

DEFORMATION OF END CROSSBEAMS OF STEEL BRIDGES RELATED TO SUSTAINABILITY OF BRIDGE JOINTS

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Both AASHTO and Eurocode stipulate that the deflection of end crossbeams of bridges must be limited to 5 mm under the influence of frequent traffic loads. After a thorough literature review and survey of data from manufacturers of bridge joints, it must be concluded that the background of this requirement is not really based on facts. Durability of bridge joints certainly does not appear to be the basis. Another valid reason can be the comfort of road user, which is effectively disturbed by driving over bumps and thresholds. Although the road user expects light shocks at the end of a bridge, the acceleration inside the vehicle must be limited. Therefore, the state of a wheel driving over a threshold has been studied, by applying analytical equations taken from literature and extended. These allow a fairly accurate simulation of the state. Force required to cross the threshold produces an acceleration of wheel, which is transmitted to vehicle's sprung chassis. In the latter one finds the acceleration to which persons are subjected. Several values of comfort can be considered. It turns out that 5 mm limit is very conservative, and one could easily allow up to double the end cross beam's deflection.

Keywords: Vehicle hitting kerb, Comfort acceleration, Parametric analysis, Frequent load combination, Bridge skewness.

1 INTRODUCTION

During the design of several steel tied arch bridges across a main fluvial artery in the North of Belgium, as reported by Dumortier and de Ville de Goyet (2021), an unexpected issue arose concerning the stiffness of the end cross beams. A requirement in Eurocode EN 1993-2 (CEN 2011), which normally does not cause any problem, seemed to have become critical. This specification, based on AASHTO (2020) design rules, requires the vertical deflection of an end crossbeam caused by frequent traffic loading to be limited to 5 mm. In most cases this does not lead to increase the stiffness, yet it did in the present case. A practical solution was implemented. However, extended literature survey as Crocetti and Edlund (2003), Connor and Dexter (1999), and Gao *et al.* (2022) did not reveal the underlying cause for the rather strict requirement. After discussion with bridge joint suppliers, sufficient evidence could be produced, demonstrating that dynamic vertical deflection of 20 mm does not cause any damage to the various types of joints.

Apart from wearing the joints, the restriction on the deflection can be caused by several other phenomena. An evident reason might be that the traffic impact factor on the load would become too high. However, the impact factor for structural safety applies mainly to ULS, since it appears in the AASHTO rules under this type of heading. This leads to the belief that the deformation limit of 5 mm refers to comfort conditions for road users. Indeed, if a large deflection should appear on



a bridge end, the effect is to cause a shock to the wheel tires of the vehicle, which resembles to driving in potholes of the road pavement or a kerb.

Various approaches can be considered for simulating the collision of a moving vehicle with a vertical obstacle or kerb. Some types of software are able to simulate collision of deformable bodies and allow almost exact results. Yet, an analytical approach requires only minor simplifications and renders results, which are close to the output from large effort FE-simulation as is clearly demonstrated by von Chappuis (2012).

2 MECHANISM HYPOTHESES

The deflection of the end crossbeam causes the existence of a threshold or kerb, as the concrete abutment is assumed to be almost undeformable. The crossing vehicle, causing this deformation, will have to rise to the top of the kerb as depicted in Fig. 1. This rise requires to increase its potential energy. The work needed for this energy is delivered by a vertical force, which combines with a horizontal collision force. A first hypothesis is that the wheel tire behaves as a shell, with no bending stiffness, and thus the resulting force has a tangent orientation to the tire belt. This hypothesis is substantiated by the fact that the reaction to forces on the tire belt can only be delivered by the air pressure of the tire. The latter will become indented but does not result in bending. The validation thus neglects variation of friction of the tire belt on the road pavement.

Another related hypothesis is concerned with the inertia force in the tire caused by its deformation can be neglected as compared to the inertia force of the whole system and its un-sprung mass. Also experiments demonstrate that the tire pressure does not change noticeably when the radial deformation takes place. Some further concepts are clarified by Fig. 2, representing the main dimensions of a normal tire.



Fig. 1. Indent of wheel tire while crossing a kerb.



Fig. 2. Dimensions and parts of vehicle tire.

3 ANALYTICAL MODEL

The analytical model is derived by von Chappuis (2012) and is based on the previous assumptions and identifies the kerb to a vertical blade, which is all together a more aggressive condition. All equations will not be mentioned here. First, the circumferential belt force F_b is determined for a wheel traveling on a flat surface. With the symbols indicated in Fig. 2, the belt force can be determined with Eq. (1):

$$F_b = p(R_b w_b - (R_b + R_m)(R_s - h_{sa}))$$
(1)

As soon as the belt hits the blade, the belt will be folded inwards. So, the force will decrease. Therefore, a corrected value of the belt force must be determined with Eq. (2).



$$F_{bcorr} = F_b \frac{2R_b - h_{bl}}{2R_b} \tag{2}$$

The equilibrium of the forces leads to $H = F_{bcorr}$. The resultant force F_r can be found from Eq. (3).

$$F_r = \sqrt{H^2 + V^2} \tag{3}$$

One of the assumptions was that the wheel tire cannot resist a bending moment and that the balance of cords can be applied for this part. From this follows the Eq. (4), which gives a relation between V and H.

$$V = H \frac{4h_{bl}}{L_{eff}} \tag{4}$$

With the Eqs. (1) to (4) it is thus possible to determine the resulting oblique force F_r resulting from the impact of a blade of height h_{bl} . This force is represented in the graph of Fig. 3 for the case of an ordinary tire of a passenger car with $R_b = 0.371$ m, $R_r = 0.241$ m $w_b = 0.235$ m, $h_{sa} = 0.021$ m and p = 2 bar (200 hPa).



Fig. 3. Resulting force as a function of blade height.

If $h_{bl} = 0.10$ m, which would correspond to a normal kerb, then the resulting force is already more than 8 kN, which is very high.

The value obtained is a force, which does not suffice to describe the effect. As mentioned in Section 1, the interesting quantity is actually the acceleration experienced in the vehicle, as this quantity determines the comfort conditions inside the vehicle. This can be found by dividing the force by the mass of the vehicle. One then finds the vertical acceleration at the level of the wheel tire. The whole of the wheel, the wheel suspension and the upper part of the vehicle can be described in a simplified manner by the classical Eq. (5) of a simple mass-spring system.

$$m_{w}\frac{d^{2}y(t)}{dt^{2}} + c\frac{dy(t)}{dt} + ky(t) = F_{r}$$
(5)

The characteristics k and c can be found for most passenger cars from (Mantille *et al.* 2019). They are summarized in Table 1. In the following the multi-link suspension system is considered and the tire is of the type P235/55-R19. The solution of Eq. (5) was found using the conditions that y(t=0) = 0 and the conditions that the upward movement of the wheel reaches a maximum at the time t_{vmax} the highest point of the blade will have been reached and so $y(t_{vmax}) = h_{bl}$. The time t_{vmax}



corresponds to a velocity dy/dt = 0 and an acceleration $d^2/y/dt^2 = 0$. The latter means that the vertical velocity and acceleration are zero at the highest point of the blade.

Table 1. Spring and damper coefficients for various suspension systems.

Parameter	Multi-link	Double wishbone	Leaf spring		
k (N/m)	130000	150000	160000		
c (N s /m)	9800	15000	1000-		

After solving the Eq. (5) the damping force is found as $F_d = c v(t)$. The force in the vehicle is then equal to $F_{int} = F_r - F_d$. After division by the mass, the vertical acceleration in the vehicle can therefore be determined. The latter is represented in the graph of Fig. 4.



Fig. 4. Resulting force as a function of blade height.

4 PARAMETRIC ANALYSIS

4.1 Trucks

In the foregoing, a heavy passenger car of the Audi Q5 type has been considered. It is important to study the effect for other vehicles. A representative truck could be the DAF XG truck from 2023. The tire type is now 385/55-R22.5 and has an air pressure of 9 bar. The characteristics are summarized in Table 2 and are in mm and kg mass, N/m and Ns/m.

Parameter	Rb	Rr	Rm	Wb	hs	hsa	Rs	mw	k	c
Value	498	286	392	385	212	36	174	8786	333156	7587

The wheel suspension is again with leaf spring but much stiffer than for a passenger car, but the damping is also more than 7 times greater. It can be seen that for the same height of the blade, the acceleration in the vehicle is much smaller. Such heavy vehicles are therefore much less affected by the effect of an irregularity in the road surface.

4.2 Tire Parameters

The influence of other parameters on the acceleration a_{int} is simulated by varying one numerical value at a time (Fig. 5). For example, the acceleration drops very quickly with increasing values



of R_b . As soon as this radius exceeds 0.38 m, the acceleration is even zero, because the entire force on the wheel is converted into deformation of the tire.

If both radii R_b and R_r vary simultaneously according to the ratio $R_r / R_b = 0.65$, which is also realistic, then the acceleration in the vehicle increases slightly to reach a maximum value at $R_b = 0.30$ m. This range is practically irrelevant. Afterwards, the acceleration decreases to again render an acceleration = 0 at $R_b = 0.39$ m and the effect of the impact of a blade is completely converted into deformation of the tire.



Fig. 5. Acceleration inside truck as a function of blade height.

The tire pressure p is also an important parameter. It appears that as long as the tire pressure is less than 1.8 bar (180 hPa) no additional acceleration is generated in the vehicle. Once again, the impact force is completely converted into deformation of the tire. Above this limit the acceleration increases almost linearly and the relation can be found from Eq. (6).

$$a_{int} = 5.5496(p - 1.8) \tag{6}$$

In Eq. (6) p is expressed in bar and a in m/s². A similar Eq. (7) has been found for the w_b parameter.

$$a_{int} = 85.106(w_b - 0.215) \tag{7}$$

Furthermore, an increase in acceleration with the spring stiffness of the suspension is observed and an almost imperceptible increase with the damping. The latter seems illogical, but it is due to the decrease in damping force.

4.3 Bridge Skewness

The skewness is an important factor for the superstructure of bridges. Skew bridge decks ensure that the bridge joint is not hit simultaneously by two wheels of the same axle. Of course, there is a difference whether only one wheel hits the joint or both wheels hit the joint simultaneously, albeit with a certain phase shift. In case both wheels are at the same height of the joint and therefore only a small skewness angle exists, the 'blue' curve of the graph of Fig. 6 is found. In case only one wheel can occur on the joint at a time, the 'red' curve of the same graph is found.

This clearly shows that for bridges with oblique cruising, driving into the joint will cause a smaller acceleration and the situation is therefore more favorable.





Fig. 6. Effect of bridge skewness.

5 CONCLUSION

The allowable values of the acceleration to which persons may be exposed are known for buildings. They depend on the frequency of the vibration and are generally severe. The conditions for passengers of a road vehicle are not the same, since no one is surprised that irregularities in the road surface occur, such as bridge joints for example. The criteria for buildings are therefore not usable.

By analogy with the limit values permitted for high-speed rail traffic according to Table A 2.9 of EN 1990 A1 (CEN 2006), an acceleration of 2 m/s^2 is acceptable, 1.3 m/s² is good and 1 m/s² is excellent. For the case of a heavy passenger car with tire pressure of 2 bar and a fairly stiff suspension, this corresponds to deflections of the end cross member of a bridge of 12.5 mm, 9 mm and 7 mm, respectively. For the truck under investigation, the limits may be even higher, given the more complex suspension system.

The conclusion is therefore that the limit values specified by AASHTO and Eurocode are perhaps on the safe side. Further research and experiments might result in more optimistic limitations of the acceptable end crossbeam stiffness.

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