Effects of motorbike front suspension component operation on torsional vibration frequencies

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Abstract. Riding a motorbike requires much more experience than other types of vehicles. However, not only insufficient skills may contribute to dangerous driving situations, but also vibrations generated in the steering system of a motorbike. This paper presents a method of vibration analysis on the example of a motorbike steering system using a transmittance model and shows how to extract the relevant frequencies for analysis. The effects of selected stiffness and damping parameters are analysed and an assessment is made of how the frequencies for wobble vibration will change over the life of the steering components. The results were presented using Bode diagrams.

Keywords: suspension, motorbike, vehicle dynamics, vibration comfort

1. Introduction

The observed trend of decreasing the number of fatalities in accidents of different vehicle groups is not present among motorbikes [1]. On the example of Poland it may be stated that motorbikes constitute 4.8% of all registered motor vehicles. However, motorcyclists are involved in 6.8% of accidents of which 9.7% are fatal accidents. The most common cause of accidents is still not adjusting the speed to the road conditions (54%) [2]. Unfortunately, due to the way statistics are collected, it is usually impossible to determine which number of accidents was caused by typical reasons (not adapting speed to the conditions, improper overtaking, failing to keep a safe distance, failing to yield right of way), and which could have been preceded by peculiar dynamic processes (including vibrations) leading to loss of control over the vehicle. In addition, this is extremely difficult to establish. Therefore, in addition to the development of active and passive protection systems in motorbikes, it is important that the motorbike is properly constructed and thus naturally stable and resistant to the generation and propagation of vibrations.

The human body has certain properties by which vibrations can be damped, and the sitting position of the rider has a significant influence here. As part of the vibrations are damped by the lower body, only a small part of the vibrations is affected by the human head. It should be noted, however, that a person in this position is 10 times more sensitive to vibrations with a frequency of 5 Hz than 100 Hz [3], and vibrations with frequencies of up to 12 Hz affect all organs in the human body [4]. Vibration can cause muscle fatigue and increase susceptibility to spinal injuries [5]. One method of reducing vibration is described in [6], [7], but nothing can replace a well-designed motorbike structure.

Unlike a car, a motorbike steering system does not allow the introduction of the gear ratio found in a car steering gearbox or other assist mechanisms. This makes the effects of the roadway on the wheel, and thus the rider's hands, much more noticeable. Sometimes they can be so great that they overcome the resistance force exerted by the driver and cause a loss of stability and control of the vehicle. For this
reason, proper front suspension design and geometry are particularly important for proper motorbike handling.

Over time, the performance of vehicle components deteriorates. For this reason, research is being conducted to assess to what extent this occurs. As indicated in the work [5], decreasing or increasing the value of the stiffness coefficient by 1/3 of the initial value is of little significance in this respect. The same work also analysed the effect of changing the value of the shock absorber damping coefficient, where the initial value of this parameter was reduced and increased by half. In this case, a rather obvious conclusion was drawn that as the value of the damping coefficient increases, the amplitude of the acceleration decreases. An obvious conclusion, as this is the purpose of using dampers in suspension. However, while in the case of coil springs there are fewer factors causing a change in parameters, for shock absorbers the parameters will depend, among other things, on the operating conditions.

In the case of coil springs used in motorbike suspensions, they can shorten over time, reducing their resting length and therefore their preload. During suspension operation, the wire from which the coil spring is made can wear away, resulting in a reduction in spring stiffness [8]. In some types of suspension, the spring preload can be adjusted, but above a certain value set by the manufacturer, replacement is necessary. For example, for the Yamaha R6 motorbike, the resting length of the spring is 247.0 mm, while the limiting length, is 242.1 mm [9].

The damping force of shock absorbers varies with the speed at which the piston moves, and this in turn causes the temperature to rise. Of course, like engine oils, the oils used to fill shock absorbers can operate over a wide temperature range, nevertheless the viscosity of the oil will change. It can also be different even for the same type of shock absorber, as shown in [10]. The sealing rings used in the suspension (see Fig. 2) also decrease in elasticity as a result of ageing processes, and small cracks can form on their surface. The sealing rings are particularly stressed during heavy or emergency braking, when the rear wheel can detach from the ground and transfer the entire weight of the motorbike and rider to the front wheel, resulting in a loss of shock absorber oil. The effect of shock absorber oil loss was investigated in [11]. Although a shock absorber with as much as 40% oil loss was tested, it was found that this causes a distortion of the performance and speed graphs compared to a new shock absorber. The instantaneous values of damping forces and damping power were also reduced. Therefore, motorbike manufacturers recommend relatively frequent inspection of the front suspension (every 7000 km or 6 months) [9]. If it is not possible to eliminate the vibrations, it becomes necessary to use damping systems. The need for increased torsional vibration damping in the motorbike steering system has already been recognised in older motorbike designs. Numerous patents have been issued with various means of reducing vibration, many of which include viscous dampers. Correct vibration damper design should include frequency analysis, but very few models have been developed for frequency analysis of motorbike steering system vibrations. It is still difficult to find works in which a detailed analysis of the influence of individual parameters describing the dynamics of the steering system has been carried out. In addition, the studies discussed earlier only concern car suspensions. The few works based on an analytical model include [12], where an analysis of the dynamics of a simplified motorbike model and an evaluation of its behaviour based on eigenvalues was undertaken. The small number of publications in recent years related to motorbike vibration can be attributed to some kind of shift away from analytical methods to engineering software and MBS and finite element methods [13], but these do not provide a description useful to the controller.

Therefore, the following section will present a model of a motorbike steering system in the form of an operator transmittance. This is not only a useful form used for controller development, such as in [14], but can also find application in mechanical system design, as will be presented in this paper.

2. Equations and mathematical expressions

The simulation studies used the motorbike steering model described in detail in the paper [15], which describes, among other things, its verification process. The following is the most important information necessary to understand the analyses carried out in the following section. The physical model corresponds to the motorbike steering system mounted in a special drum test rig presented in [15], which was used to verify the mathematical model.
Figure 1. A physical model of the motorbike steering system

\( \psi, \varphi \) – steering and steering wheel angles and rotations, \( z \) – vertical displacements, \( J \) – moment of inertia, \( m \) – mass, \( c \) – damping coefficient, \( k \) – stiffness coefficient, \( p_1, p_2, p_3 \) – end point of suspension, wheel rim and wheel contact with the road, \( X, Y, Z \) – principal axes of the coordinate system, \( r_d \) – dynamical radius, \( l_k \) – offset of the running wheel axle from the steering axis of the head tube, \( w_{p_o} \) – tyre profile height, \( a_1 \) – actual overtaking distance

This model has five degrees of freedom, so that the operation of the suspension and its torsional compliance are taken into account. Despite the simplifying assumptions discussed in [16], it represents the steering dynamics well and is sufficient to demonstrate the method of analysis presented in this paper. The procedure described in the following section allows the use of any other model, as the procedure will not change. It is easy to see the usefulness of the method described hereafter, especially at the initial design stage, where estimates and preliminary calculations of the structure are carried out. As it was written earlier, it can also be applied to other types of vehicles, as only the mathematical model itself is changed, which is only the first step in the procedure.

Despite the use of a relatively simple physical model, the resulting system of equations has a complicated form, which is mainly due to the inclusion of the head angle of the frame. However, it reproduces the real system sufficiently well, as confirmed by the graph below (see Figure 2).

Figure 2. Comparison of the frequency characteristics of the angular acceleration of the contact wheel \( \dot{\psi}_1 \) experiment and simulation model [15]
Ten different transmittances can be written for such a system [17], but four of them are important, namely: the transmittance for the vertical displacements of the masses $m_1$ and $m_2$ under vertical excitation with force $F_z(G_1(s) = \frac{Z_1(s)}{F_z(s)})$, and the transmittance for angular displacements of the masses $m_1$ and $m_2$ under torsional excitation $M_z(G_3(s) = \frac{\Psi_1(s)\eta_1(s)}{M_z(s)})$, $G_4(s) = \frac{\Psi_2(s)\eta_2(s)}{M_z(s)})$. These transmittances can be obtained by following the steps listed below and are as follows:
1. Development of the mathematical model;
2. Linearisation of the equations describing the system dynamics;
3. Write the equations in operator form and determine the transmittances;
4. Perform stability analysis of the system;
5. Decomposition of transmittance into summation form;
6. Frequency analysis of each member of transmittance.

On the other hand, analyses based on the transmittances obtained in this way were performed using the LabVIEW software and an application developed with its assistance for the purposes of this paper. It is worth noting that, in contrast to dual-mass models describing automotive suspension, the operator transmittances obtained here are of higher order. It should be noted, however, that in contrast to the above-mentioned models, the model used here has a feature that has a key effect on its dynamics - the angle of inclination of the head tube.

The decomposition of the transmittance into simple fractions is a relatively simple task in the case where one-time calculations are necessary. However, in the case of more extensive analyses, it is necessary to repeat the same calculations several times, which at the same time prompts the use of appropriate tools to facilitate the calculation process. However, it is rare for there to be a ready-made tool with advanced capabilities dedicated to very specific tasks. Although the analytical determination of the summation form of the operator transmittance does not cause major difficulties, it is not so with performing such calculations in a software manner and a number of difficulties can be encountered, as mentioned in [15].

3. Simulation results

Using the developed program, the Bode diagrams given below were obtained together with the significant values read from the diagrams, which were tabulated. In addition to the characteristic values for the graphs in question (frequency, maximum amplitude value, natural vibration period, relative damping coefficient), the result of the Hurwitz stability analysis for the highest sub-designator is also included. The most important logarithmic amplitude and phase diagrams are presented here, as there would be additionally 18 diagrams analogous to the ones presented below. However, it was decided to omit them, because of the lack of changes of the analysed values at the change of the selected parameter, and also some duplication of diagrams would occur due to the applied decomposition into simple fractions.

The presented graphs concern the transmittance describing vertical and torsional vibrations occurring on the steering wheel ($G_2(s) = \frac{Z_2(s)}{F_z(s)}$) and $G_4(s) = \frac{\Psi_2(s)\eta_2(s)}{M_z(s)})$, which arise at the handlebars of the motorbike and from the road wheel and are transmitted to them. The focus was on handlebar vibrations because, although these vibrations affect the unsprung mass, their negative effects can be reduced by introducing additional damping directly on the handlebar (the aforementioned torsional vibration damper). The excitation signal for both torsional and vertical vibrations was a force pulse mapped by the Dirac function $\delta(t)$.

It can be seen from fig. 3, tab. 1 that the torsional damping coefficient in the suspension, $C_{sz}$, that it relates to high frequency steering wheel resonances. This also indicates that the rings used, in addition to their sealing function, also damp vibrations. Which at the same time means that their replacement may not only be necessary in the case of visible oil leaks, but also when the elastomer from which they are made begins to lose its original properties. Therefore, the solution proposed in the work [18], compensates for insufficient damping of sealing rings with an increase in the mass of a motorbike, and in this case, it may be enough to design them properly, without deterioration of other properties. It is also worth noting that the value of the $C_{sz}$ parameter in the context of modelling the dynamics of a motorbike, apart from the vibration damping mentioned, does not affect the stability of the steering
system and if the research does not focus on the vibrations of the system, this value does not need to be precisely determined.

It is important to note that the value of the damping factor affects both low and high frequency vibrations. However, for low frequencies these vibrations are completely damped, while for higher frequencies they are only slightly damped. This is also manifested by the value of the coefficient of the amplification constant, which changes the value of the logarithmic modulus of the transmittance, which influences the damping of the system.
Of course, in addition to the sheer flexibility of the O-rings, the stiffness of the steering column and which suspension design of the motorbike is considered will also be important. Some benefits of inverted suspension are also apparent, as due to the increased stiffness of the steering, the effect of constant amplification is reduced and the resonant vibration frequency of the handlebars is increased, which is good for the rider's health.

4. Summary and conclusions
Motorbike dynamics is a complex issue both in terms of research and mastery of safe driving skills by the driver. The former is evidenced by the still small number of publications compared to car research, and the latter by road accident statistics. To a large extent, the effects of gyroscopic moments and the countersteering technique, as well as the vibrations generated in the structure of the motorbike, make it difficult for novice drivers to drive a motorbike. These can sometimes be caused by interactions transmitted from the road to the other components of the motorbike, while in other cases they can be self-excited. There are also more types of vibration that are specific to motorcycles (e.g. high-side, chatter). In addition, a number of factors can make it difficult to stabilise the motorbike after their onset, and torsional vibration dampers are not fitted to every motorbike just as steering assist systems are rare. This paper presents the results of analysis for selected model parameters using the method of analysis of motorbike steering vibrations with the use of a tool developed with the help of LabVIEW.

It is important to note that some of the analysed parameters are influenced by the user himself through the way of driving the motorbike or during periodic replacement of components. While the use of a torsional vibration damper only has a positive effect on the behaviour of the steering system, the selection of damping settings of telescopic shock absorbers requires experience as it is associated with low frequency vibrations. However, for the values of the coefficients of longitudinal elasticity, similar results were obtained as in the work [5], namely their insignificant influence on the amplitudes and frequencies of vibrations. In the case of sealing rings, it can be seen that they play an important role in damping high-frequency vibrations, and therefore it is worthwhile to determine to what extent their elastic-damping properties deteriorate even before replacement dictated by loss of shock absorber oil due to leakage. It is also important to note that there may be a change in vibration frequency over time, as well as the natural damping capacity resulting from the design.

In future work, it is planned to carry out analyses in relation to weight distribution on the vehicle and steering geometry, as well as studies to determine the extent to which suspension performance deteriorates over time.

5. Literature
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