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BERGEN BYBANE

NOISE REDUCTION BY TRACK DESIGN

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Introduction

A city line now is build from the areas in south to the city center of Bergen. Part I of the project goes from Nesttun to the city center. In the future the line is probably extended to the airport.

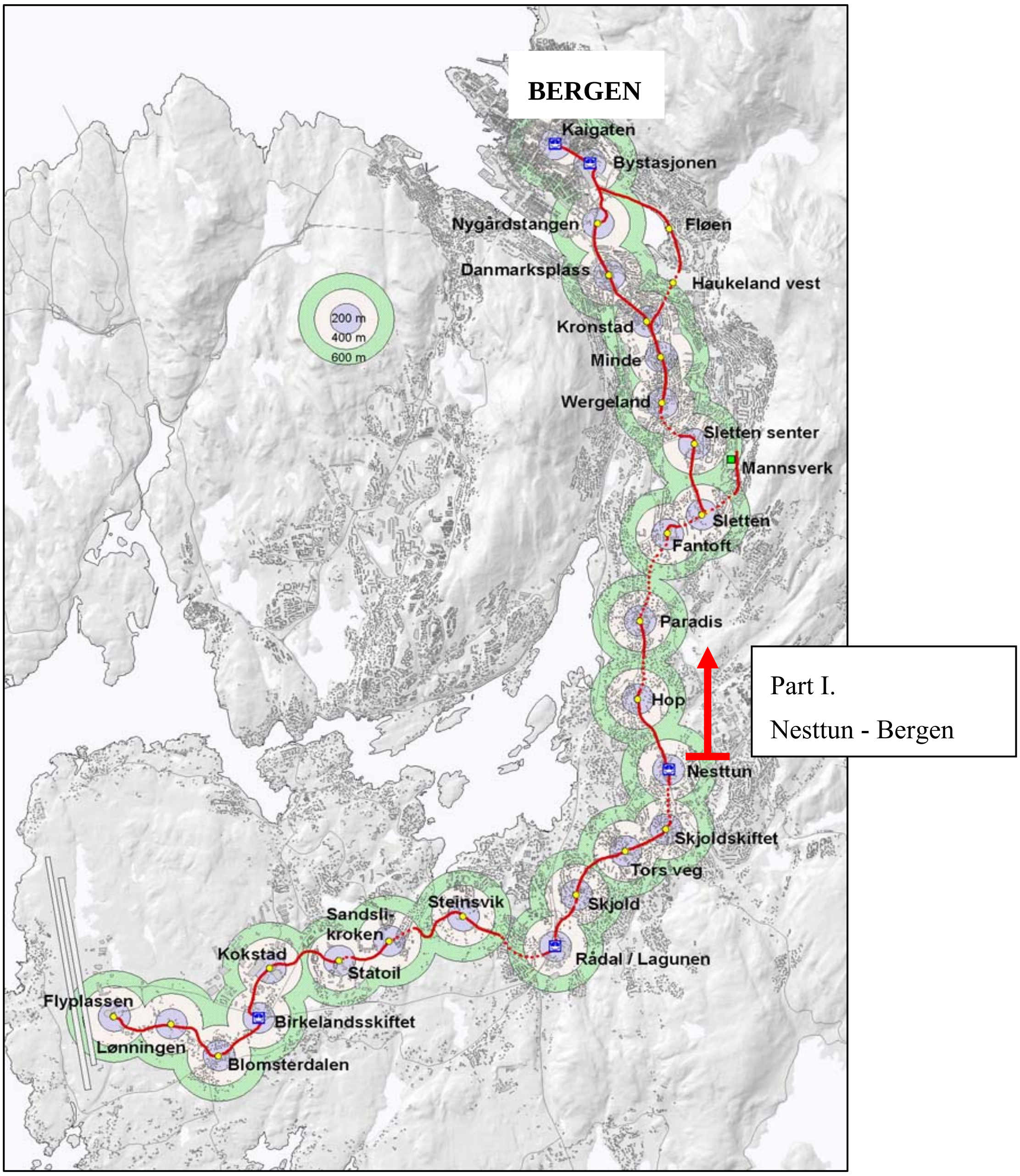


Fig 1. The planned Bergen city line. The construction work on part I now has started.

The limit for ground borne noise transmission is $L_{A,maks}$ = 37 dB in rooms in which the ground borne noise is the dominating part of the noise from the trams. The track form is ballasted track in parts far away from the city center and embedded rails on a concrete slab in the city streets. For cases in which high reduction of ground borne noise is required elastic mats will be laid out below and on the sides of the concrete slab.

This paper concerns the embedded rail system.

1 Fundamentals of embedded rail systems

A section through the concrete slab and the embedded rail system is shown in fig 2.

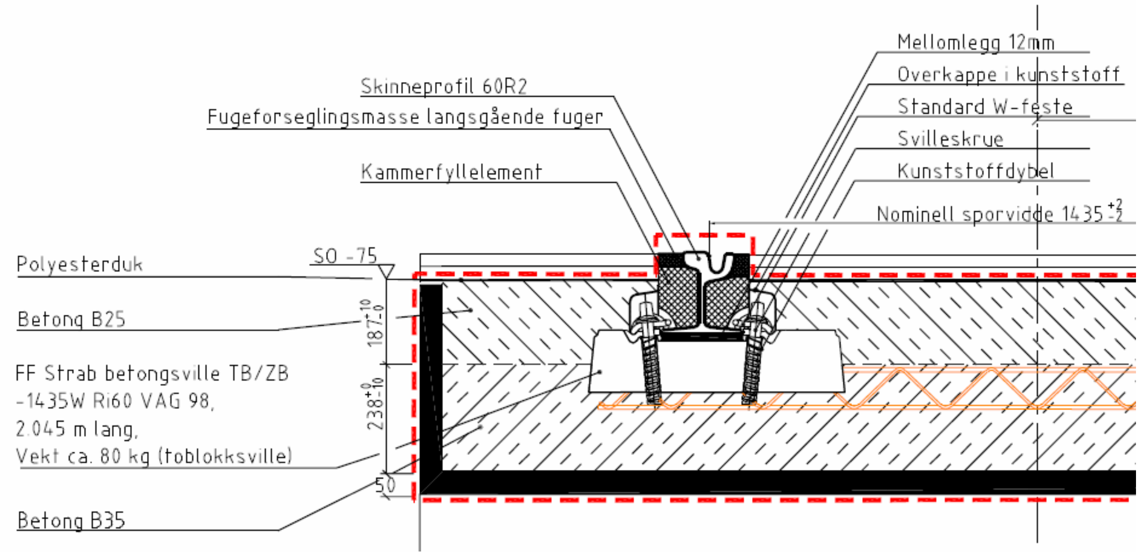


Fig 2. Section through the track construction in the city streets. The embedded rail system is the fastening of the rail in the concrete slab. The drawing is from the tender document. Other system could be suggested from the tenderer.

The fundamental property of the embedded rail system is to insure electric isolation between the rail and the slab. In addition the stiffness of the rail support may influence on several factors as:

- Ground borne noise transmission
- Airborne noise transmission
- Growth of rail roughness
- Passenger and driver ride comfort

There are different kinds of embedded rail systems. For instance the system can be based on dry mounting of the system, or on pouring of liquid masses in slots in the slab. However from a theoretical point of view the embedded systems are alike. The parameters for calculations of acoustical characteristics and deflections are:

- k_{dyn} . Dynamic stiffness for frequencies 2 – 5 Hz of elastic support, which is vertical stiffness of material below and shear stiffness of the material on the sides of the rail. Use for rail deflection calculations.
- k_{acou} . Acoustic stiffness = dynamic stiffness for frequencies 25 – 500 Hz of elastic support, including materials below and on the sides of the rail. Use for ground borne noise reduction calculations.
- η. Loss factor of the support and embedding material. Use for ground borne noise reduction calculations.

This paper gives calculations of the ground borne noise reduction from embedded rail systems and recommendations for dynamic properties of the embedded system. The influence on the airborne noise from the track is discussed as well.

2 Ground borne and airborne noise level from the track when the rail is caste directly to the slab

For the slab track cases the structure borne noise calculations in the project have mainly been based on data from measurements in Oslo. The track form in Oslo is a 300 mm slab on gravel or mineral wool. The rail is set in place on the slab having a 10 mm elastic continuous rail pad. However a new layer of concrete then is caste on the slab, and after that two layers of asphalt. There is no isolation between the rail and the concrete / asphalt, the rail therefore is stiff fastened to the slab and cannot be moved.

The ground borne noise levels in dwellings along the tram line are calculated as maximum A-weighted noise levels from an empirical calculation method based on dBA values. The required noise reduction is the difference between the calculated dBA value and the noise limit. In order to calculate the noise reduction from the embedded rail system it is necessary to specify the ground borne noise spectrum. Measurements of ground borne noise in different buildings near to tram lines in Oslo shows some variations in frequency content. However the maximum values usually are in the frequency region 80 - 100 Hz. The dimensioning spectrum shape which is used for the calculations of the ground borne noise reduction is shown in fig 3. The spectrum is normalized to a total ground borne noise level of 37 dBA.

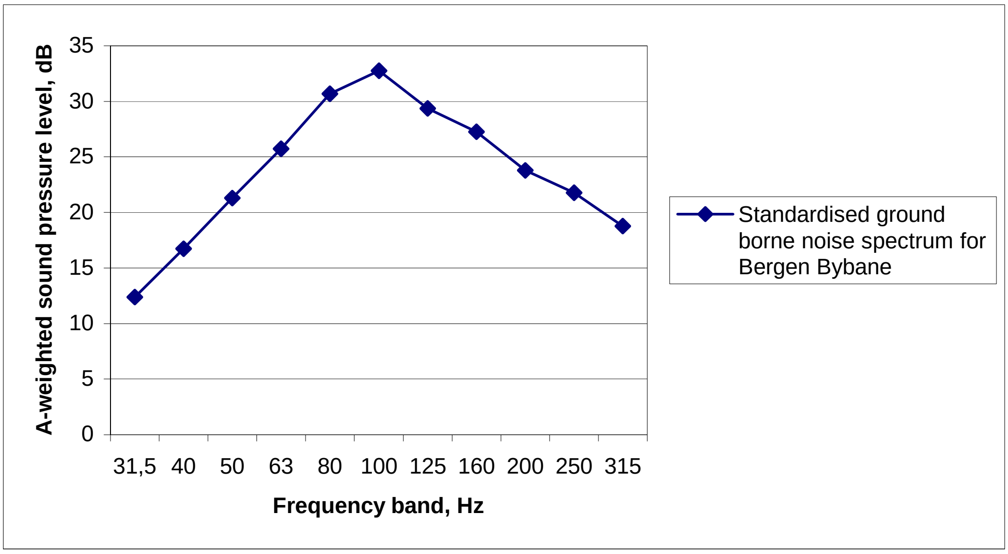


Fig 3. Spectrum shape for the ground borne noise level when the rail is stiff fastened to the concrete slab.

The spectrum in fig 3 is typical for cases in which high ground borne noise levels are measured.

The airborne noise level from the trams is very dependant on the quality of the rail. In fig 4 is shown typical noise levels from old and new rails. In the old rails there are wave corrugations. It can be seen that the corrugated rail gives an amplification of the noise around 100 Hz. Vibration from the rail is transmitted to the concrete slab, and noise is radiated from the slab.

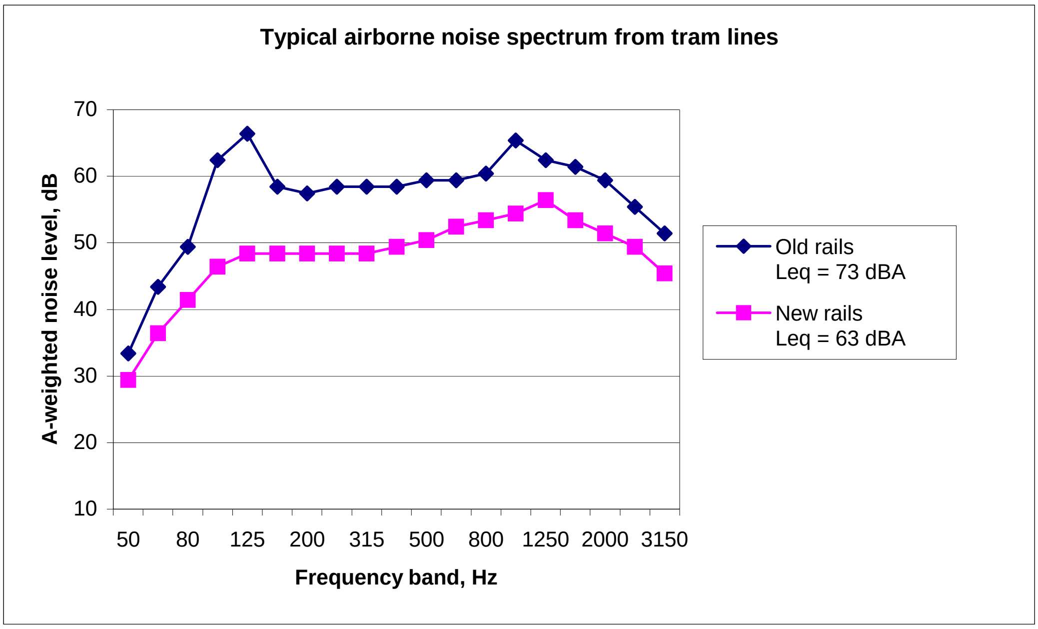


Fig 4. Typical airborne noise spectra from old and new tracks.

The reason for the differences is that there are wave corrugations in the old rails. An example is shown in fig 5.

Fig 5. Example of a severe case of wave corrugations in the rail surface.

When a wheel passes a wave corrugated rail having a corrugation wave length l, vibration is generated in the contact point with a frequency of f = l / v. Typical speed of the trams is v = 30 km/h, and in some cases v = 40 km/h may be reached in the city streets. In fig 6 is shown the wave corrugation frequency as a function of speed and wave length.

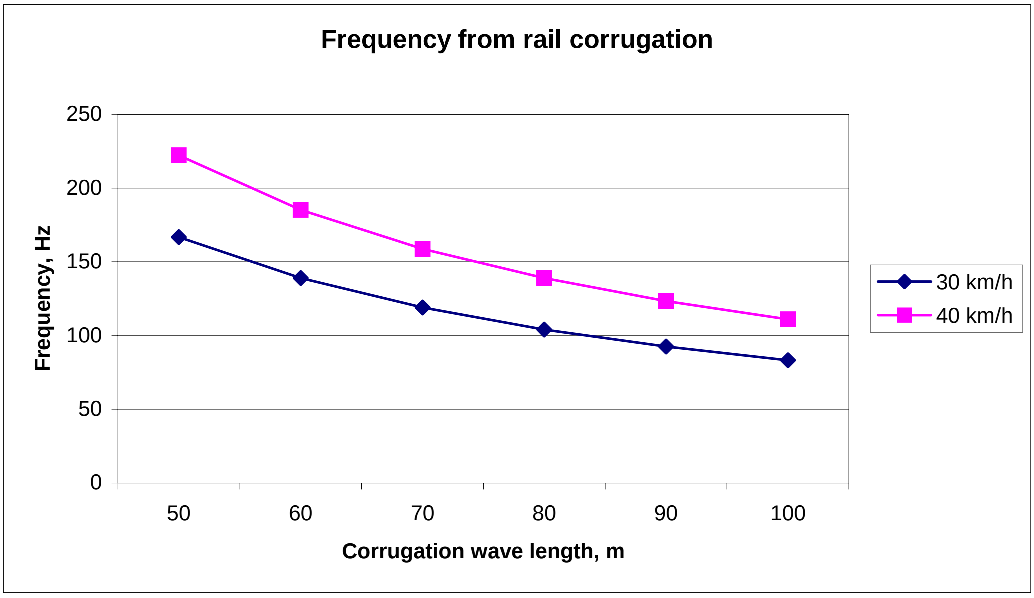


Fig 6. Generated frequency as a function of speed and wave corrugation wave length.

Typical corrugation wave length is 70 - 100 mm. It is seen that for the most usual speed of 30 km/h, the frequency is around 100 Hz.

It therefore is of vital importance that the embedded rail system gives high vibration isolation around 100 Hz. This in order to reduce the ground borne noise transmission and to reduce the airborne noise transmission from the concrete slab.

3 Vibration isolation from embedded rail systems

The embedded rails system will have a resonance frequency, f_0, which is calculated from the mass of wheel plus rail and the dynamic stiffness of the support. Around the resonance frequency the ground borne noise will be amplified, and for frequencies higher than 1.4 f_0, there will be a reduction. The classical theoretical vibration isolation curve fore the simple mass / spring system is shown in fig 7 for resonance frequency $f_0 = 63$ Hz.

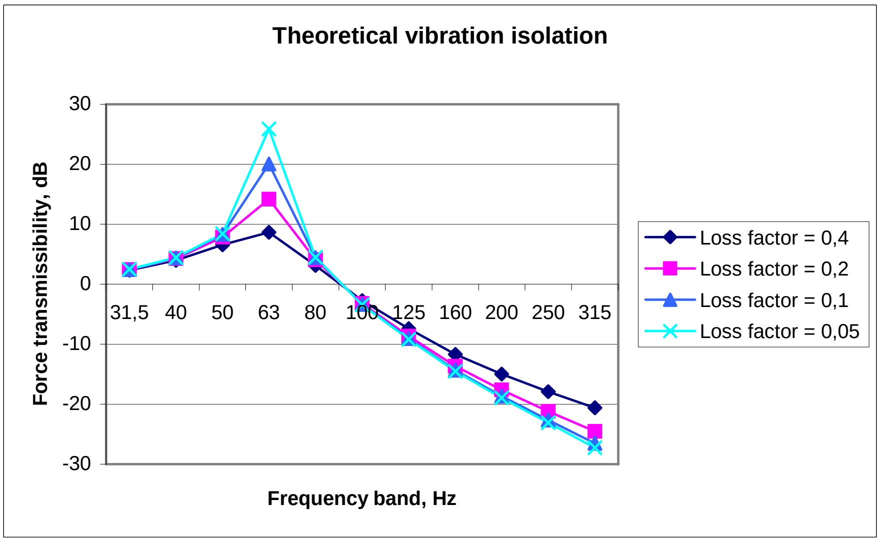


Fig 7. Vibration isolation for a mass / spring system. Different loss factors. Hysteretic damping.

The amplification at resonance is highly dependant on the material damping in the elastic support and the embedding materials. The loss factor is defined as two times the fraction of critical damping, $\eta = 2\,\zeta$.

For the vibration isolation of railway tracks the measured amplification at resonance for remedial actions is smaller than calculated theoretically from material damping in the elastic suppport. This is because the vibration transmission along the rails will contribute to the damping. From experience the amplification at resonance for the embedded rail systems is assumed to be around 5 - 10 dB, and probably nearer to 5 than 10 dB. In the calculations the the maximum amplification is set to 7 dB. For higher frequencies the experience from measurements is that it is difficult to reach higher vibration isolation than 10 – 15 dB. In the calculations the maximum reduction is set to 10 dB. The values for vibration isolation which is used in the calculations is shown in fig 8, for a resonance frequency of f_o = 63 Hz.

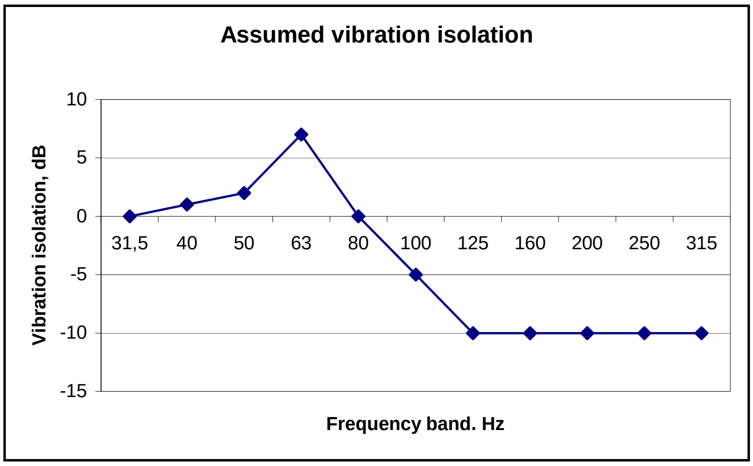


Fig 8. Values for vibration isolation, f_o = 63 Hz.

The curve in the figure is for f_0 = 63 Hz. The amplification at resonance and reduction above is assumed to be equal for all other resonance frequencies.

4 Ground borne noise reduction

The embedded rail system will give amplification at the resonance frequencies and considerable noise reduction at higher frequencies. In fig 9 is shown calculated ground borne noise levels for embedded rail systems having different resonance frequencies. In the calculations is assumed a 7 dB max amplification at resonance and a max reduction of 10 dB for higher frequencies, se fig 8. These are typical values from measurements. However efforts should always be made to optimize the system in order to obtain lower amplification at resonance and higher reduction at higher frequencies

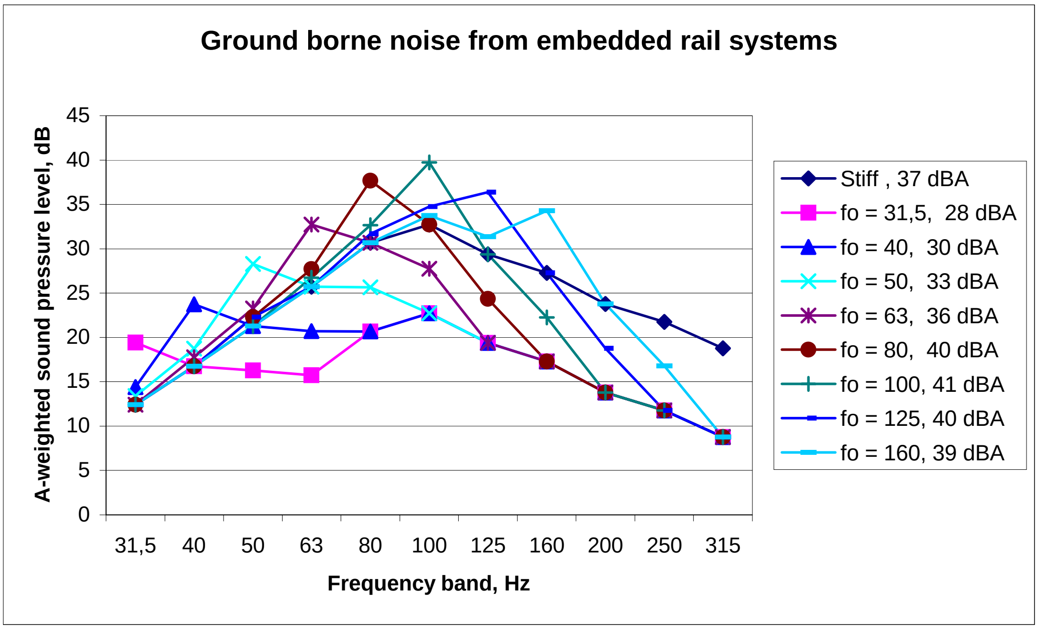


Fig 9. Ground borne noise level for embedded rail system. Different resonance frequencies.

The noise reduction is equal to the difference between the dBA value for the stiff system and for the vibration isolated cases. The main results from fig 9 is given in table I.

Resonance frequency	31,5 Hz	40 Hz	50 Hz	63 Hz	80 Hz	100 Hz	125 Hz
Noise reduction	9 dB	7 dB	4 dB	1 dB	-3 dB	-4 dB	-3 dB

Table I. Noise reduction from embedded rail systems, different resonance frequencies

It is seen that the softest embedded rail system may give considerable noise reduction, however if the stiffness is too high there will be very little noise reduction or even an amplification of the noise. The values in fig. 9 are A-weighted noise levels which mean that it can be seen directly the contribution from each frequency band to the total dBA value. The calculated results show clearly that the amplification at resonance is very important, and it should be reduced as much as possible by using high material damping in the support. The solution on the sides of the rail is important as well and the contribution to the system damping should be emphasized. High damping do reduce the vibration transmission along the rail as well.

5 Resonance frequency as a function of rail deflection

In order to reach high values for ground borne noise reduction the resonance frequency should be as low as possible. For a given tram and rail a low resonance frequency must be obtained by using softest possible rail support. However if the support is too soft the rail deflection will be too high.

The resonance frequency is calculated from the support stiffness below the rail, the rail stiffness and the unsprung mass of the wheel set. This is the mass which is in direct contact with the rail, below the carriage vibration isolation system.

The rail deflection is calculated from the axle load, the support stiffness and the rail stiffness.

The support dynamic stiffness in the frequency region 2-5 Hz is used for the rail deflection calculations and the acoustic stiffness is used for the calculations of the resonance frequency. This is the dynamic stiffness in the frequency region around the resonance frequency. Usually the acoustic stiffness is higher than the dynamic stiffness, and the dynamic stiffness is higher than the static stiffness.

In fig 10 is shown calculated resonance frequencies as a function of rail deflection for the trams in Oslo. The deflection is from a moving tram. The rail deflection for a stationary tram is higher than for a moving tram since the static stiffness is less than the dynamic stiffness. The effect of the dynamic addition to the static axle load is smaller than the effect of dynamic to static stiffness.

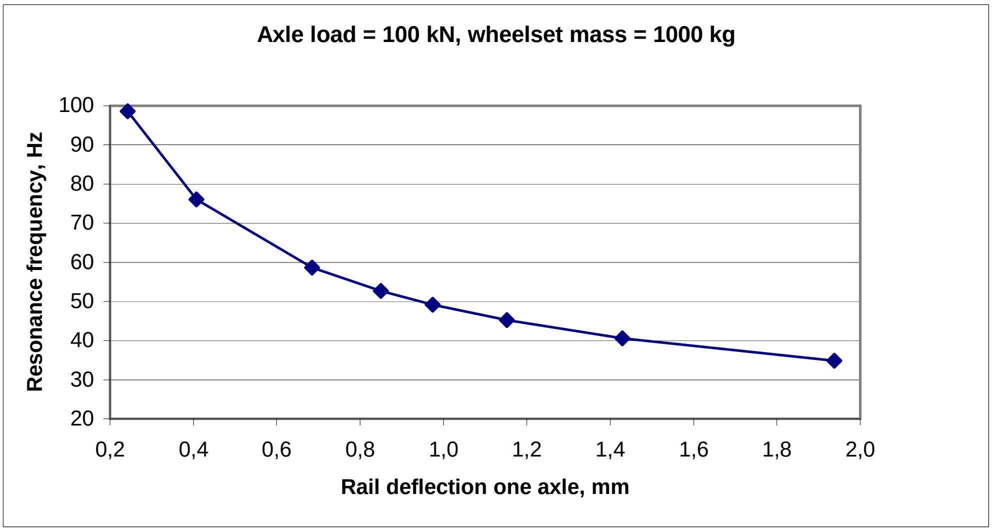


Fig 9. Resonance frequency as a function of rail deflection. Axle load = 100 kN. Ri60N rail. Mass of wheel set = 1000 kg. Relation $k_{acou} / k_{dyn} = 1.2$.

The input values in fig 9 is values for the SL95 tram in Oslo. The rail deflection for the softest embedded rail systems in the market is around 1.0 mm from one axle. The calculated resonance frequency for this case is $f_0 = 48$ Hz, and the calculated ground borne noise reduction is 5 dB.

The most uncertain input value in the calculation is the unsprung mass of the wheel set. In fig 10 is shown the dynamic system for a tram.

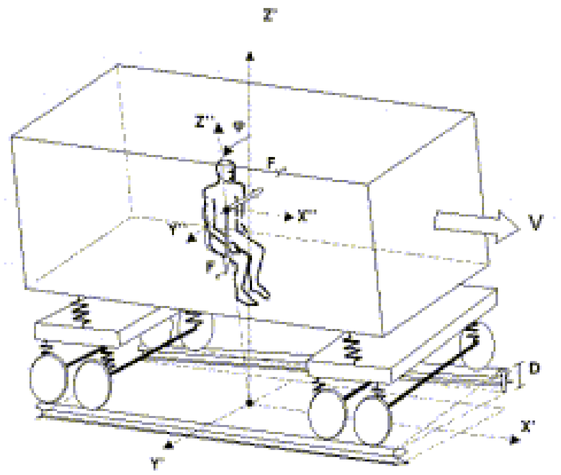


Fig 10. Vibration isolation system for a tram in principle.

The wagon is softly vibration isolated from the bogie, and the wheel set is vibrationisolated from the bogie. In the calculations it is assumed that the wheel set is dynamically uncoupled from the bogie and the wagon in the actual frequency region.

The wheel set mass is the sum of the wheels, the axle, the bearings and the brake disks. However other masses may be included as well for a tram wheel set, for instance the gear box.

In low floor trams there may be free running wheels which is that the wheels are connected directly to the bogie, and there is no axle. The wheel set mass then becomes very low.

In fig 11 is shown calculated resonance frequencies as a function of the wheel set mass.

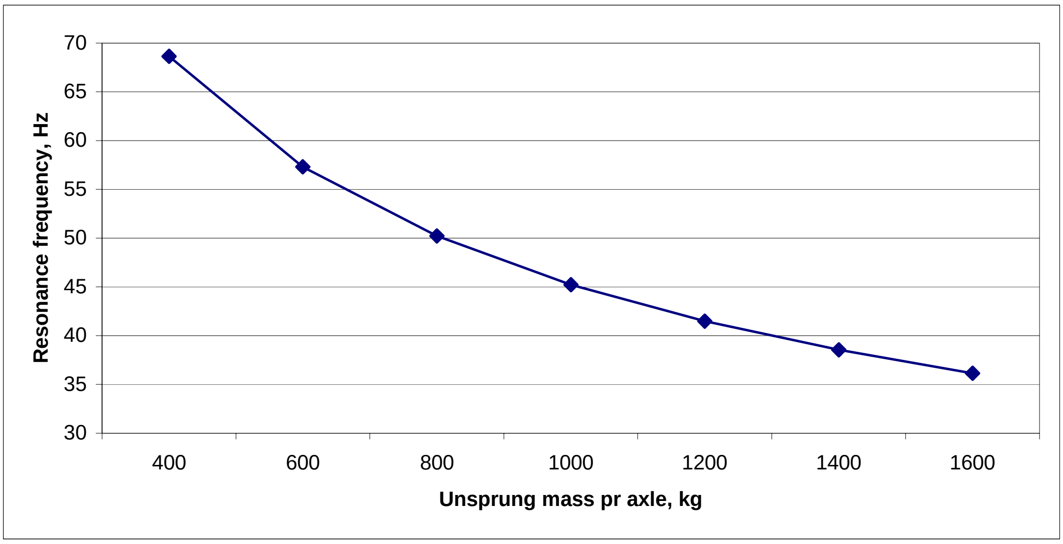
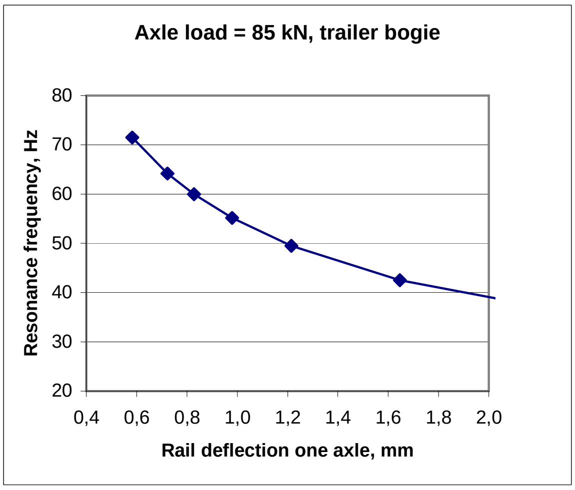


Fig 11. Calculated resonance frequency for an embedded rail system as a function of wheel set mass. Parameters are as in fig 9 and 1 mm rail deflection for one axle.

It is seen that when the wheel set mass is small, the resonance frequency is so high that the embedded rail system will give an amplification of the ground borne noise level. In the Bergen Bybane project the dimensioning axle load is 85 kN which is for a 2/3 loaded tram. There is differences between the wheel set masses for the trailer bogie and the moter bogie. Calculated resonance frequencies is shown in fig 12.



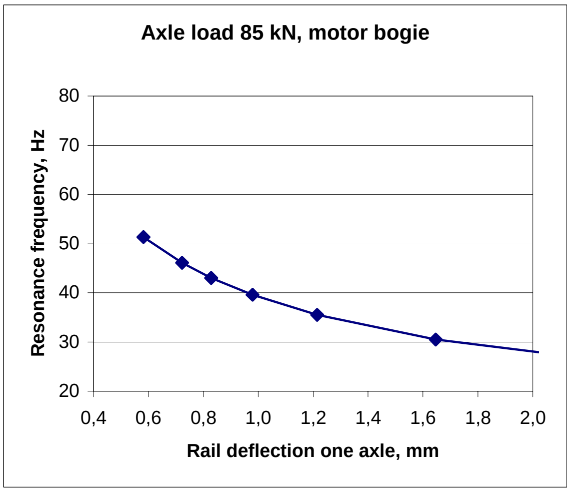


Fig 12. Resonance frequency and rail deflection for the trailer bogie and the motor bogie.

The resonance frequency becomes f_0 = 56 Hz for the trailer bogie. From table I it can be seen that the ground borne noise reduction then will be only around 3 dB. For the motor bogie the resonance frequency is 40 Hz, and the noise reduction is 7 dB.

In tram wheels usually there is an elastic cuff between the wheel rim or tread and the rest of the wheel. Theoretically this elasticity may reduce the effective wheel set mass. If the cuff was very soft, the unsprung mass would be very low, the resonance frequency would increase considerable, and the vibration isolation would be near to zero. However the general experience is that this is not the case, the embedded systems do give noise reduction for all kinds of wheels. In this report it is assumed that the cuff is so stiff that it do not influence on the calculated vibration isolation for the track.

6 Total deflection

Because of small distances between the axles the total deflection becomes higher than the deflection from each axle. In fig 13 is shown the total deflection from the Bergen Bybane tram for the case in which the deflection from one axle is 1 mm. The deflection limit is set to 1.2 mm for 85 kN axle load in the Bergen project. This was found to be the max limit from a maintenance point of view.

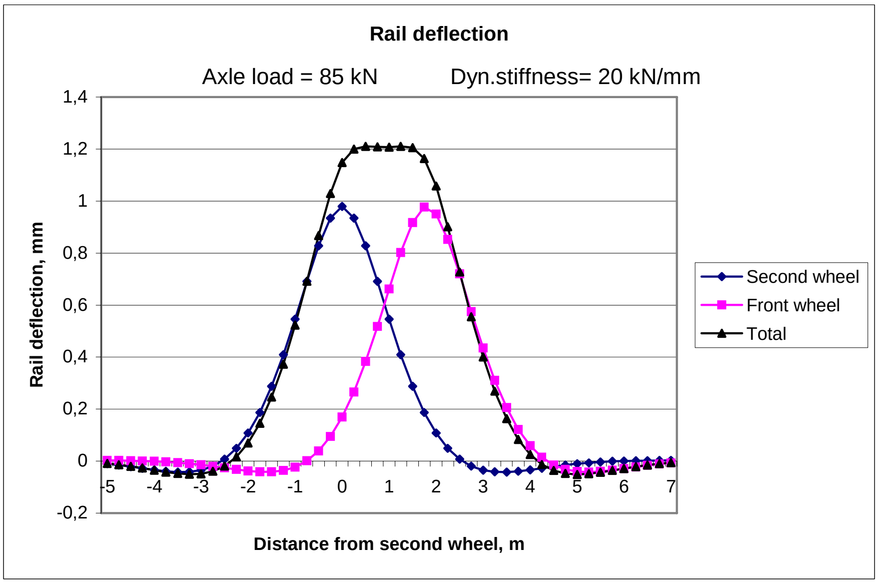


Fig 11. Total deflection of the rails

7 Conclusions

A well designed embedded rail system can give typically around 3 - 7 dB reduction of ground borne noise. Concerning low frequency airborne noise from the track it is important that the system resonance frequency is much lower than f_0 = 100 Hz. A too stiff system may give considerable amplification of the noise.

In order to reach a highest possible noise reduction from the embedded rail system for a given tram the important factors are:

- Resonance frequency f_0 = 45 - 50 Hz or lower
- High material damping in the elastic support material and in the material on the sides of the rail.

A high wheel set mass gives a low resonance frequency and therefore a high noise reduction for a given track.

A system which give low amplification at resonance and high reduction at higher frequencies is preferred. Reduced amplification at resonance is more critical than increased reduction at higher frequencies.