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Contents

Abstract

The report investigates consequences of modified track spacing on pressure waves induced by meeting high-speed trains. The aim is to investigate consequences in terms of an increased risk of fatigue of car body components and which potential counter-effects may have.

The study is intended as a first investigation that should be valid under fairly generic conditions. Simplified analyses are employed due to the generic conditions considered. Under more specified conditions, refined analyses can be employed to improve the analysis and improve quantification of the various mitigating actions.

1 Aim, approach and limitations

The problem investigated can be summarised as:

> The width of the railway corridor has a massive influence on the cost of a new railway. Space between the tracks has recently been increased at least partly due to awareness that shockwaves between meeting trains may cause fatigue of chassis bodies. However, increased track spacing may drastically increase the cost high-speed network. Proper risk assessment should therefore be a predecessor to any decision on track spacing.

The current report is an outline of such a risk assessment. The influence of the track spacing on the risk of chassis fatigue is carried out in subsequent steps:

1. Evaluate how load magnitudes increase with decreasing track spacing
2. Evaluate how the stress levels in the chassis are influenced by the increased load magnitude
3. Evaluate how the increased stress levels affect the predicted fatigue life

In all steps, it is evaluated how an increased fatigue loading can be compensated for by counter-measures. From a risk analysis point of view this corresponds to avoiding any increase in risk levels as compared to the original configuration.[1] Another approach would be to evaluate absolute risk levels and compare them to acceptable risk levels. This would however require more pre-knowledge and limit the analysis to predefined conditions. Neither is suitable for this first, generic study.

Only the case of pressure loads in open track is considered. This excludes pressure loads at tunnel entrances etc.[2] Note however that this only relates to the pressure load magnitudes – discussions on stress levels (chapter 3), influence on fatigue life (chapter 4) and potential mitigating actions (chapter 5) are still valid.

Finally, any other effects of pressure waves, such as its influence on the vehicle dynamics, are not considered in the current study.

2 Influence of track spacing on pressure loads

The considered load that is is the pressure wave from meeting trains (or stemming from a train passing a building next to the track etc.). The general characteristics of such a pressure load on open track[3] is discussed in the European code EN 14067-4:2013 [1]. In

[1] In the current context taken as current Swedish conditions.

[2] In general, reduction of pressure waves in tunnels is expensive. Consequently, tunnels are likely to induce the highest pressure loads. This does however not mean that these loads are decisive for the fatigue life of the vehicles. The reason is that pressure loads in open track may be more frequent and have different characteristics (i.e. induce more load cycles). Whether this is the case will depend on the local conditions of the network considered.

[3] Pressure loads in tunnels are discussed in EN 14067-3.

the standard (and in the following), the considered load corresponds to train-induced pressure loads acting on flat vertical structures parallel to the track. This does not explicitly include a meeting train,[4] but for comparative purposes it is considered to be sufficient.

The characteristic value of the distributed load due to the air pressure is

$$p_{1\mathrm{k}} = \frac{\rho v_{\mathrm{tr}}^2}{2} k_1 c_{\mathrm{p1}} \tag{1}$$

Here ρ is the air density (kg/m^3), v_{tr} is the train speed [m/s], k_1 is a shape function (0.6 for high-speed trains and 0.85 for regular passenger trains). The aerodynamic coefficient c_{p1} can be evaluated as

$$c_{\mathrm{p1}} = \frac{2.5}{(Y+0.25)^2} + 0.02 \quad \forall Y \geq 2.3\,\mathrm{m} \tag{2}$$

with Y being the distance from the centre of the track [m].

In figures 1 and 2, $p_{1\mathrm{k}}$ has been evaluated with reference values $\rho = 1$, $v_{\mathrm{tr}} = 167$ ($= 600$ km/h), $k_1 = 0.6$ and $Y = 4.5$. To emphasise the influence of the meeting trains, the speed has been taken as double a (considered) operational speed. This should overestimate the effect quite radically since it ignores the aerodynamical shape of the train body as compared to e.g. a noise barrier.

New Swedish lines are currently built for 250 km/h. Presuming that these trains correspond to $k_1 = 0.85$ and high-speed trains to $k_1 = 0.6$ gives a decrease in $p_{1\mathrm{k}}$ of 70%. An increase in speed from 500 km/h (139 m/s) to 600 km/h (167 m/s) increases $p_{1\mathrm{k}}$ by 44%. In combination, we get a 2% increase in $p_{1\mathrm{k}}$ magnitudes.[5]

Note that the above analysis presumes that both trains that meet are aerodynamically beneficial (i.e. $k_1 = 0.6$). This does not hold e.g. for the case of a high-speed train meeting a freight train.

Consider that ρ and k_1 are constant. A change in track spacing from Y to $\bar{Y}$ is then equivalent (in the sense that magnitudes of $p_{1\mathrm{k}}$ remain the same) as a change to train speed $\bar{v}_{\mathrm{tr}}$ given from

$$\bar{v}_{\mathrm{tr}} = v_{\mathrm{tr}} \sqrt{\frac{\frac{2.5}{(\bar{Y}+0.25)^2} + 0.02}{\frac{2.5}{(Y+0.25)^2} + 0.02}} \tag{3}$$

The relationship in equation 3 is graphically presented as $\bar{v}_{\mathrm{tr}}/v_{\mathrm{tr}}$ as function of $\bar{Y}/Y$ for reference values of Y =3.5, 4.5 and 5.5 meters. As seen, a doubling of the track spacing is equivalent to a decrease of the speed to some 59–65% depending on the reference track spacing.

Since the calculations above feature comparisons under generic conditions, they are fairly crude. For specified conditions, detailed numerical simulations can be performed to obtain a much better estimation of actual pressure levels and their time evolutions.

[4] More in detail, the reference case of a flat vertical structure does not consider for example the influence of the more rounded shape of the train car and the opening below the floor.

[5] $139^2 \cdot 0.85/2 = 8\,211$ versus $167^2 \cdot 0.6/2 = 8\,367$ gives $\left(167^2/139^2\right) \cdot (0.6/0.85) = 1.44 \cdot 0.7 = 1.02$.

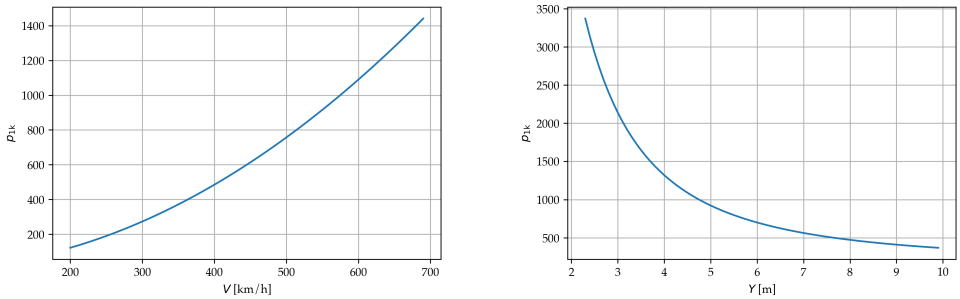

Figure 1: Influence of vehicle speed (left) and track centre distance (right) on characteristic pressure wave magnitude.

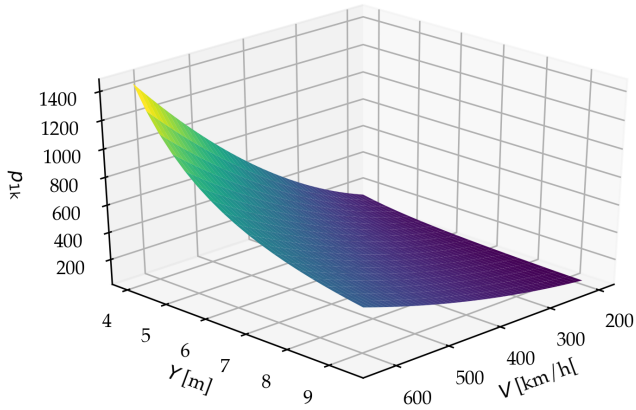

Figure 2: Influence of vehicle speed (left) and track centre distance (right) on characteristic pressure wave magnitude.

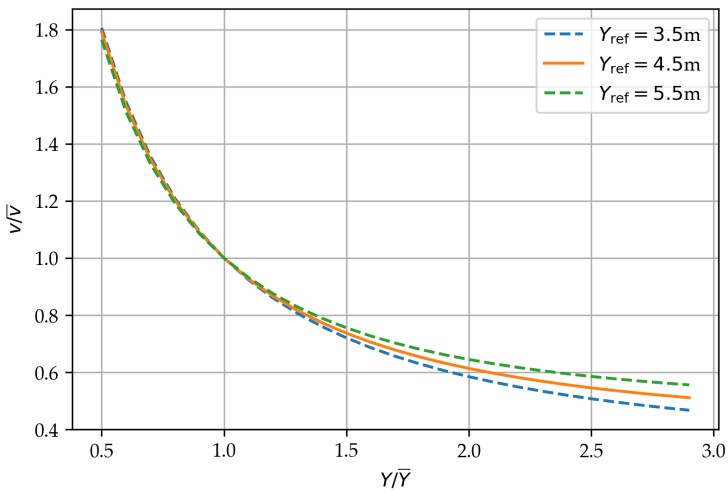

Figure 3: $\bar{v}_{\mathrm{tr}}/v_{\mathrm{tr}}$ as function of $\bar{Y}/Y$.

3 Stress levels due to increased pressure loads

When two trains meet, the pressure load will act on the side panels of the trains. This load will cause stresses in the panel in itself and in supporting framework.

3.1 Stress levels in wall panels

To estimate the influence on stress in the panels, we can study the simple case of simply supported rectangular plate.[6] In the centre of the plate, the bending moments will increase as

$$M = \alpha p b^2 \tag{4}$$

Where α is a coefficient that depends on the ratio between length and width of the plate[7] and b is the width of the plate.

This means that a change in contact pressure from p to $\bar{p}$ can be compensated for by a modified plate width $\bar{b}$ as

$$\bar{b} = b\sqrt{\frac{p}{\bar{p}}} \tag{5}$$

In addition, for a constant plate thickness, the stress level is

$$\sigma = \beta \frac{M}{h^2} \tag{6}$$

[6] In reality, the plate is continuous and extends over several spans. Further, the supports are not rigid as is implicitly presumed. General conclusions are however still valid.

[7] It will also differ depending on which direction of the bending moment that is studied.

where h is the height of the plate and β a constant. Consequently, an increase in bending moment from M to $\bar{M}$ can be compensated for by adopting a modified plate height $\bar{h}$ as

$$\bar{h} = h\sqrt{\frac{\bar{M}}{M}} \tag{7}$$

Note that equations 6 and 7 are valid for small deformations.[8] Also note that an increased plate thickness will increase the weight of the car body and decrease available passenger space.

A stress concentration (e.g. a window) will further increase stress levels locally. The *increase* in stress levels only depends on the geometry (i.e. not the applied loading).[9] An increased nominal load due to an increased pressure load can thus be compensated for by decreasing the stress concentration. How large decrease that can be obtained depends on the existing configuration and how much modification that can be accepted from a design point of view, cf. section 5.1.

3.2 Stress levels in support beams

Consider a structure where the wall panel is supported by evenly distributed vertical beams. The load[10] on the beam will be dependent on the beam spacing, c, as

$$q = p \cdot c \tag{8}$$

A change in pressure from p to $\bar{p}$ can thus be compensated for by changing the beam spacing to $\bar{c}$ as

$$\bar{c} = \frac{p}{\bar{p}}c \tag{9}$$

The bending moment in the beam is given by

$$M = \gamma q L^2 = \gamma p c L^2 \tag{10}$$

where L is the length of the beam and γ is a constant. Thus, a change in pressure from p to $\bar{p}$ can (given a fixed c) be compensated for by changing the beam length to $\bar{L}$ as

$$\bar{L} = \sqrt{\frac{p}{\bar{p}}}L \tag{11}$$

The stress level in the beam will be given by

$$\sigma = \frac{M}{I} z_{\max} \tag{12}$$

Here the area moment of inertia I and the distance from the centroid to the surface $z_{\max}$ will depend on the geometry of the beam. In general, an increased pressure load can be compensated for by a modified beam geometry. This may influence weight and passenger space.

[8] More in detail, they presume Kirchoff–Love plate theory to be valid.
[9] If the response is elastic, which is a reasonable assumption.
[10] Expressed as the load intensity, i.e. load per unit length.

4 Influence on fatigue life of increased pressure loads

4.1 Reductions in plain fatigue life

The discussion in section 3 investigates how an increased contact loading can be compensated for by changes in wall geometry. If this is not possible, the fatigue life will be reduced once the stress amplitude exceeds the fatigue limit. In general, the fatigue life for stresses above the fatigue limit can be estimated by the relation

$$N = \frac{C}{\Delta\sigma^m} \tag{13}$$

Here C is a constant, $\Delta\sigma$ is the stress range (twice the stress amplitude) and m a material parameter (typically 3 or larger). The decrease in fatigue life can thus be significant even for a moderate increase in stress level. The total life corresponding to a number of cycles of varying amplitude can be estimated as

$$\frac{1}{N} = \sum_i \frac{1}{N_i} \tag{14}$$

It should be noted that pressure waves correspond to a minority of the total number of load cycles. However if the decrease in life for these load cycles is large, the reduction in fatigue life may still be significant. It is however not straightforward to state exactly how large the reduction would be – equation (13) is only valid for stress amplitudes above the fatigue limit. For stress levels below the fatigue limit, the induced damage is much lower than implied by equation (13).[11]

4.2 Fatigue of connections

Connections between beams in the supporting framework are important from a fatigue point of view. The loading of the connections will depend on the nominal load and the detailed configuration of the connection. The fatigue life of connections can be estimated from equation 13 although it is then commonly reformulated as

$$N = 2 \cdot 10^6 \left(\frac{\mathrm{FAT}}{\gamma \cdot \Delta S} \right)^m \tag{15}$$

Here FAT is the fatigue class, ΔS the nominal stress, m a fatigue exponent (commonly taken as $m = 3$ for stresses above the fatigue limit), and γ is a safety coefficient. An increased ΔS due to an increased pressure loading can thus be compensated for by increasing FAT. This can be done by modifying the connection and/or through improvement procedures, see section 5.2.

[11] The designation 'fatigue limit' might imply that these load cycles would not induce any damage. This is however not the caee for variable amplitude loading with some load cycles exceeding the fatigue limit although the slope (defined by m) is much more shallow.

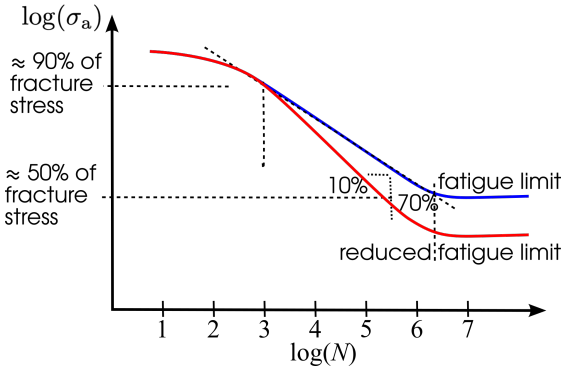

Figure 4: Fatigue–life (SN or Wöhler) curve.

5 Actions to enhance fatigue life

In general fatigue mitigation actions aim to decrease the stress level. The influence can be seen from figure 4. For ideal (laboratory) conditions, an increase in stress amplitude by 10% from the fatigue limit reduces the fatigue live by 70%. Under real conditions, the effect is usually less dramatic since the fatigue limit is reduced e.g. due to surface roughness. In figure 4 this is indicated by the reduced fatigue life curve in red.

The aim with fatigue enhancing actions is either to decrease the fatigue load (e.g. lower σ_a) or increase the fatigue resistance. The latter typically relates to minimising the fatigue limit reduction, replace the material with a material that has a higher (reduced) fatigue limit, or to employ fatigue treatment by inducing a compressive residual stress in the surface material.

5.1 Design actions

As discussed in section 3.1, the influence of the increased pressure load can be counteracted by modifying the global and/or the local geometry. In summary, these actions can be described as

- stiffer car panels with shorter wall sections and shorter supporting beams,
- decreased nominal stress levels e.g. through thicker wall plates,
- reduced stress concentrations through larger notch radii at window corners, connections etc.

The amount of compensation that can be obtained depends mainly on the current situation – a less optimised structure is easier to enhance. An idea of the possible stress reductions that can be achieved by just reducing stress concentrations (i.e. modifying the local geometry in highly stressed areas) is obtained from figure 5, which shows a notched plate.[12] If the current notch is fairly sharp (say $r/d = 0.05$), relatively small

[12] A window in a train chassis would be more like a single notch, but the character of the stress concentration would be similar.

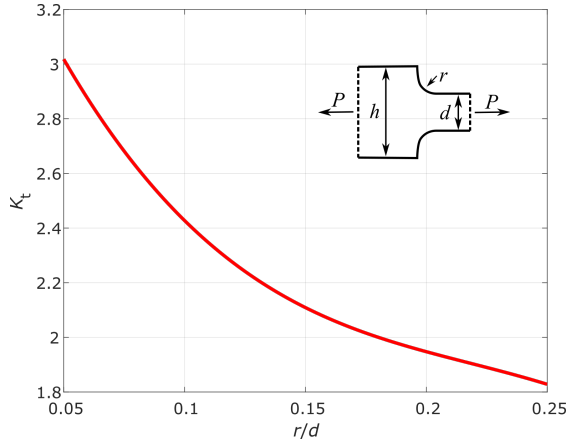

Figure 5: Stress concentration factor as function of r/d for a notched plate with $h/d = 2$. Data from [5].

adjustments (reduction to $r/d = 0.18$) would be required to decrease the stress concentration factor (and thereby the maximum stress at the notch) by 50%. For more rounded notches, the effect of further rounding is less.

How much reduction in stress concentrations that can be achieved also depends on what is acceptable from a design point of view. Here we can take an airplane fuselage as an extreme example where the body is subjected to a major over-pressure in the plane in addition to very high wind loads.[13] To avoid fatigue in the fuselage, window openings on planes are much smaller and rounder than on trains. In this context, a core question is of course how large design modifications – e.g. decreased window sizes – train passengers would accept.

5.2 Fatigue strength enhancement

Given a certain geometry, the fatigue resistance can be improved through a range of actions. The first is to reduce surface roughness. Depending on the unmodified state and the material quality, this may have fairly large effects. An idea of potential gains can be obtained from figure 6.

If decreased surface roughness is not sufficient, surface treatment methods such as cold working and shot peening can be employed. These are generally relatively expensive, but can increase the fatigue limit (and thereby the fatigue life) considerably, see e.g. [4]. The methods are however not without caveats: They act by inducing a compressive residual stress field, which means that the material will be plastically deformed and thereby may loose tolerances. Further, there will be tensile residual stresses below the compressed surface layer. This may promote fatigue crack initiation in these areas and/or promote the growth of a crack once it is initiated.

[13] This of course differs from the pressure load that a train car is subjected to. However, the general argument is still valid.

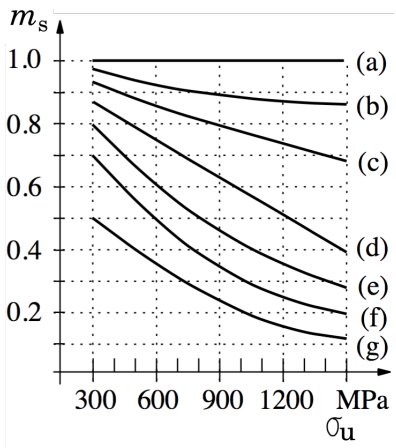

Figure 6: Influence on surface roughness on the fatigue limit. Here σ_u is the ultimate (fracture) stress and the reduced fatigue limit is obtained as $\sigma_{e,red} = m_s \sigma_e$ where σ_e is the nominal fatigue limit. The curves represent surface conditions as (a) polished, (b) ground, (c) machined, (d) standard notch, (e) forged, (f) corroded in tap water and (g) corroded in salt water. Adapted from [2].

For welds, shot peening and cold working are options although they may be complicated due to the geometry. Additional fatigue enhancement methods for welds are grinding, TIG dressing, and over-stressing. The fatigue improvement is mainly limited to weld toe cracks. To appreciate the potential of fatigue improvement of welds, the fatigue class (FAT) for load carrying fillet welds[14] are considered to have been increased by 12% when grinding or TIG dressing has been employed, see [3]. The corresponding increase in fatigue life can be evaluated from equation 15.

5.3 Operational actions

Actions for limiting the risk of fatigue initiation includes controlling load and environment. This includes corrosion prevention, and monitoring/controlling strain levels in susceptible components.

In addition to monitoring of operational loads, inspections can also be employed to identify damage. Here, damaged components can either be repaired/removed, or identified cracks can be continuously monitored and removed before they reach critical sizes corresponding to lack of structural integrity. This defect tolerant approach is currently employed e.g. for aerospace structures and allows for lighter constructions where stress levels can be allowed to exceed the (reduced) fatigue limit.

[14]Evaluated using a hot-spot method

6 Concluding remarks

This overview report outlines consequences of altered track spacing on the pressure loads on wall panels. It shows how increased load magnitudes influence the fatigue life. Potential counter-actions including altered design and fatigue improvement actions are described. In this manner, the risk level can be kept constant even though load magnitudes increase.

The conclusion of the report is that from a technical perspective increased load levels due to higher speeds at current track spacing can be allowed while maintaining current risk levels regarding chassis fatigue. The main issue is the cost of the required counter-actions in relation to the cost of an increased track spacing.[15]

In this context it should be noted that both the analysis methods that have been used and the presumptions that have been made are crude. This was necessary since the analysis should be generic in the sense that nothing is presumed regarding the train design. In an operational analysis where more is known about the train design, numerical simulations can be employed to obtain a better estimation of pressure loads and resulting stress levels. Further, knowledge of the detailed design will improve the fatigue analysis and the quantification of benefits from fatigue improvement actions. This, together with knowledge of the track construction cost would provide ample input for a cost-benefit analysis that can decide suitable track spacing for the case under consideration.

Acknowledgements

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[15] 'Cost' is here used in a general sense that includes also factors such as decreased passenger comfort.

[5] Walter D. Pilkey, Deborah F. Pilkey, and Rudolph Earl Peterson. *Peterson's stress concentration factors*, volume Third edition. Wiley, 2008.