditions of high levels of damping in transmission elements. This is achieved by introduction of additional conditions so as to enable existence of pulling force in gear contact area or other transmission elements. The model can be described by the following equations:

$$M_{mz} = \begin{cases} c_s(\varphi_{rel} - \alpha/2) + k_s \cdot \dot{\varphi}_{rel} & for\, \varphi_{rel} > \alpha/2 \\ c_s(\varphi_{rel} + \alpha/2) + k_s \cdot \dot{\varphi}_{rel} & for\, \varphi_{rel} < -\alpha/2 \\ 0 & for\, |\varphi_{rel}| \le \alpha/2 \end{cases}$$

$$M = \begin{cases} 0 & if\, M_{mz} < 0\, and\, \varphi_{rel} > 0\, or\, if\, M_{mz} > 0\, and\, \varphi_{rel} < 0 \\ M_{mz} & otherwise \end{cases}$$

where α - clearance

M – driveshift torque

φ_1, φ_2 -deflection angles

$\varphi_{rel} = \varphi_1 - \varphi_2$

c_s – coefficient of stiffnest

k_s – coefficient of damping

It should be noticed that the tested vehicle was equipped with the clutch of complex characteristic, producing the effects of torsional vibration similar to those caused by clearances in the system.

Clearances in the system are present in minor or major extent at each attachment spots of the transmission elements, but in this phase only the influence of clearances in main gear, differential gear and half-shafts assembly have been analyzed as dominant.

Figure 4. shows separately the results of influence of clearance for the case of engine excitation in ramp form (ΔT =0.3s) on cardan shaft torque and longitudinal driver's acceleration. From the figure it is obvious that longitudinal vibration increase with clearance increase, especially in the first vibration periods. Besides, due to clearances, cardan shaft torques shows broken amplitudes in case of zero torque.

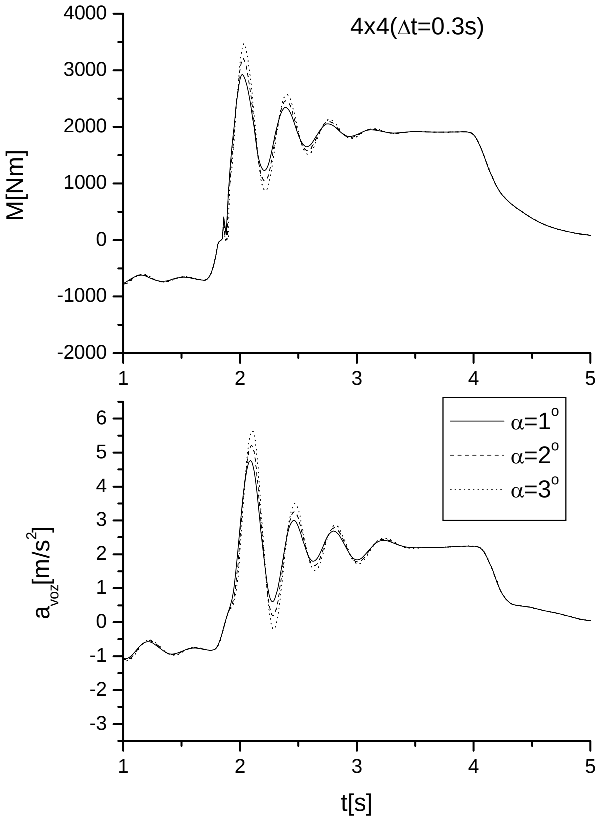

Figure 4.: Comparison between cardan shaft torque (M) and driver's longitudinal acceleration (a_{voz}) for various extent of clearance

6. The influence of gear engaged

Variation of gear engaged has undoubtedly significant influence on vehicle and driver longitudinal acceleration. Figure 5. presents the variation of cardan shaft torque and driver's longitudinal acceleration calculated on the basis of the adopted model for 2-nd 3-rd and 4-th gear engaged in condition of shunt variation of engine excitation ("ramp"). The calculation has been performed for unloaded vehicle of 4x4 wheel formula and engine torque increase time was ΔT =0.3s. It is obvious that torque shunt on cardan shaft significantly decreases in case of 3-rd gear engaged, even at the first vibration amplitude and vibration damping comes very fast. For the engine excitation in "ramp" form in 4-th gear, shunt torque of cardan shaft is minimal, so it can be concluded that in higher gears it is not significant.

The similar situation is in case of driver's longitudinal acceleration, although there is noticeable acceleration shunt in 4-th gear, due to vehicle cab vibration.

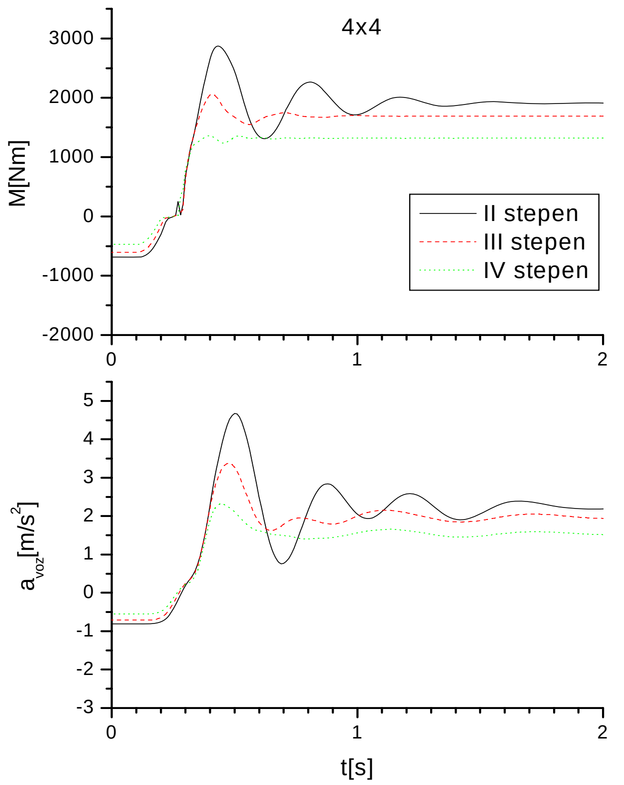

Figure 5.: Cardan shaft torque (M) and driver's acceleration (a_{voz}) in cases of various gears engaged

7. The influence of vehicle wheel formula

The previous discussion has pointed out on significant influence of duration time of torque increase on the levels of longitudinal vibration. The following text will focus on comparative analysis of dynamic processes of a vehicle subjected to the variation of wheel formula, either 4x4 or 4x2.

The analyses performed by use of the adopted mathematical model indicate that engagement of various wheel formulas, through variation of vehicle dynamic parameters, shows a significant influence on vehicle longitudinal acceleration, i.e. vibration comfort in "fore and aft" direction. Figure 6. presents comparative review of calculation results of variables concerning both cases (4x4 and 4x2) for duration time of torque increase Δt =0.3s. Differences in vibration period and amplitude levels are obvious.

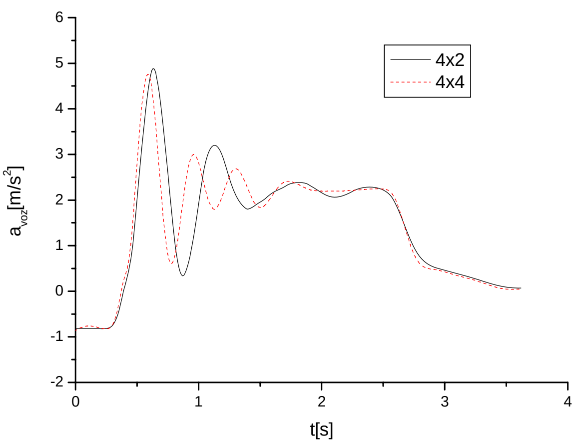

Figure 6.: Driver's seat acceleration for various wheel formulas of unloaded vehicle

It is well known that mass ratio of loaded and unloaded vehicle is fairly high, and consequently the influence of overall vehicle mass on the levels of longitudinal accelerations is significant. Therefore, the analysis of driver's longitudinal acceleration for the case of loaded vehicle with 19t overall mass (Figure 7.). Duration time of torque increase Δt =0.3s and measure of clearance are the same as in analyses concerning unloaded vehicle.

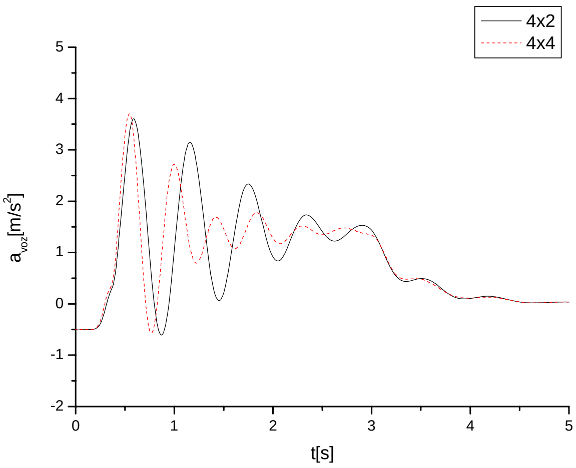

Figure 7.: Driver's seat acceleration for various wheel formulas of loaded vehicle and increased measure of clearances

Similar to the case of unloaded vehicle, it is clear that there is a noticeable difference with respect to amplitudes and vibration periods. Vehicle with wheel formula 4x4 shows relatively short period of vibration damping, while this is not the case for vehicle with 4x2 wheel formula. It can be undoubtedly predicted that driver's response to vibration comfort of a 4x2 vehicle will be unfavourable due to a long and intensive vibration in whole period of vehicle acceleration.

8. Conclusions

The analyses of longitudinal vibration of loaded and unloaded vehicle confirms the significance of the influence of excitations originated from the engine, clearances in driving system, engaged gear and exceptional influence of vehicle wheel formula.

The dominant influence of excitation coming from the engine is mostly a result of duration time of engine torque increase.

About clearance influence, it has been confirmed that longitudinal vibration are increased with clearances, especially in first vibration periods.

The choice of higher gear engaged causes decrease of cardan shaft torque shunt, noticeable even in the first damping amplitudes.

Engagement of wheel formula 4x4 or 4x2, without other dynamic parameters variation, directly influences variation of driver's vibration comfort. Vehicle of 4x4 wheel formula shows relatively short damping period, unlike 4x2 vehicle. This fact is more significant for loaded vehicle.

Driver's vibration comfort is estimated to be reduced in case of 4x2 wheel formula due to a long and intensive vibration in whole period of vehicle acceleration.

9. References

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