DYNAFREIGHT | Innovative technical solutions for improved train DYNAmics and operation of longer FREIGHt Trains

D2.2 – Noise reduction on freight locomotive bogies

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EXECUTIVE SUMMARY

Railway transportation is generally considered as environmentally friendly. In this context, one of the biggest challenges to deal with is the railway noise, which refers to the acoustical field excited by rolling stock in action and perceived as annoying by passengers and neighbours of the railway.

In Task 2.2 of DYNAFREIGHT project the actions taken have focused on noise reduction of a freight locomotive running gear.

The first step in identification of the main acoustic sources was taken by performing measurements with an acoustic array in order to localize and characterize the main sources. Under different running scenarios and different power modes a locomotive has been analysed.

Two practical solutions to mitigate the noise emission of the running gear have been elaborated:

- The design, manufacture, mounting and assessment of lateral skirts by measurements (integrated in Task 2.5)
- Wheel-set optimization by simulation and lab testing

This document summarizes the methodology and the results of this elaboration.

ABBREVIATIONS AND ACRONYMS

- Lp: Sound pressure level
- Lw: Sound power level
- f: Frequency
- λ: wavelength
- v: vehicle speed
- R(λ): Roughness spectra in wavelength range
- R(f): Roughness spectra in frequency range

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- HUD: University of Huddersfield
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1. INTRODUCTION

This deliverable summarizes the elaboration of Task 2.2. on noise reduction. There will be no specific survey whether the acoustic field may be perceived as annoying and therefore described as noise. In this report, the term noise comprises an acoustical field which might be perceived as annoying or not.

The main objectives are the identification of the noise sources of a six axle (Co'Co') freight locomotive under different running scenarios and the study of two practical solutions to mitigate the noise propagated during pass-by. No braking has been taken into account.

The identification of the main sources has been elaborated in a wide range during pass-by, taking three different speeds of a Co'Co' freight locomotive and two different power modes into account. One practical solution was the wheel-set optimization, which results 1 dB noise emission reduction by using brake discs. The mitigation potential of lateral skirts has been elaborated by measurement. The single value (averaged from 100Hz-10000Hz) of noise reduction of 1dB at 80km/h and 4.2dB at 120km/h.

This task has been done in close cooperation with Task 2.5 Bogie Model Integration and Implementation of DYNAFREIGHT, where the design, manufacturing and installation on the lateral bogie skirt has taken placed, so Task 2.2 has done the measurements with and without bogie lateral skirt in order to evaluate the mitigation potential of this solution.

2. IDENTIFICATION OF THE ACOUSTIC SOURCES IN A FREIGHT LOCOMOTIVE UNDER DIFFERENT RUNNING SCENARIOS

To enable a practical and effective elaboration of noise mitigation measures of a system, its characterization concerning acoustic sources is crucial. This characterization includes the localization of the acoustic sources with respect to the spectra of the propagating sound.

2.1 THE METHODOLOGY TO THE ACOUSTIC SOURCE IDENTIFICATION

In this project a practical method by means of measurements has been chosen focusing acoustic wave propagation during pass by. The examined locomotive is dual powered (electro-diesel) with Co-Co wheel arrangement. Different running scenarios have been taken into account as they affect the acoustic wave propagation. In each power mode pass by measurements at 80km/h, 120km/h and 160km/h have been performed with an acoustic array, as this device offers the best opportunity when it comes to acoustical source localization regarding specific frequencies.

2.2 MEASUREMENTS AT FAUREI TEST TRACK

The above mentioned acoustic measurements have been performed at the Faurei test track in Romania at a fixed measurement area and different distances in order to have reproducible boundary conditions during the whole measurement campaign.

2.2.1 Track analysis

Track decay rate (due to DIN EN 15461) and rail roughness measurements (due to DIN EN 15610) have been performed at the Faurei test track to determine the characteristics of the track in the measurement area. The analysed data shows that the track decay rate exceeds the curves set out in the standard. The rail roughness is higher than that specified in the standard in particular within the mid-range wavelengths. This fact needs to be considered when it comes to assessing the propagating sound as the characteristic of the track affects its dominance for the rail wheel interaction.

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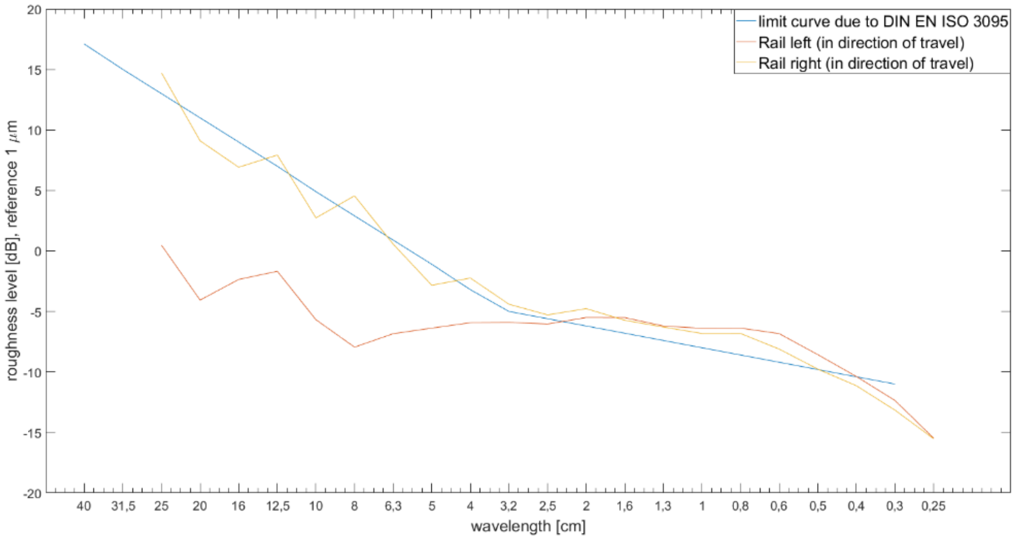

Figure 1: Rail roughness of the test track in FAUREI

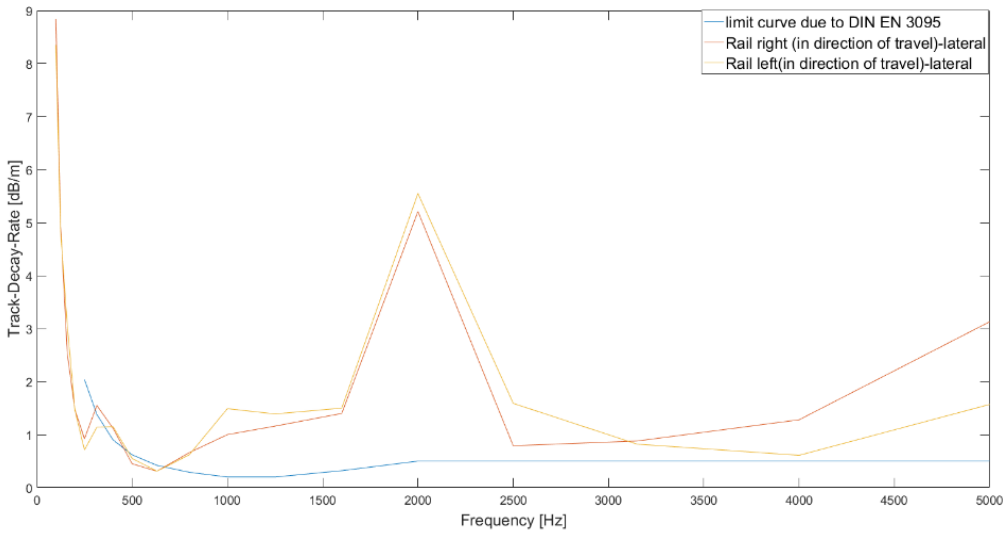

Figure 2: Lateral track decay rate of the test track in FAUREI

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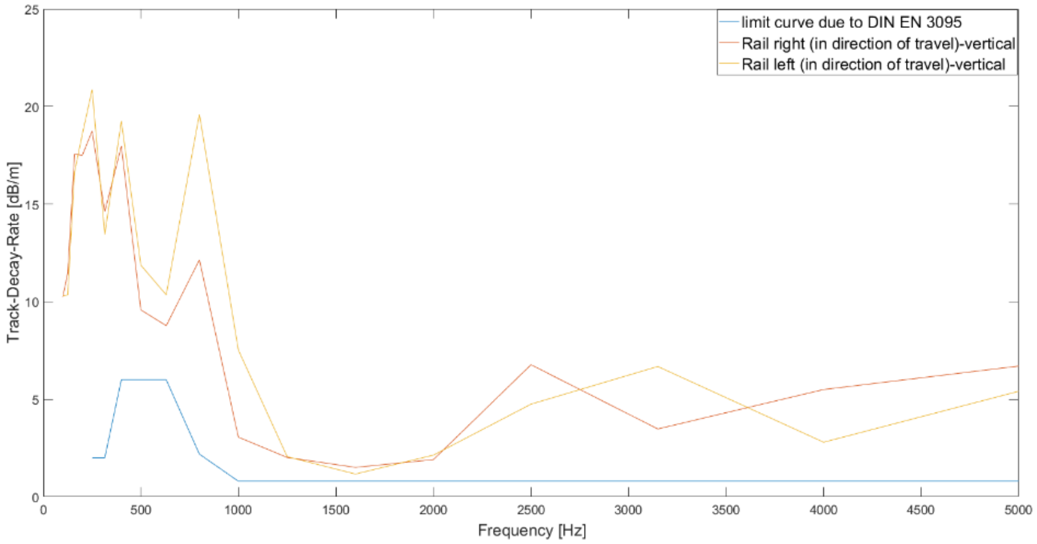

Figure 3: Vertical track decay rate of the test track in FAUREI

2.2.2 Identification of the acoustic sources under different running scenarios

For the identification of the acoustic sources of the examined locomotive measurements with an acoustic array (including 112 microphones and of 1.7m diameter) have been performed. The figures in this chapter show the main sources for different running scenarios.

The dynamic range of the sound pressure level reaches from max (red area) to a minimum (purple area) of 20 dB less than the red area.

One may note that the propagated acoustic field is widely dependent from the energy mode, the analysed frequency and the speed of the locomotive. In any case, the main sources are located in the area of the bogie as well as due to the wheel rail interaction. All analysed scenarios are attached in the appendix of this report. On the basis of these results, mitigation measures have been elaborated, represented in section 3.2 and 3.3 of this document.

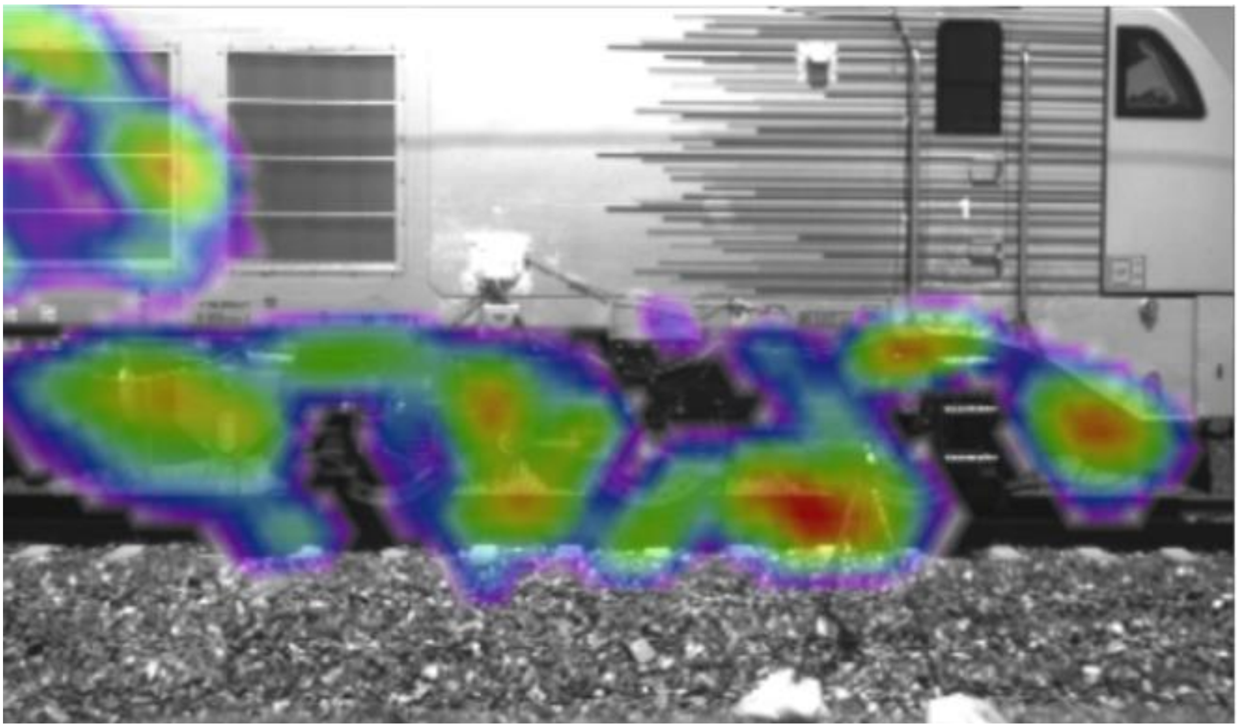

Figure 4: Diesel Mode_80kph-3rdOctave2000Hz

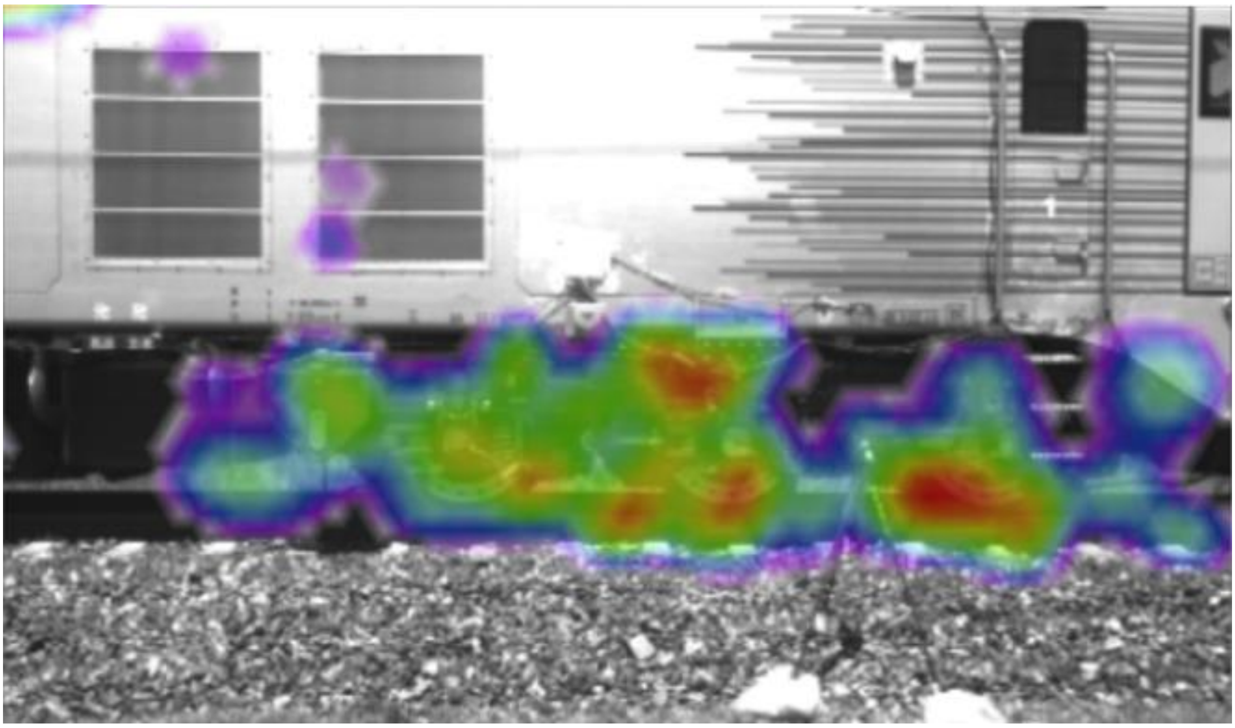

Figure 5: Diesel Mode_120kph_3rdOctave2000Hz

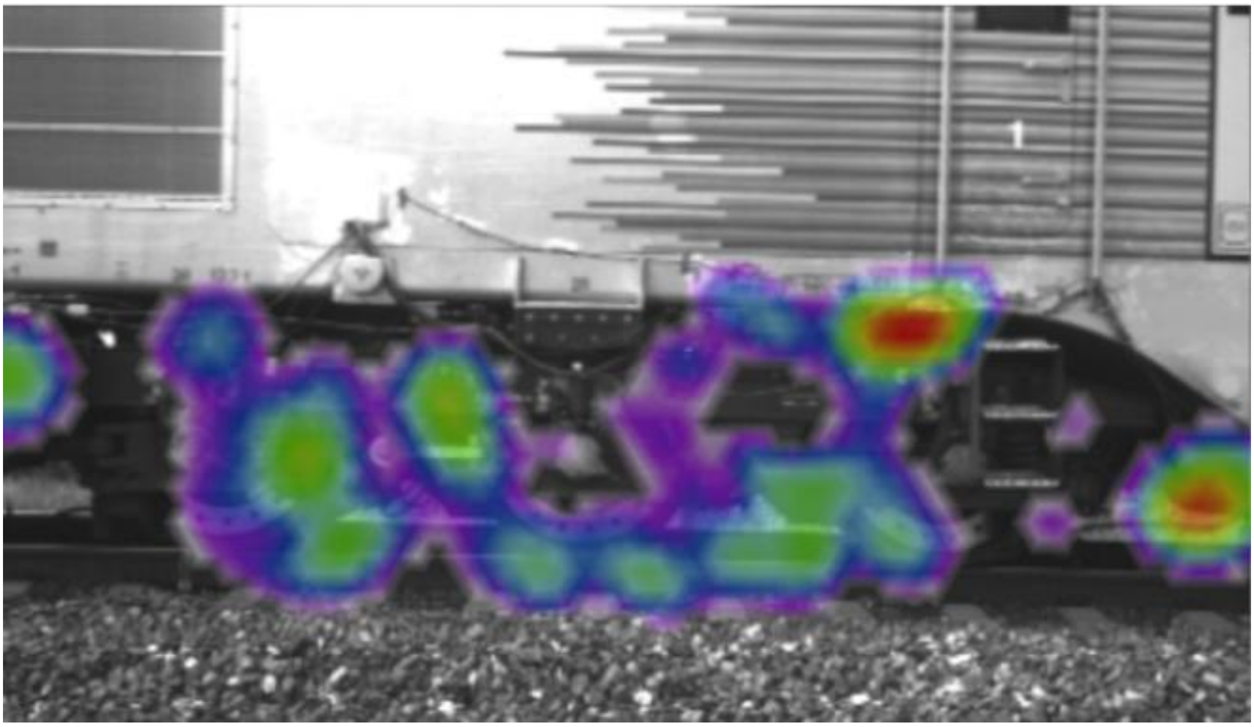

Figure 6: Electric Mode_80kph_3rdOctave2000Hz

3. MITIGATION MEASURES OF RUNNING GEAR RELATED NOISE EMISSIONS

In this chapter an overview of existing methods to mitigate rolling stock related noise emission is given. For the objectives of this project two practical solutions have been chosen and elaborated. The practical potential of lateral skirts regarding noise emission as well as the effect of wheel set optimization are the contents of the further sections in this chapter.

3.1 OVERVIEW ON STATE OF THE ART ANALYSIS OF EXISTING NOISE MITIGATION MEASURES

Nowadays, noise emission and noise pollution is an essential question in rail traffic. The basis for the acoustic design of rolling stock is the TSI Noise (European Railway Technical Specifications for Interoperability), where limit values of standstill, starting and pass-by noise for various types of vehicles are specified. These values are not too strict and the manufacturers could already produce trains with better performance, therefore a renewal of these specifications is foreseen in the future. That may require the use of additional mitigation measures on the vehicles.

Figure 7 shows the evolution of limit values for pass-by noise for different vehicle types.

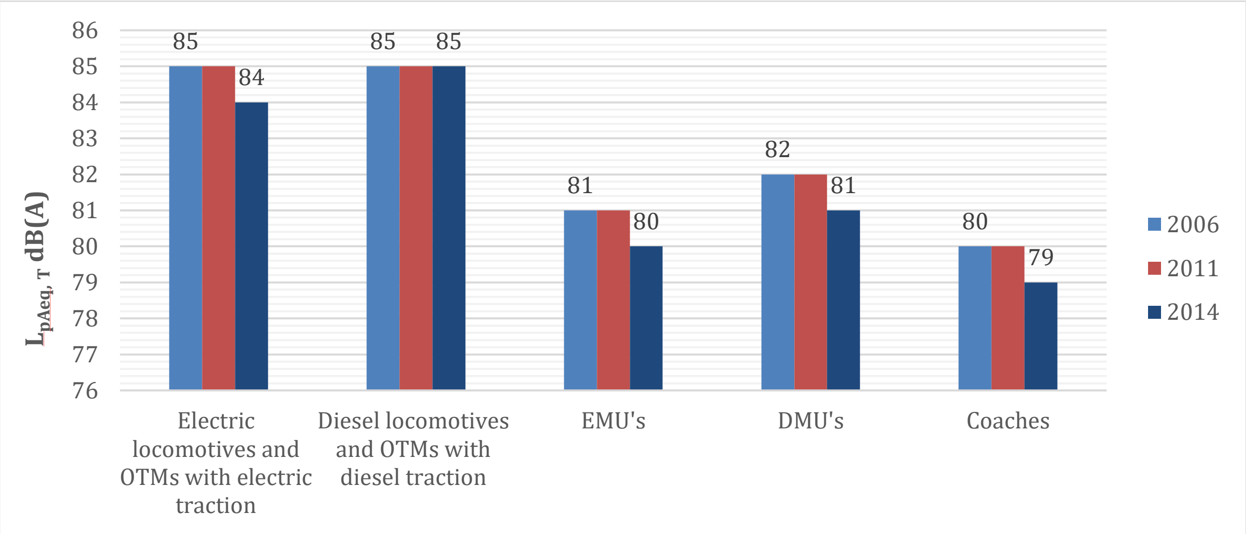

Figure 7 Development of TSI Noise limit values for pass-by noise [1]

The main noise sources of trains with propulsion are:

- Traction noise
- Rolling noise
- Aerodynamic noise

Aerodynamic aspects are playing a more important role for car-body design, especially for higher speeds, starting from around 150 km/h. This task focuses on parameters and measures related to running gear (traction and rolling noise).

Rolling noise is generated in the wheel-rail contact (see figure 8). The combined wheel and rail roughness induces vibrations in the wheel and rail that are radiated and propagate through the air to a receiver position. Moreover the vibration is transmitted to other elements of the running gear and to the carbody. Depending on the vibration amplitude, geometry and size of the vibrating surfaces the sound radiation of these structures may become important. Motors and propulsion systems radiate sound directly and can also transmit vibrations to other elements through the coupling elements.

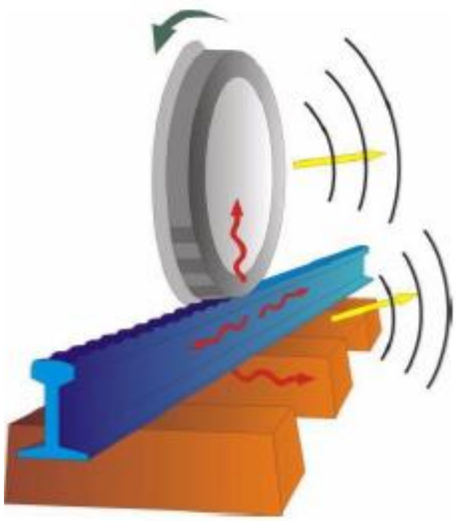

Figure 8 Illustration of the mechanism of generation of rolling noise [2]

There are several opportunities to mitigate running gear related noise emission:

1. **Mitigate the noise generation**

It is the most effective option to mitigate noise at the source, therefore it is crucial to perform an intensive wheel monitoring and maintenance to avoid wheel flats, wheel out-of-roundness and a high wheel roughness. In order to achieve smoother contact surfaces, the use of disc brakes or new types of brake shoes (other than cast iron) is essential. The project STAIRRS [3] showed that the use of new types of brake shoes is the most cost-effective measure. This solution is already widely applied. New freight wagons have disc or K-brakes and old wagons can be retrofitted.

As for traction noise, motors with better acoustic qualities could be designed (less direct sound radiation and less vibration transmission).

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2. Attenuate transmission

Large vibrating surfaces such as the car-body and the bogie frame contribute to noise generation. They are excited through structure-born transmission from strucutre-borne sound sources. Therefore, it is a good solution to isolate these parts from the wheelset and the propulsion system with rubber springs and bushings or for example for secondary suspension with air-springs that can decouple the vibrations of the two elements.

3. Reduce vibrations of different elements

The application of tuned wheel absorbers (see Figure 9), disk brakes (further examined in Section 3.3) and wheel dampers against brake screech are some examples to reduce the vibration amplitude and thus sound radiation. A new and innovative solution is the coating of the inner side of the wheel together with the axle (see Figure 9). The reduction potential of this solution is around 2 dB.Wheel absorbers can bring ca. 2,5 dB reduction under normal operation, but up to 15 dB for curve squeal. [4]

Figure 9 Wheel with tuned absorbers and coated wheelsets [4]

4. Mitigation at the propagation path

There is an option to mitigate sound at the propagation path with barriers. The closer these barriers to the sources are, the more effective their mitigation properties are. On vehicles they can be done with bogie skirts that have an absorption layer on the side of the wheels. In Section 3.2 this option is further investigated.

In order to decide, which measure to use, one should consider all the costs (design, manufacturing, installation, maintenance, etc.) and all the benefits (direct and indirect) of the solution. This project only focuses on direct benefits of the chosen measures.

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3.2 Noise reduction potential of lateral skirts in practice

The lateral skirt has been designed and produced in Task 2.5 *Bogie Model Integration and Implementation* of DYNAFREIGHT on the basis of the work presented in chapter 2.

To assess the mitigation potential of the lateral skirts the same microphone array has been used like the device in use for the noise source identification. The measurements were performed at the test track in the Centre d'essais ferroviaires in France in electric mode and at speeds of 80km/h and 120km/h.

3.2.1 Spectral analysis

A comparison between the noise propagated from the bogie with and without the skirts results an average sound pressure level reduction of 1 dB at 80km/h and 4.2 dB at 120km/h in a frequency range from 100 Hz – 10 kHz.

The table below shows the value of the sound pressure level of a single microphone right at the moment of pass by of the bogie with lateral skirts minus the sound pressure level of the same microphone at the moment of the pass by of the bogie without lateral skirts. As a result of this, negative values show a mitigating effect and positive values show obviously a higher level of propagated sound.l This shows the high frequency dependency of the lateral skirts regarding their noise mitigation potential.

Frequency	Difference at 80 km/h	Difference at 120km/h
100 Hz	-1.5 dB	-1.1 dB
125 Hz	-1.5 dB	-3.7 dB
160 Hz	-3.1 dB	-0.3 dB
200 Hz	-7.1 dB	1.2 dB
250 Hz	3.4 dB	-1 dB
315 Hz	0.9 dB	-6.2 dB
400 Hz	-2.7 dB	-3.7 dB
500 Hz	-1.2 dB	-8.3 dB
630 Hz	-1.9 dB	-10.5 dB
800 Hz	0.2 dB	-5.5 dB
1000 Hz	-3.1 dB	-6.5 dB
1250 Hz	1.1 dB	-1.2 dB
1600 Hz	-0.7 dB	0.1 dB
2000 Hz	-3 dB	-3.9 dB
2500 Hz	1.7 dB	-1.1 dB
3150 Hz	1.1 dB	-0.2 dB
4000 Hz	0.4 dB	-3.3 dB
5000 Hz	-1.5 dB	-2 dB
6300 Hz	0.8 dB	-0.9 dB
8000 Hz	-0.6 dB	-3.4 dB

10000 Hz	-0.9 dB	-3.5 dB

Table 1 Spectral analysis-lateral skirts

3.2.2 Array analysis

The results of the measurements with the microphone array are shown as colored maps in the figures in this sub-section and in the appendix.

Again the dynamic range is 20dB from highest level (red) to the lowest level (purple). The red-framed rectangle stands for the lateral skirt.

The diffraction of the acoustic waves around the lateral skirts as well as the characteristics of the propagated sound from the surface of the lateral skirts itself can be seen at specific frequencies.

Table 2 represents the value of the maximum sound pressure level within the area that represents the bogie and is shown in the further figures as the basis to build a difference between the level of propagated sound without and with the lateral skirt. Again, negative values represent a mitigating effect.

Frequency	Level-Max Difference at 80km/h	Level-Max Difference at 120km/h
1000 Hz	-3.8 dB	-8.2 dB
1250 Hz	0.8 dB	-4.8 dB
1600 Hz	-9.2 dB	-6.4 dB
2000 Hz	-5.2 dB	-4.1 dB
2500 Hz	0.1 dB	-6.2 dB
3150 Hz	9.9 dB	1 dB
4000 Hz	-0.3 dB	-4 dB

Table 2 Spectral analysis-lateral skirts

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Figure 10: Co'Co' freight locomotive with mounted lateral skirts

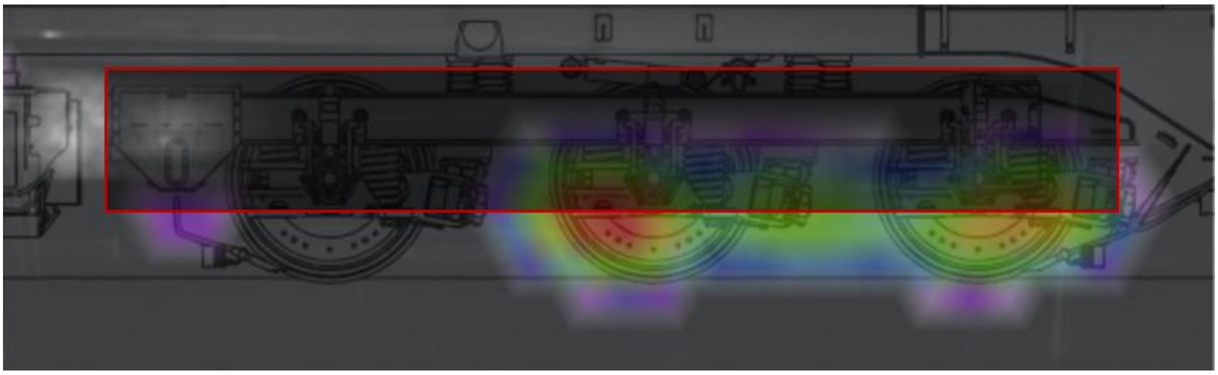

Figure 11 ElectricMode_80kph_3rdOctave1000Hz

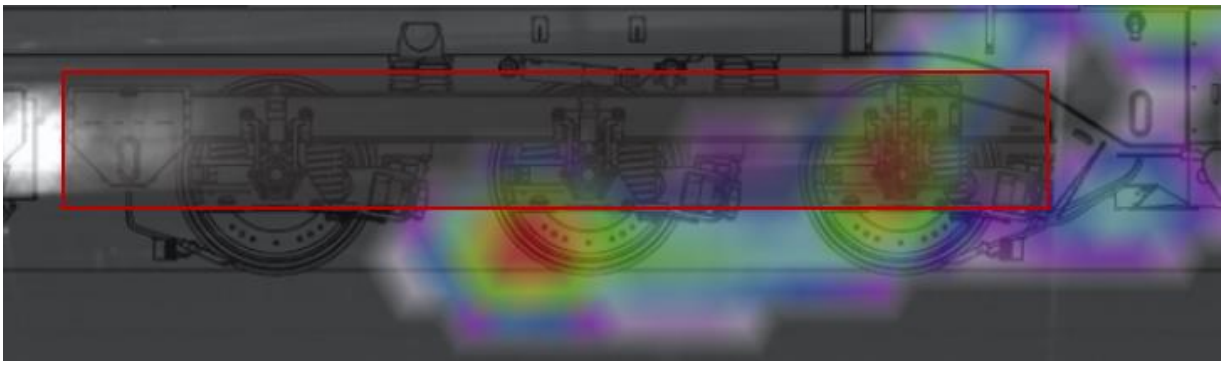

Figure 12 Electric Mode_120kph_3rdOctave1000Hz

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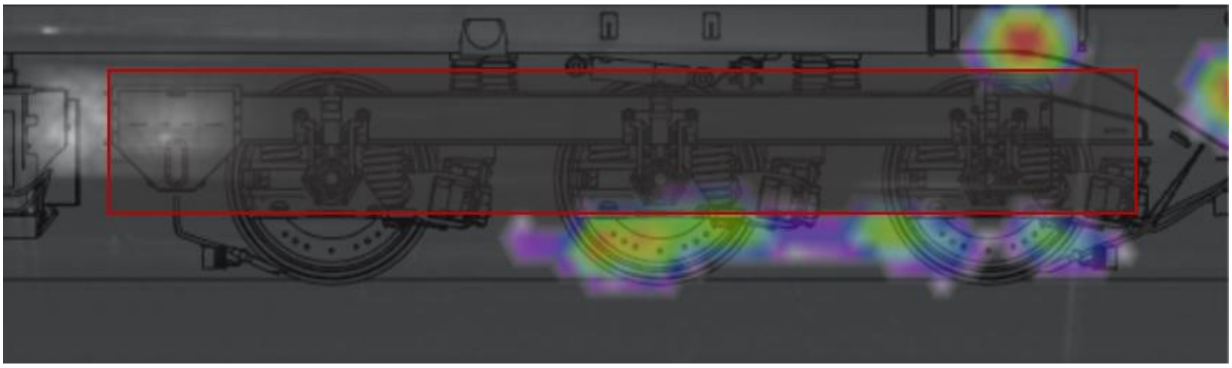

Figure 13 Electric Mode_80kph_3rdOctave_2000Hz

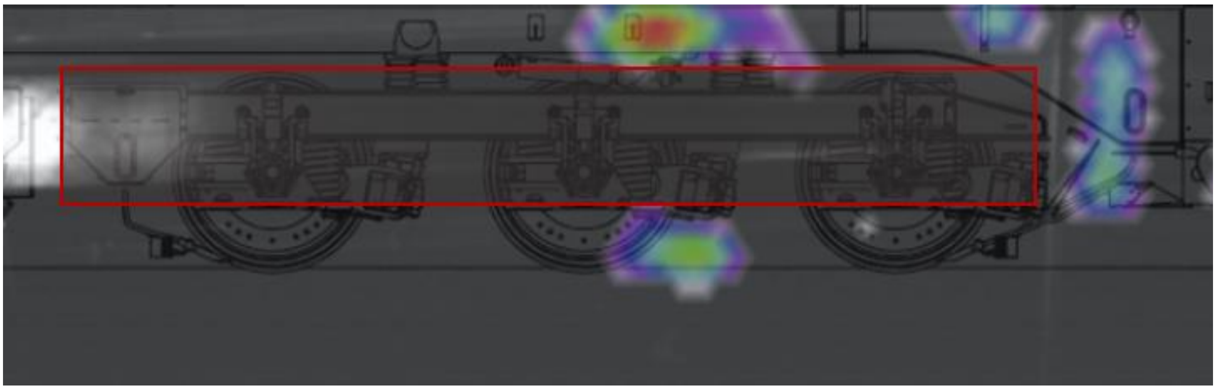

Figure 14 Electric Mode_120kph_3rdOctave_2000Hz

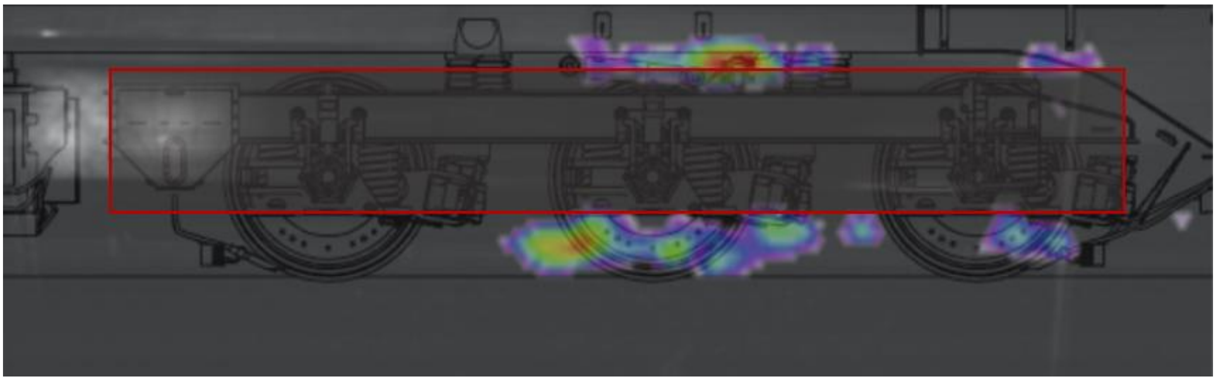

Figure 15 Electric Mode_80kph_3rdOctave_4000Hz

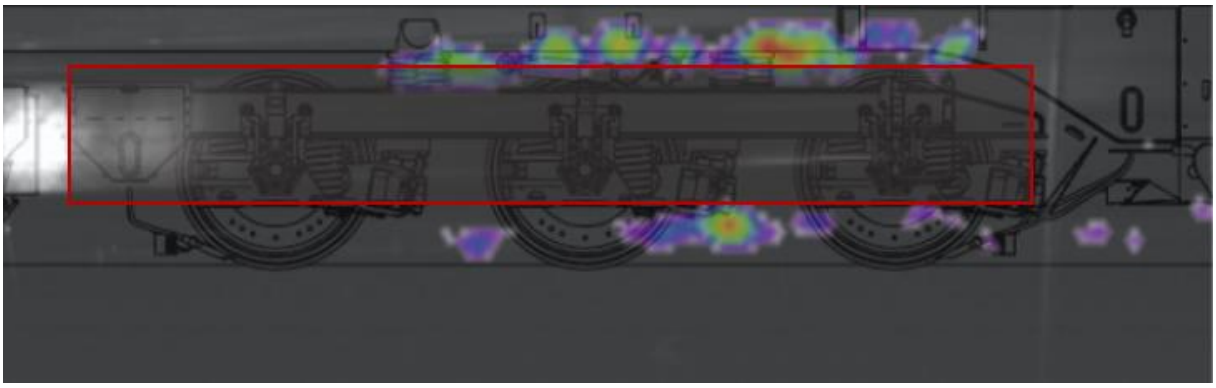

Figure 16 Electric Mode_120kph_3rdOctave_4000Hz

3.3 WHEEL-SET OPTIMIZATION

This chapter describes the results of the analysis work done to identify and compare the noise emission of the solid wheel of a Co'Co' freight locomotive from Stadler in two conditions:

- Wheel without the brake disk
- Wheel with the brake disk

The comparison is made first through a theoretical analysis performed by using ANSYS FEM software to determine the wheels modal model and the Stardamp software (based on TWINS algorithm **Error! Reference source not found.**) to determine the noise emission of the wheels c onsidering the wheel rail interaction.

In order to measure the right damping factor inducted by the brake disk, it was necessary to perform a modal analysis impact test in laboratory on the Co'Co' freight locomotive selected wheel, with and without the brake disk.

3.3.1 General information

General information regarding measuring units and software parameters are given in this subsection.

3.3.1.1 Measuring Units

If not differently specified, the following measuring units and notations were used:

- Length: [mm]
- Frequency: [Hz]
- Lp (Sound pressure level at 7.5m from the track and 1.2m height): [dB(A) re 20 μPa]

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- Lw (Sound power level): [dB(A) re 10^{-12} W]
- Roughness: [dB re μm]
- FRF Accelerance: [dB re m/s2/N]

3.3.1.2 Software parameters

The parameters adopted for the calculations were set in order to be in accordance with the measurement made by TUB and STAV. Anyway these parameters are quite negligible since they are fixed for all the analysis and so the comparison results between the two conditions of the wheel are therefore not affected by them. In particular:

- Wheel tread roughness: typical roughness of a wheel braked by disks
- Rail roughness: standard roughness defined in TSI
- Train speed: 80km/h (typical for freight cars)
- Track type: ballast
- Sleeper type: concrete monoblock
- Rail type: UIC 60E1
- Pad stiffness: high stiffness, normal damping

Figure 17 shows the selected roughness spectra R(λ) for the rail and the wheel and their combined total roughness spectra.

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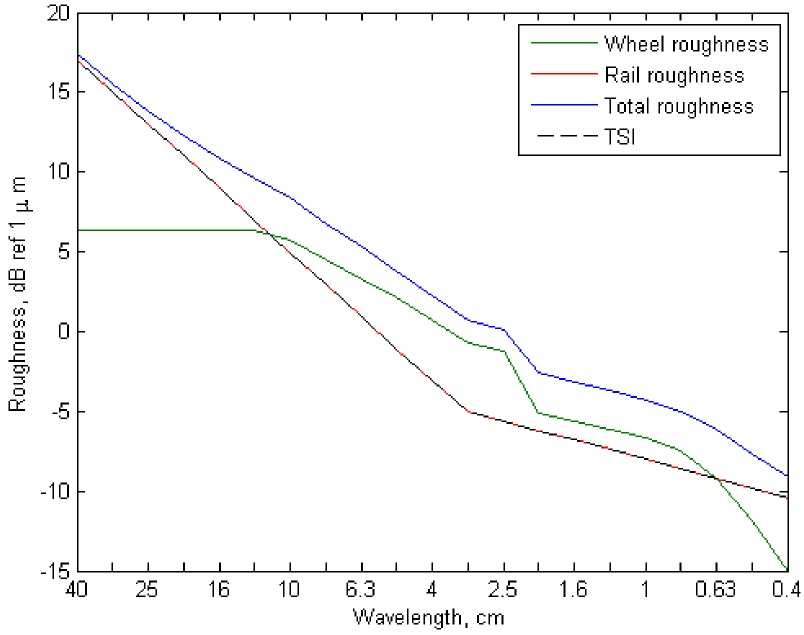


Figure 17 Roughness adopted by the Stardamp software

3.3.2 Modal analysis impact test

This experimental test was performed to measure the damping factor introduced by the assembly of the brake disk in order to compare the two configurations of the wheel with the simulations with Stardamp software.

3.3.2.1 Test procedure

The test consists in exciting the wheel with an instrumented hammer (Figure 18) and measuring the induced wheel vibration by a triaxial accelerometer placed close to the excitation point. In this way it is possible to calculate, for each position in which the accelerometer is placed, the Frequency Response Function (FRF) ([5],[6]) between the acceleration and the excitation force; also called accelerance.

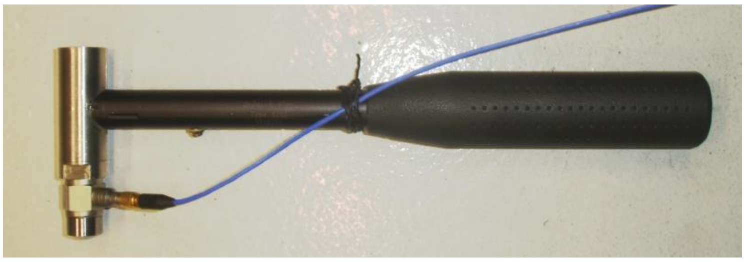
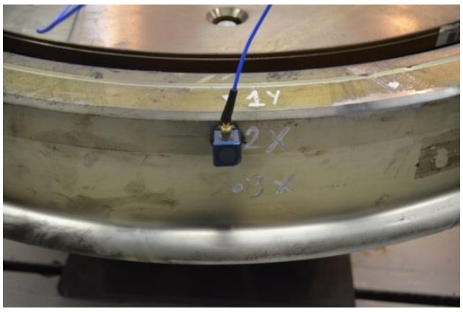


Figure 18 Impact hammer

During the test the wheel is resting through its internal hub lateral side on an elastic rubber support. In this condition the wheel is free to vibrate when excited on the tread surface. Figure 19 identifies the position and the direction in which the wheels were excited and where the accelerometer was placed.



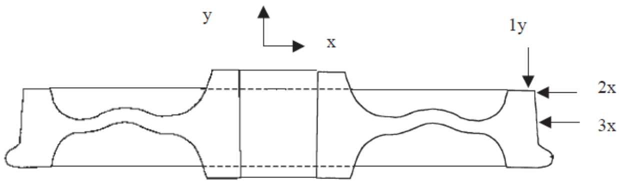


Figure 19 Nomenclature of the excitation/response points for the vibro-acoustic test

3.3.2.2 Test results

Figure 20 shows the results of the FRF (accelerances) for each measuring point. The other important result it's the damping factor calculated for all the resonance frequencies from 0 Hz to 6 kHz with the peak-picking method (example in Figure 21); this method consists of evaluating the damping as a function of the "width" of the peak: the first step is to identify the amplitude of a resonance peak in the FRF diagram, then measure the Δf at an amplitude of 3dB under the peak, so the damping is calculated as:

$$c = \frac{\Delta \mathrm{f}}{f_{peak}} * 100$$

Where:

- c [%] = damping [%]
- Δf [Hz] = "width" of the peak at 3dB under the peak amplitude
- f_{peak} [Hz] = frequency of the resonance peak

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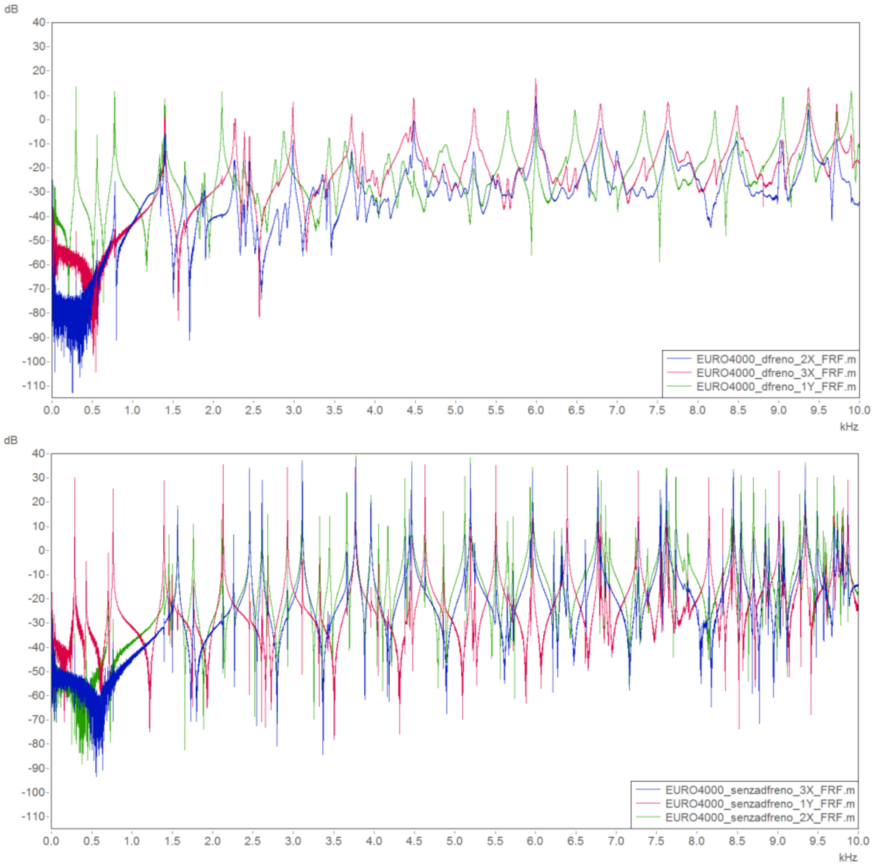

Figure 20 Experimental accelerance FRFs in the three positions: 1Y, 2X, 3X. Wheel with the brake disc (above), wheel without the brake disk (below)

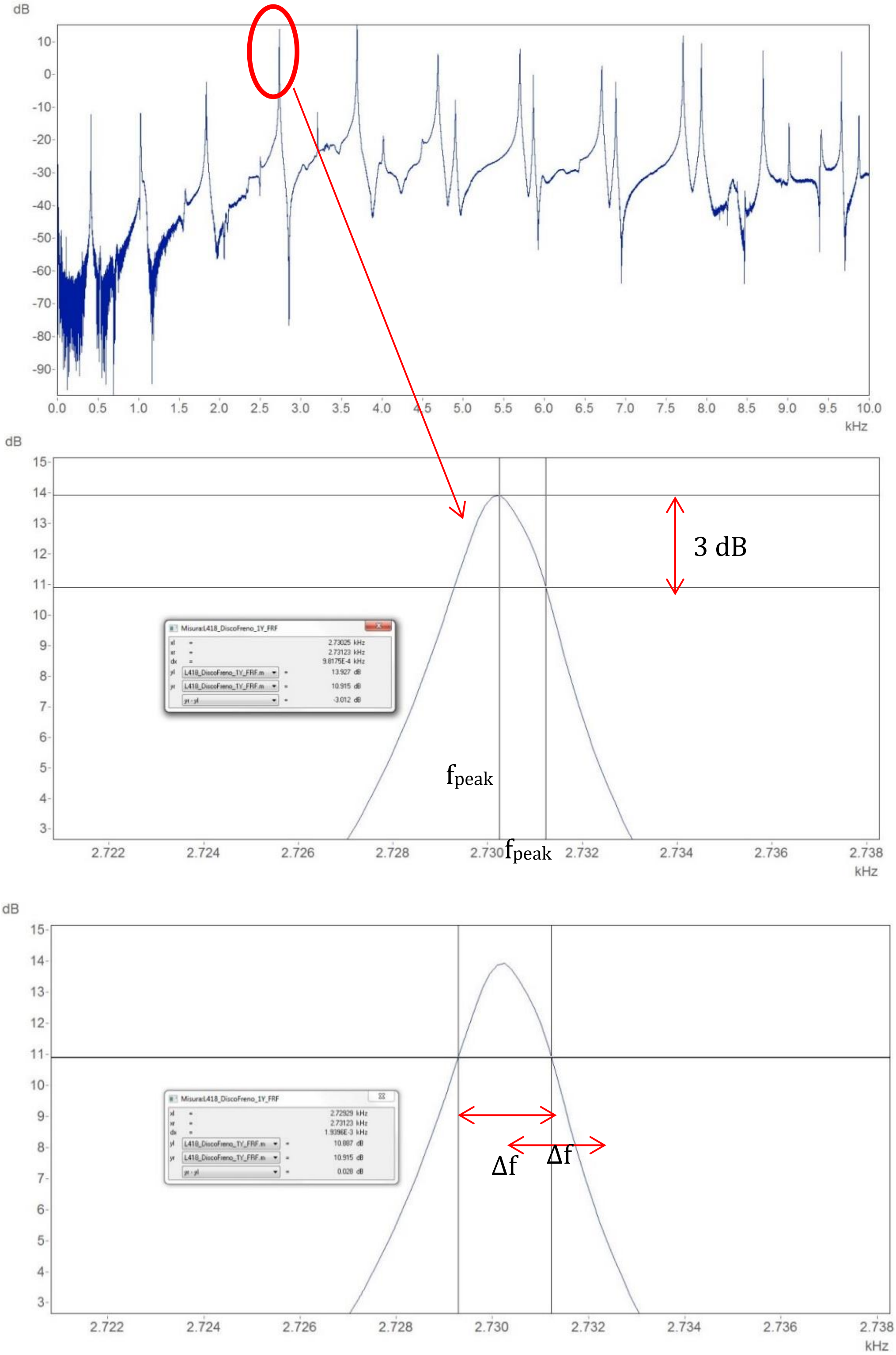

Figure 21 Peak-picking method, example

The following tables summarize the damping factor obtained for both configuration of the wheel:

Wheel with brake disk	
Resonance frequency [Hz]	**Damping [%]**
290.4	0.14
551.1	0.33
766.6	0.10
1390	0.16
1641	0.55
1885	0.39
2100	0.14
2260	0.53
2367	0.65
2442	0.20
2765	0.89
2869	0.70
2982	0.19
3431	0.48
3629	0.69
3710	0.37
3844	0.38
4170	0.82
4479	0.19
4878	1.58

Wheel without brake disk	
Resonance frequency [Hz]	**Damping [%]**
282.3	0.02
423.4	0.26
699.6	0.13
1391	0.03
1453	0.09
1492	0.13
1560	0.07
1755	0.11
2124	0.01
2256	0.02
2455	0.01
2606	0.01
2678	0.02
2923	0.01
3108	0.01
3323	0.02
3444	0.04
3658	0.03
3767	0.01
3958	0.04
4054	0.02
4168	0.07
4381	0.02
4461	0.01
4629	0.00
4859	0.01

Table 3 Damping factor for all the resonance frequency. Wheel with brake disk (left), wheel without brake disk (right)

The next figure shows the same results in a graph to simplify the comparison.

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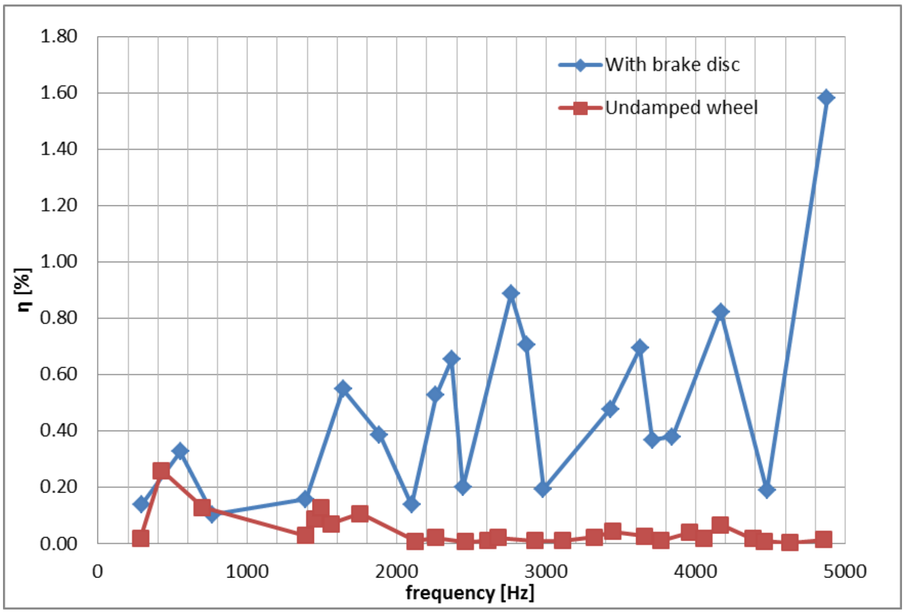

Figure 22 Damping factors

As shown in Table 3 and in Figure 22, the brake disc has a great influence on th wheel's acoustic behaviour, in fact the disc mass reduces significantly the number of resonance frequencies while also improves the damping and so the results is that the wheel with the brake disc vibrates at lower frequencies for very short time instead of the one without the disc that vibrates for longer time at higher frequencies that are more disturbing.

3.3.3 Noise emission calculation with the STARDAMP software

In order to simulate the noise emission of a railway wheel rolling at a certain speed over a rail the STARDAMP software is used; this software is based on the TWINS algorithm [8] that models the wheel and track dynamic interaction and then the noise emission.

The inputs to the software are: the modal model of the wheel and of the track, the roughness spectra of the wheel tread and of the rail, the wheel speed (Figure 24).

The modal model of the two wheels is made with the Finite Element Software ANSYS using harmonic elements "PLANE83" (see Figure 23 the mesh used for the analysis of the wheel).

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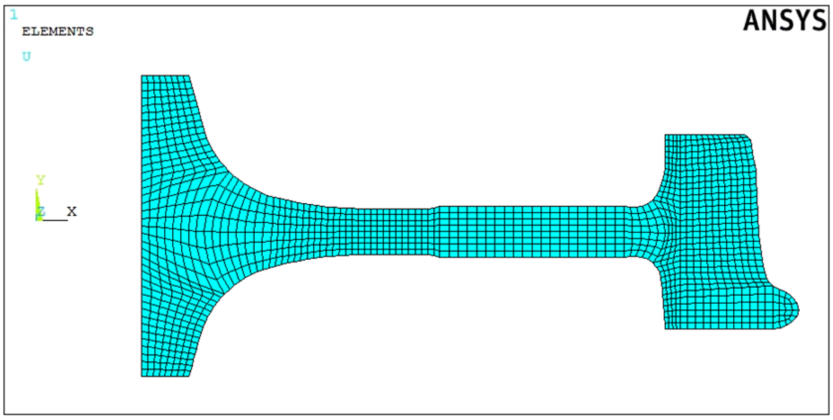

Figure 23 2D mesh for the FEM modal analysis performed on the selected wheel

From the built finite element models, the results have to be exported to Stardamp in terms of frequencies and associated modal shapes; the response of the wheel to the force due to the wheel-rail interaction is evaluated in terms of displacements of the nodes on the web in the axial direction, for each mode of vibration.

Figure 24 shows the trend of the resonance frequencies of the wheel. The contributes are plotted considering the number of nodal circles and the nodal diameters.

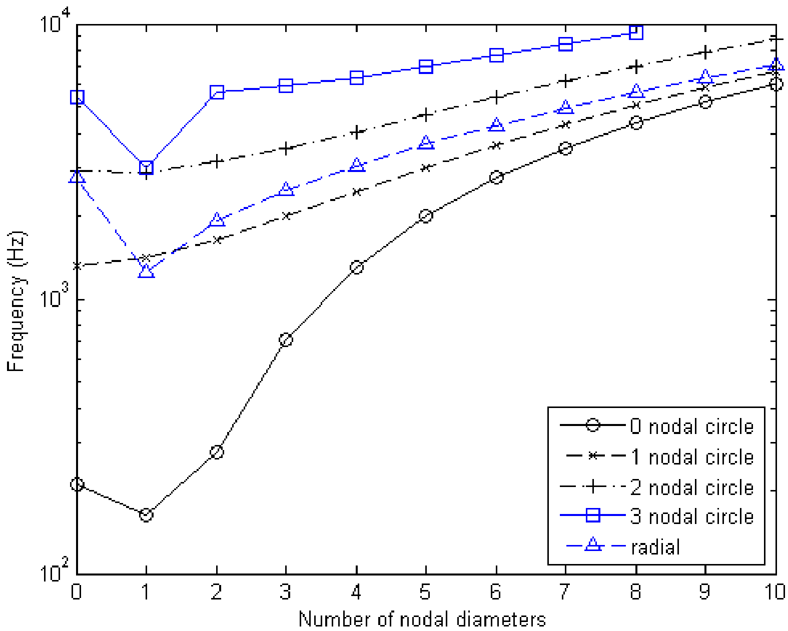

Figure 24 Frequencies associated to the modal shapes of the selected wheel

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The damping ratios at the various frequencies, for the wheel without the brake disk, were set at the default value by Stardamp, only considering the structural damping of the material (10^{-4}); while for the wheel with the brake disk, the damping ratios were set based on the results in Table 3.

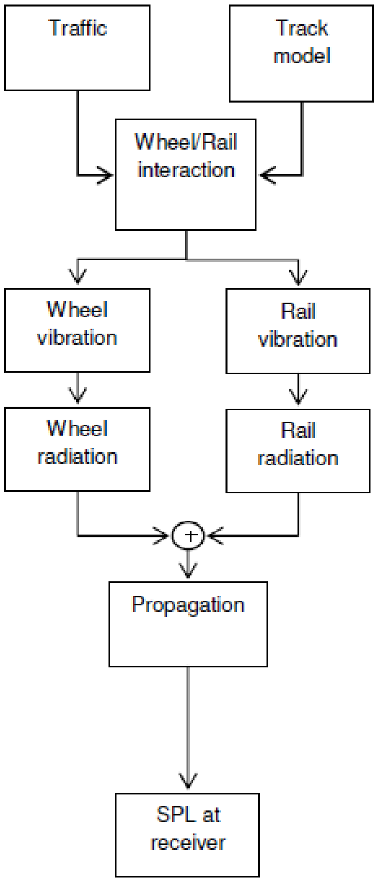

Figure 25 Flow chart for rolling noise calculation procedure

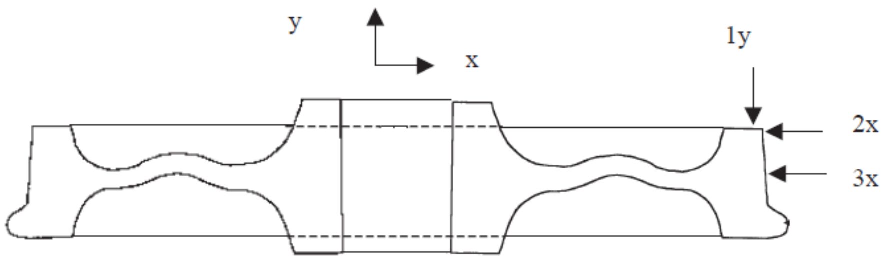

Figure 26 Nomenclature of the excitation/response points for the validation of the FEM modal model

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3.3.4 Results

Results of the STARDAMP analysis are expressed in terms of A-weighted Sound Pressure Level (Lp) at 7.5 m / 1.2 m over 1/3 octave band, emitted by the noise source. Figure 24 show the diagrams of noise emission of the wheels and the track separately and combined (total sound pressure). Table 4 summarizes the different contributions in terms of the overall frequency band Sound Pressure Level, simplifying the global comparison.

Lp [dB (A) ref.2x10^{-5} Pa]	Total	Wheel	Track
Wheel without brake disk	77	72	74
Wheel with brake disk	76	70	74

Table 4 Overall levels of A-weighted sound pressure at 7.5 m / 1.2 m estimated by the STARDAMP software

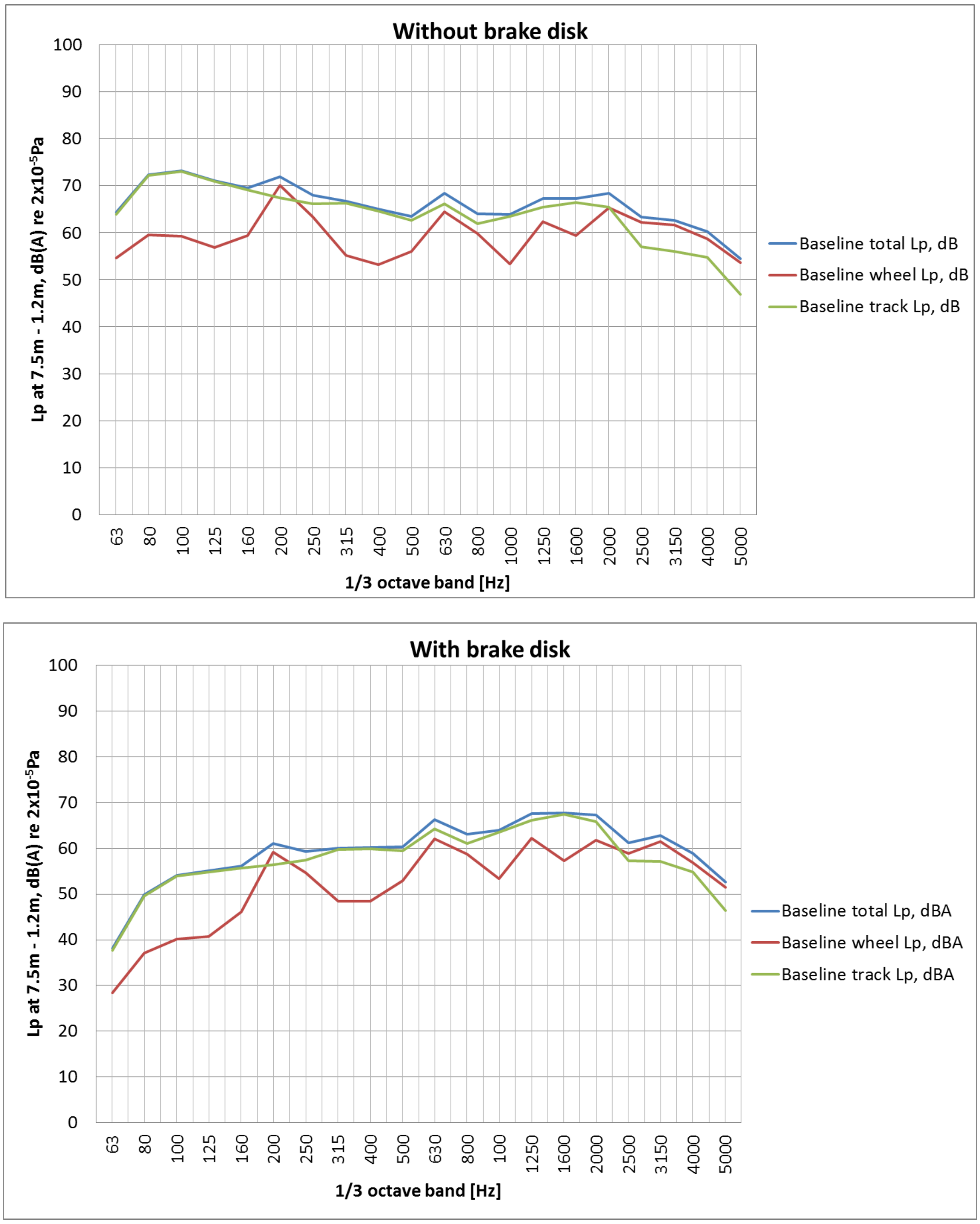

Figure 27 A-weighted Sound Pressure Level emission at 7.5 m / 1.2 m of the selected wheel and track, separately and combined. Without brake disk (above) and with brake disk (below)

3.4 AN OVERVIEW ON FURTHER NOISE MITIGATION MEASURES ON RUNNING GEAR RELATED NOISE EMISSIONS

In this task, the potential of two practical solutions has been assessed.

Further noise mitigation measures on running gear noise including:

- Coated wheelsets
- Coated bogies
- Wheels with noise optimized planks
- Viscoelastic suspension
- Spring inserts

shall be assessed practically or by simulation in further projects to achieve accurate data.

4 CONCLUSION

The identification of the main sources of a locomotive during pass by under different running scenarios has been elaborated by measurements. The characteristic of the noise propagated by the locomotive depends on the power mode, on the frequency and on the speed of train. In every configuration and under various running scenarios the wheel-rail interaction and the bogie are the main contributors to the noise emission.

Two practical solutions to mitigate the noise emission have been worked out.

The noise mitigation potential of lateral skirts has been assessed by measurements. The analyses shows that the mitigation effect of the lateral skirts is highly frequency and train speed dependent. In average, the examined lateral skirts reduce the noise 1 dB at 80km/h and 4.2 dB at 120km/h in a frequency range from 100Hz-10kHz.

The wheel-set optimization has been performed by simulation and lab testing. The application of brake discs shows a noise reduction potential of 1dB.

More mitigation measures have been shown to be elaborated in projects in the future.

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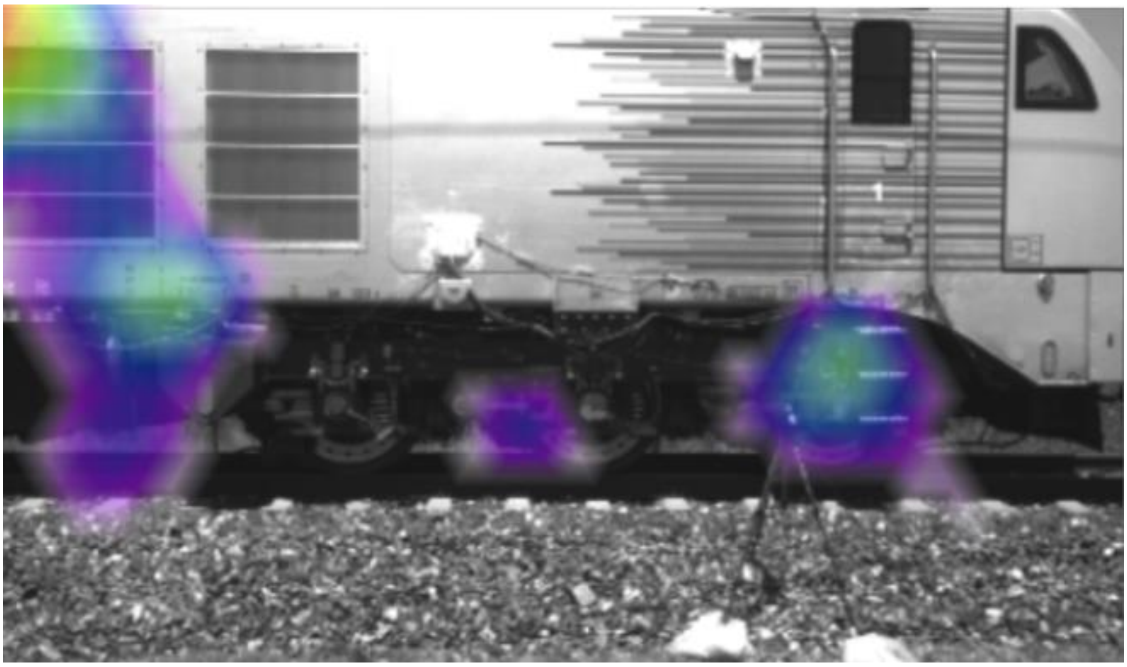
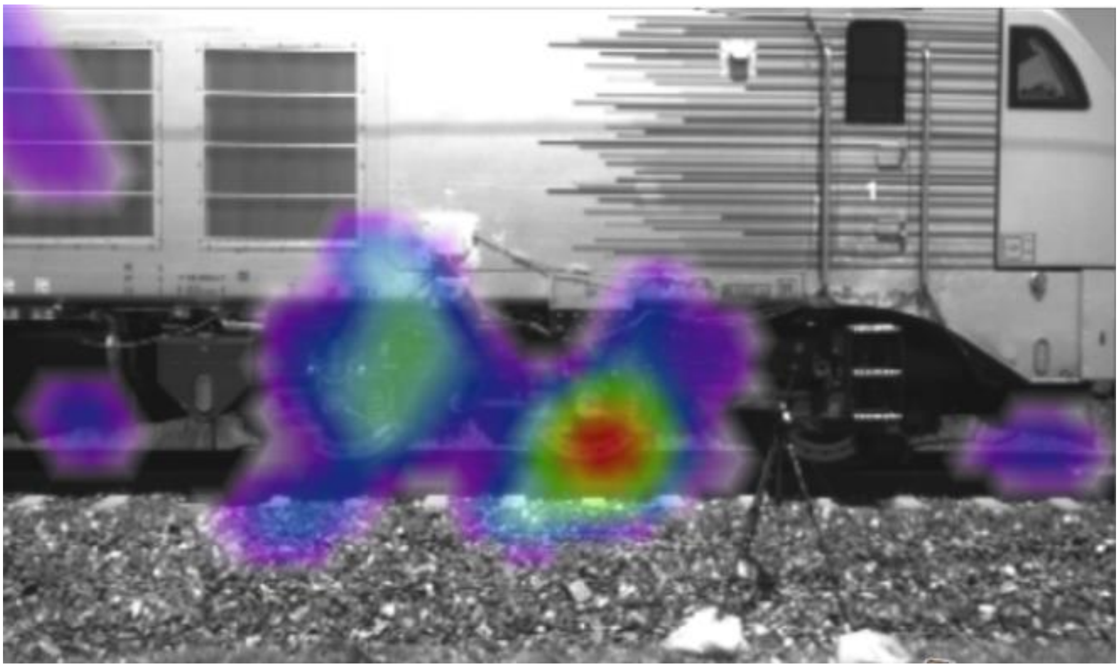
[7] EN13979 2009 – Railway applications – Wheelsets and bogies – Monobloc wheels – Technical approval procedure

[8] Stardamp tool user manual V1.3

1. Noise source identification measurements results



Diesel Mode_80kph_3rdOctave1000Hz



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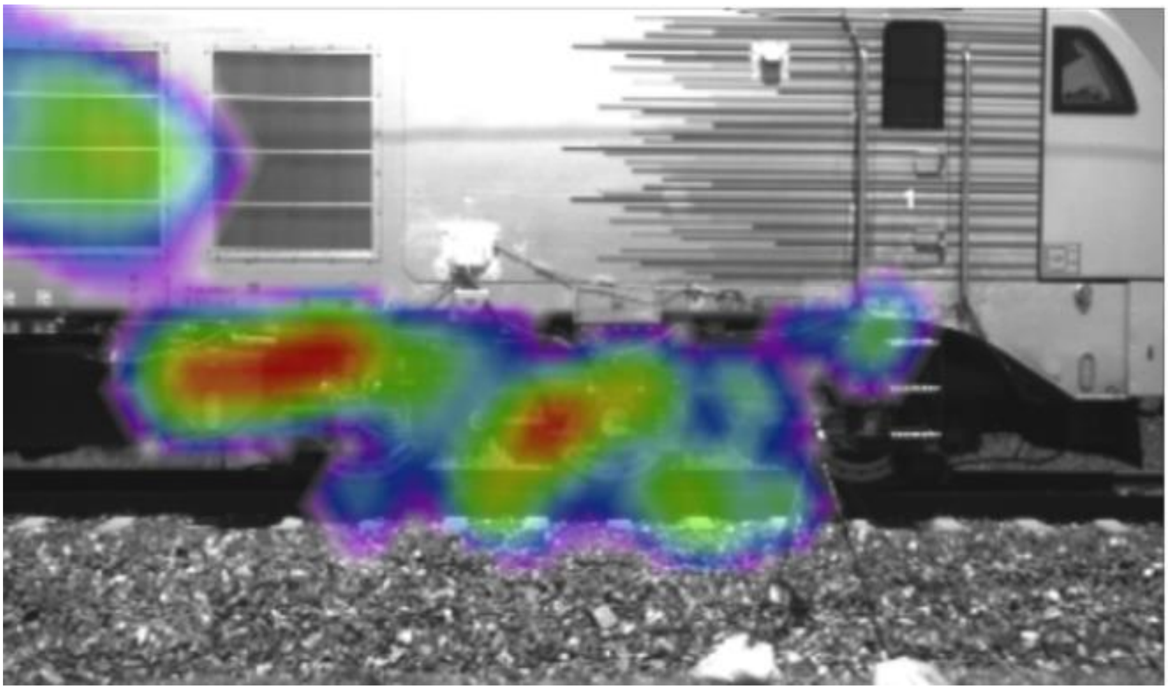

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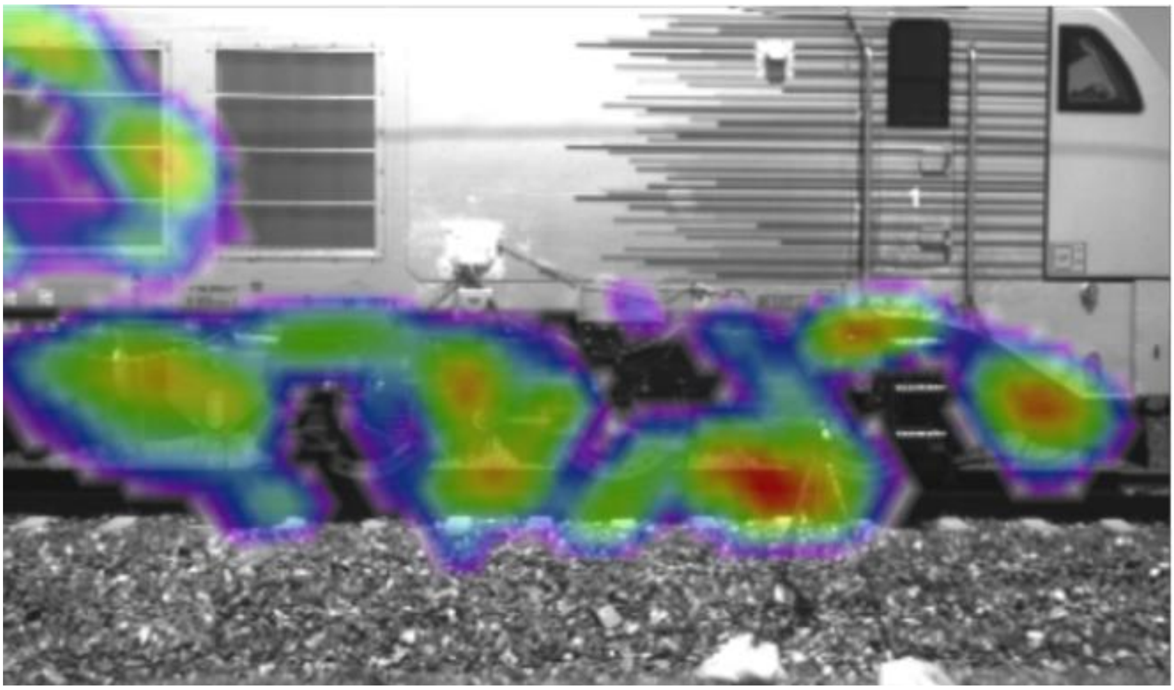

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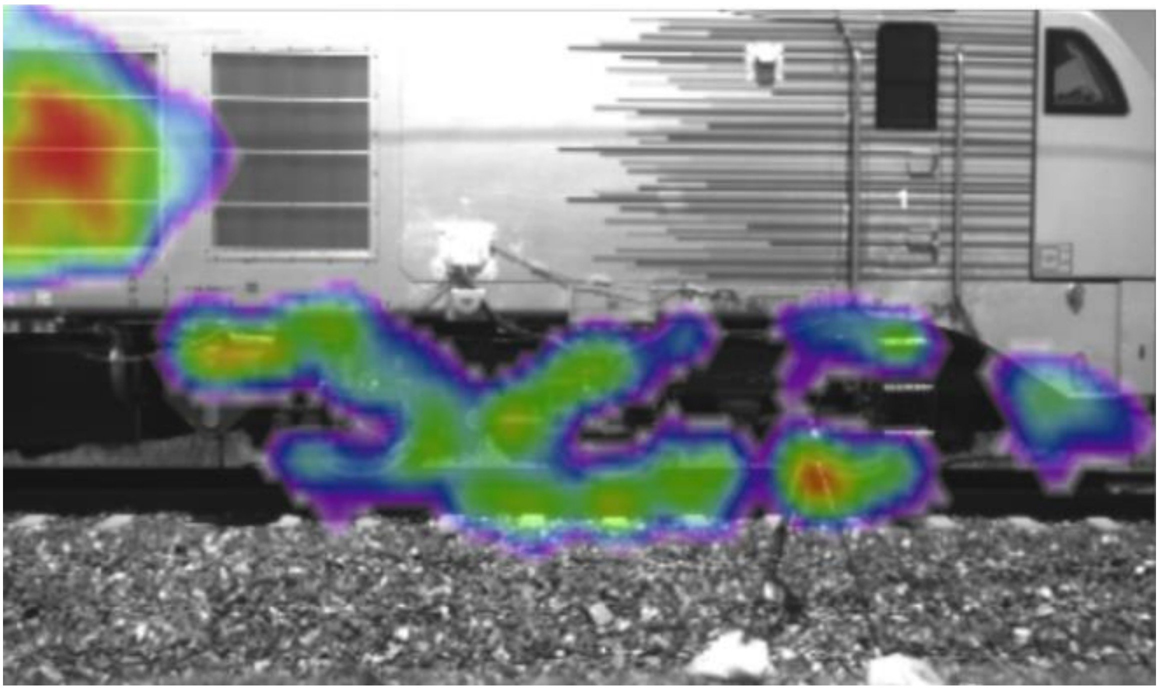

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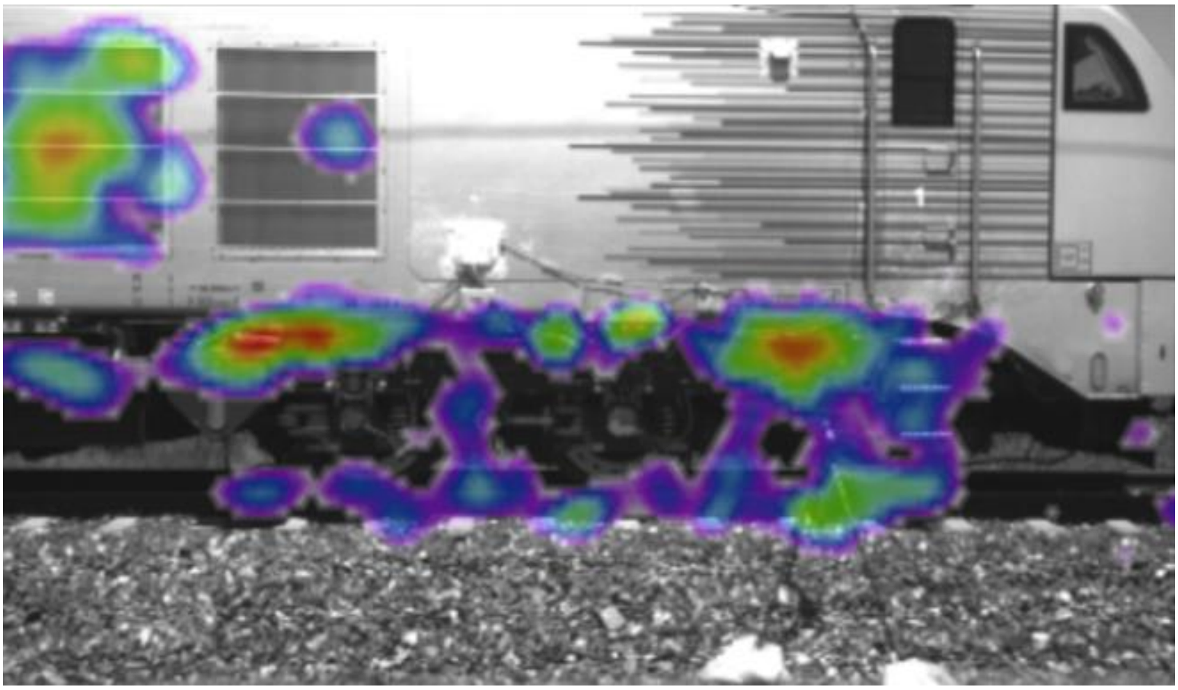

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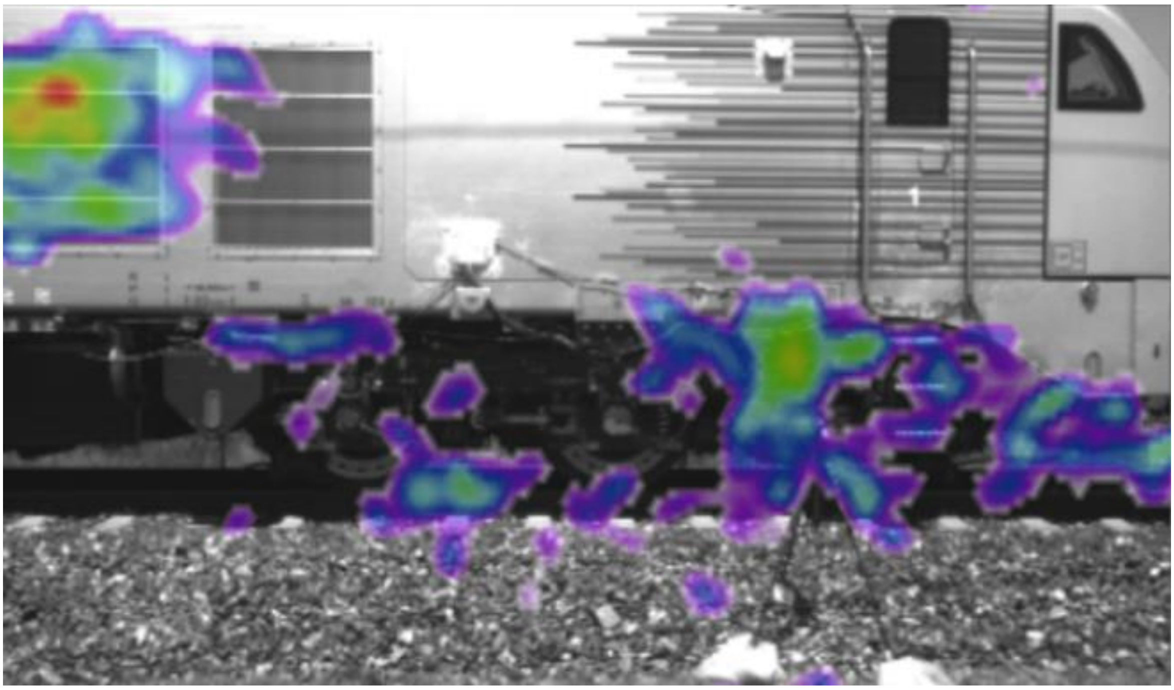

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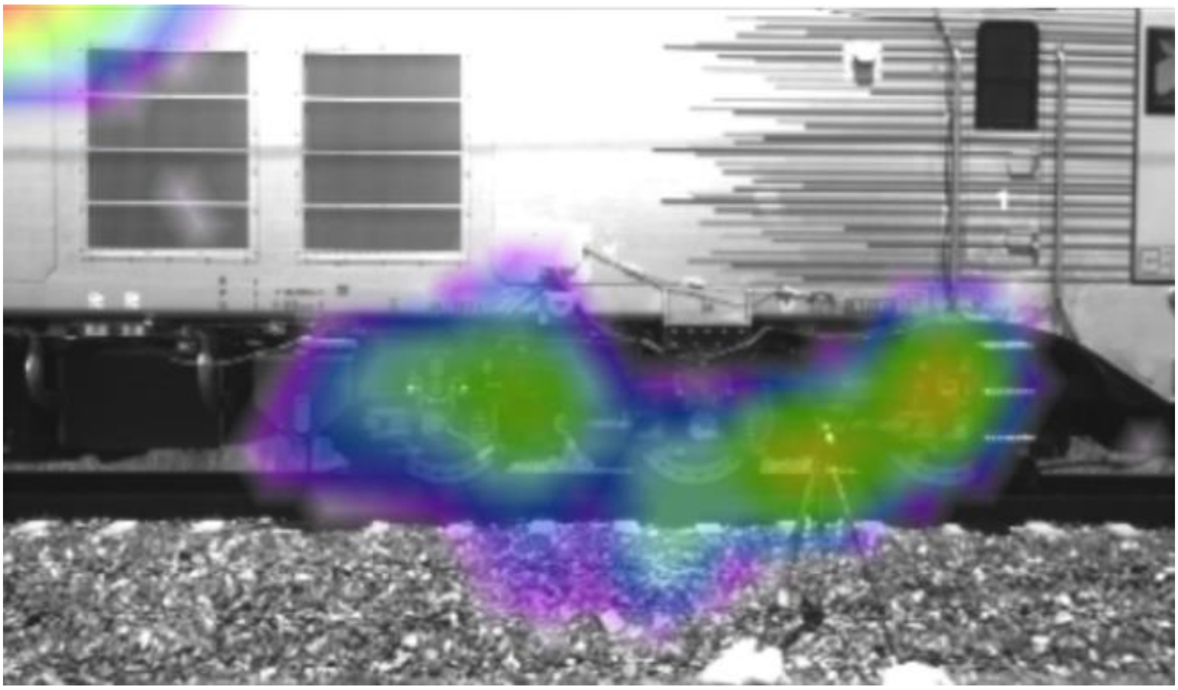

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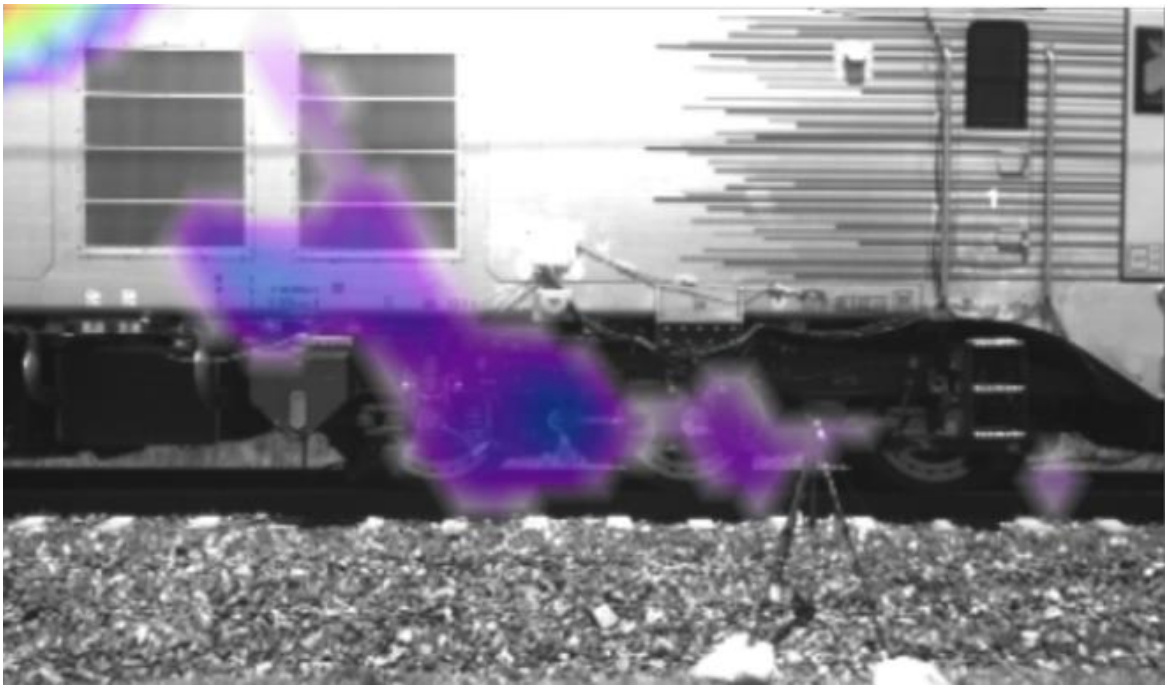

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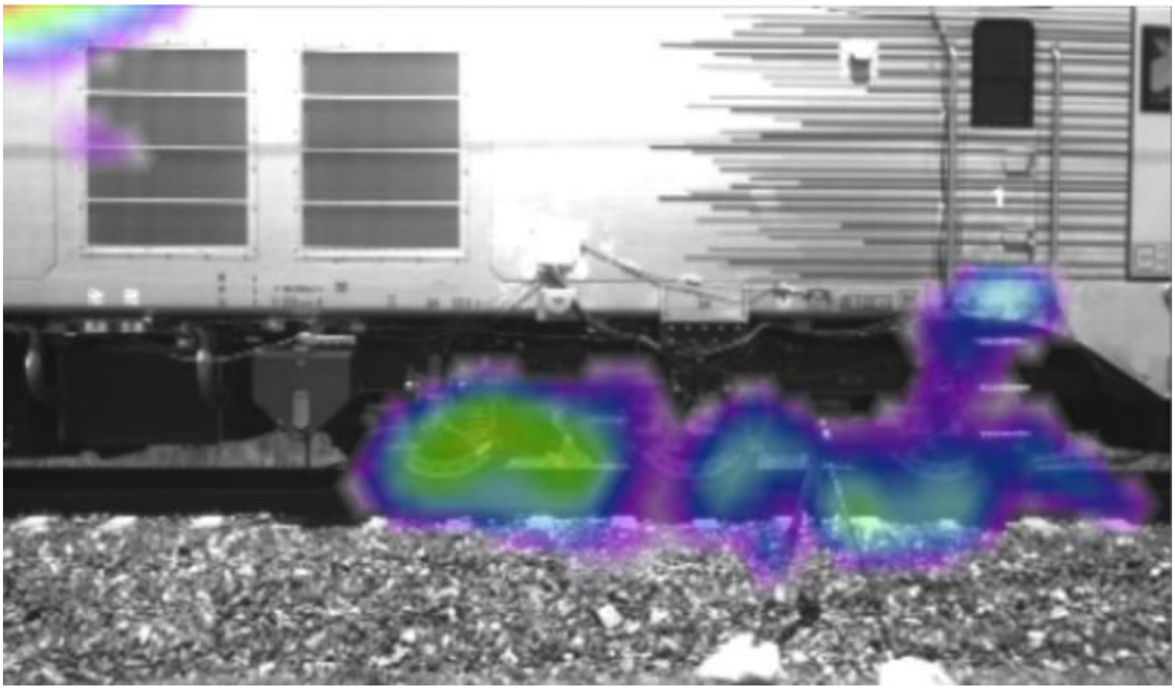

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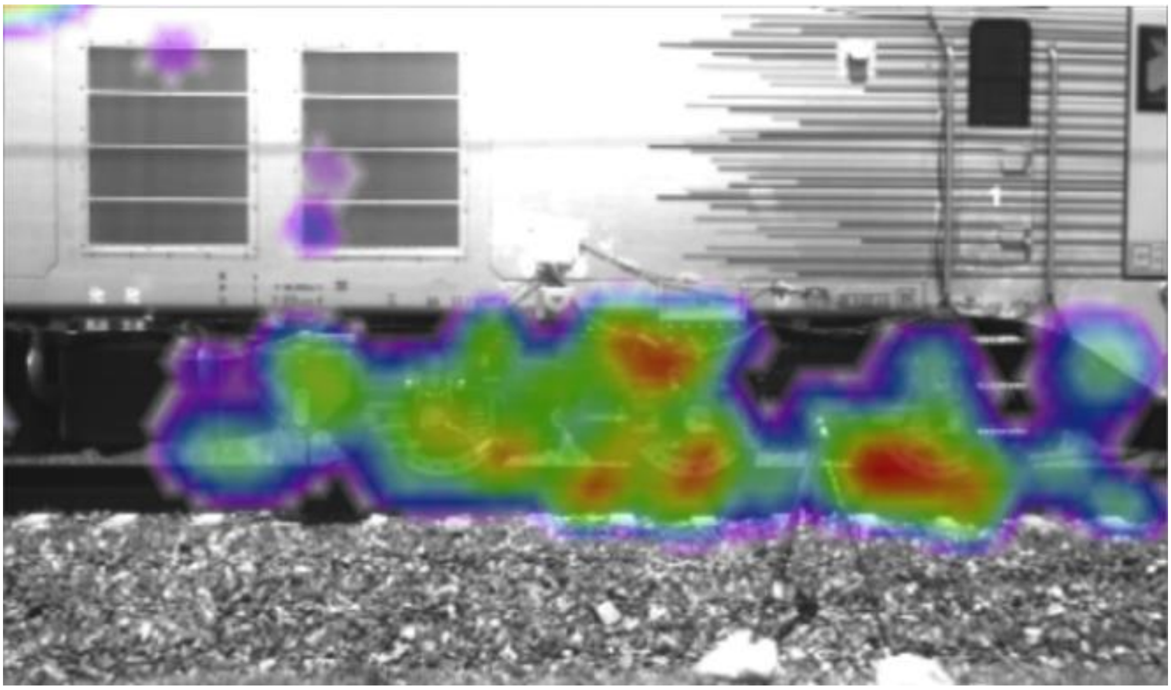

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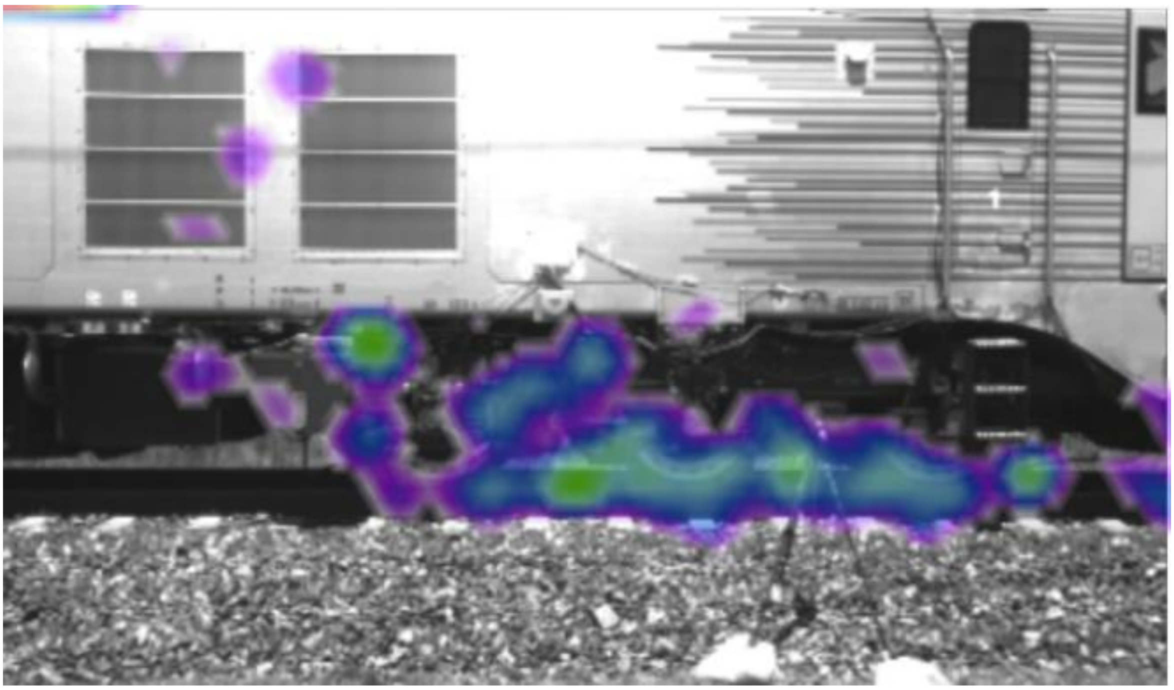

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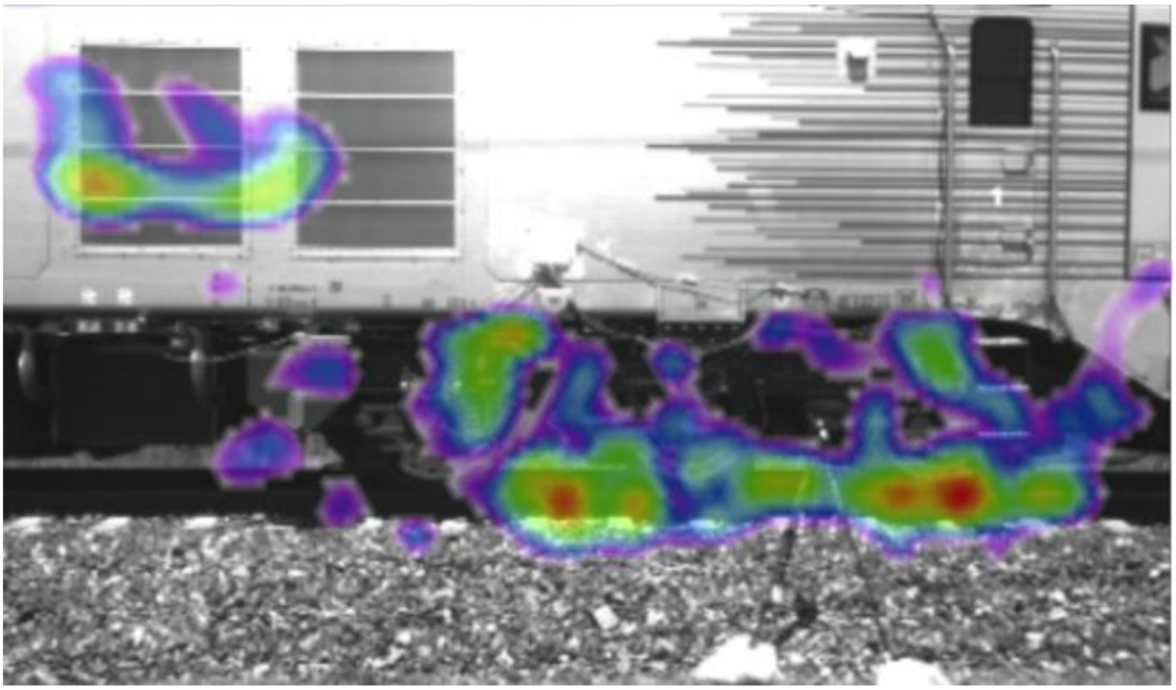

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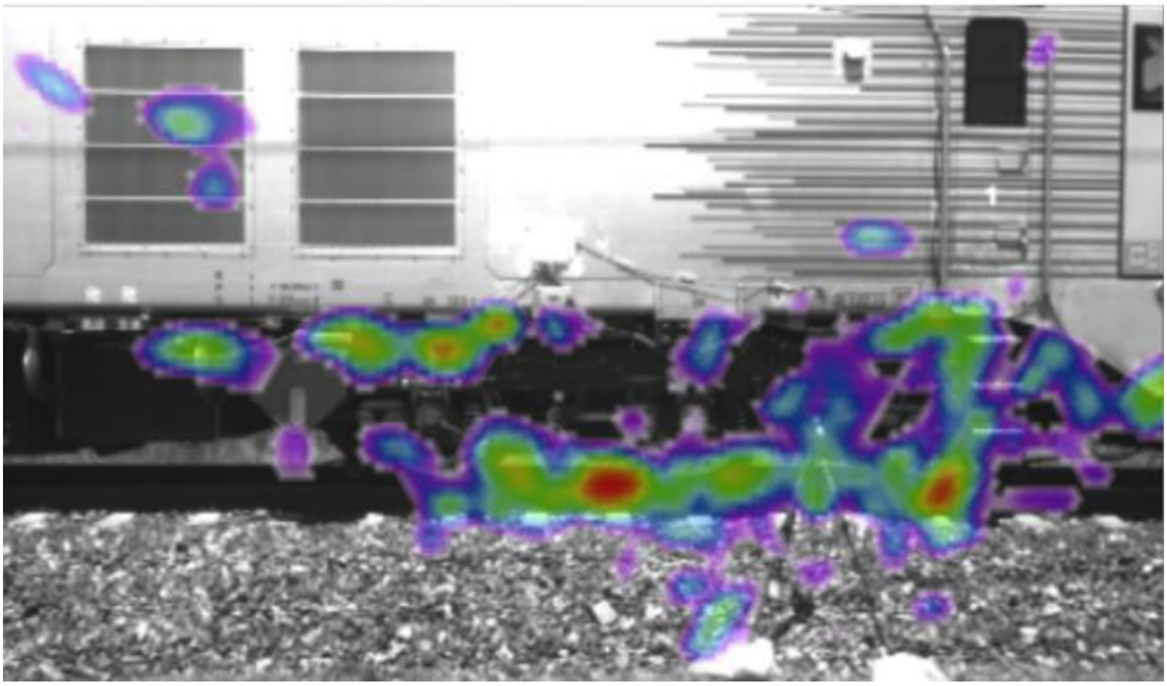

Diesel Mode_120kph_3rdOctave4000Hz

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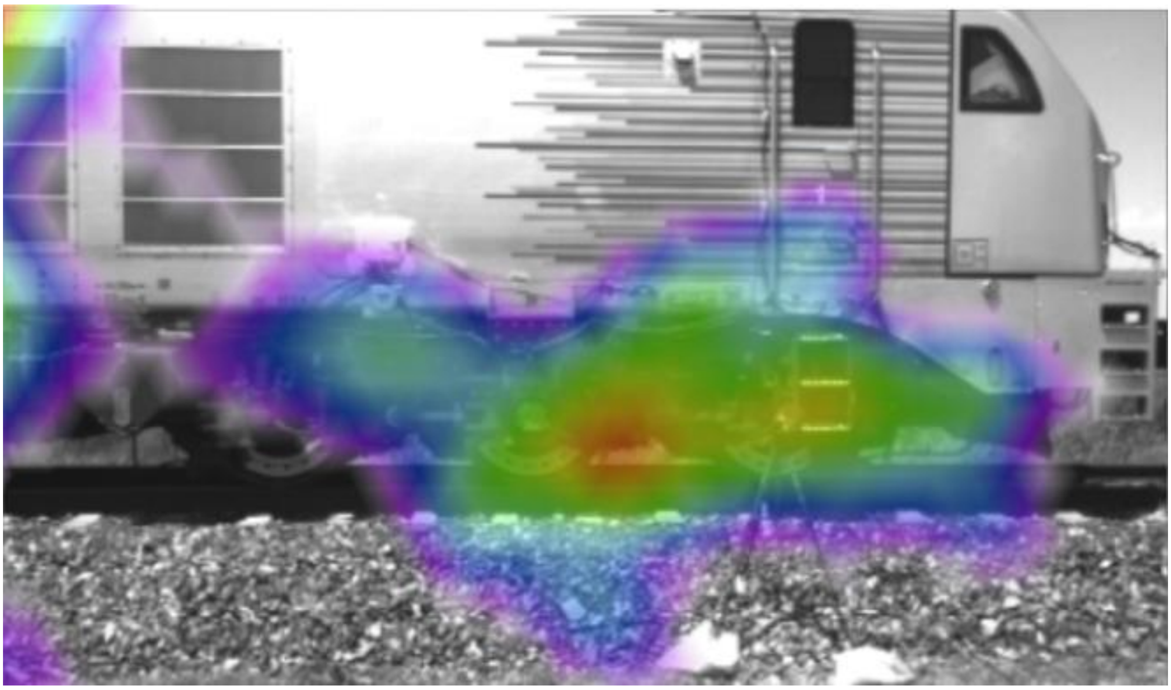

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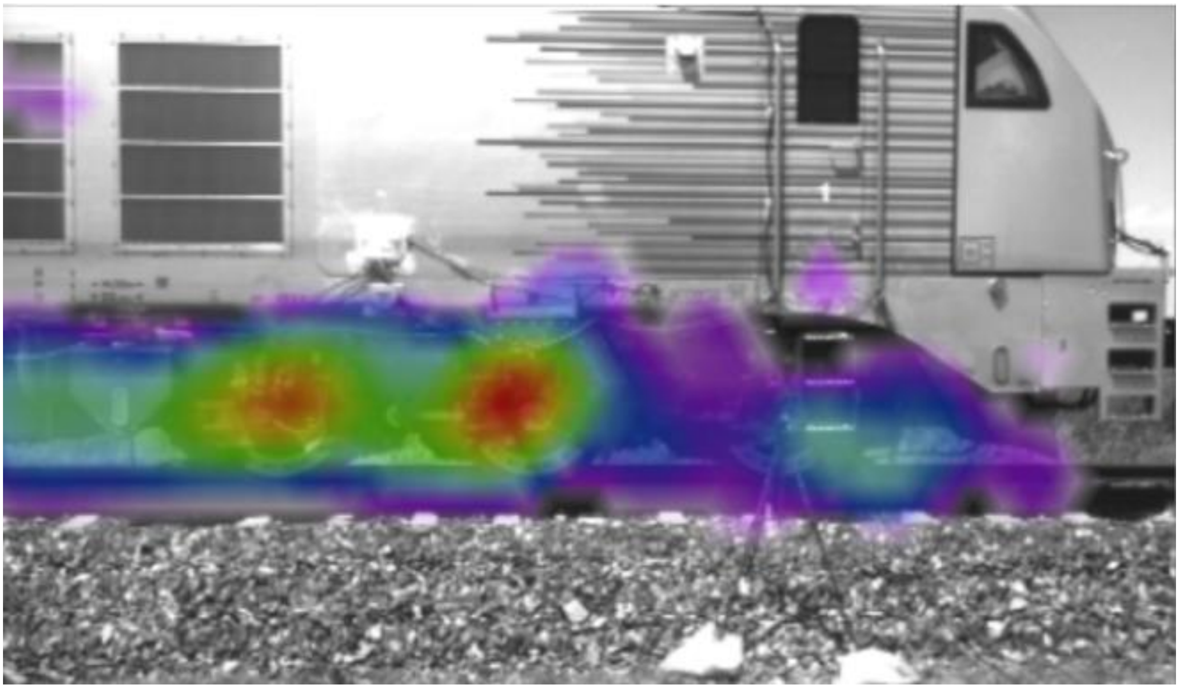

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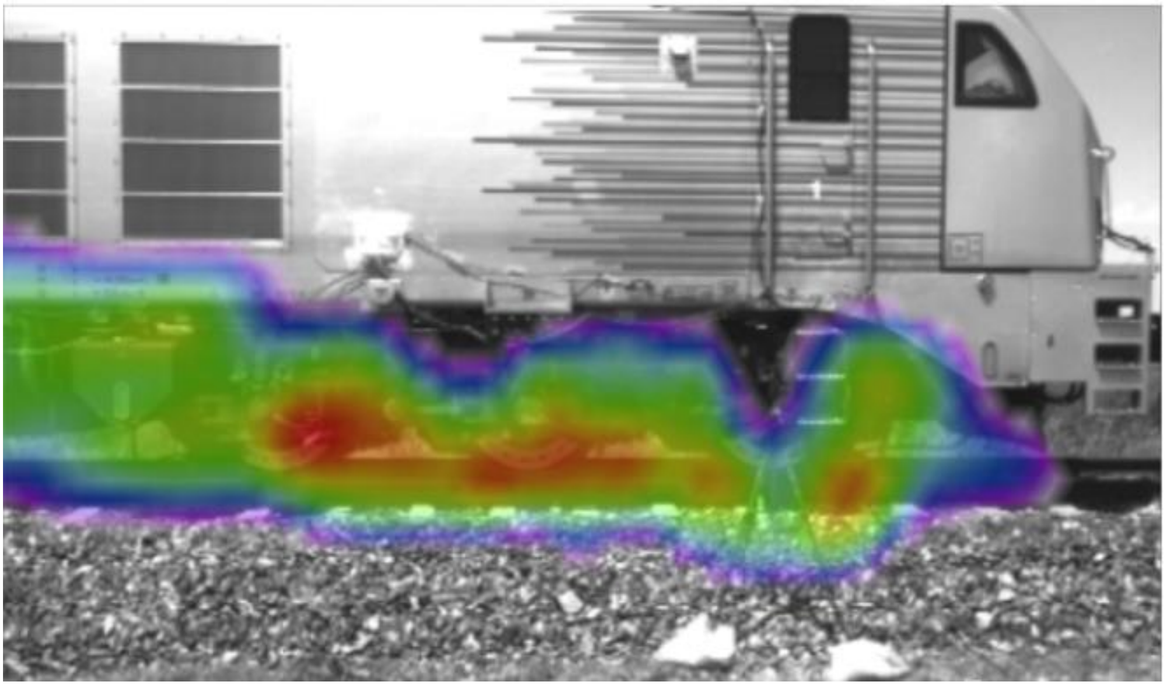

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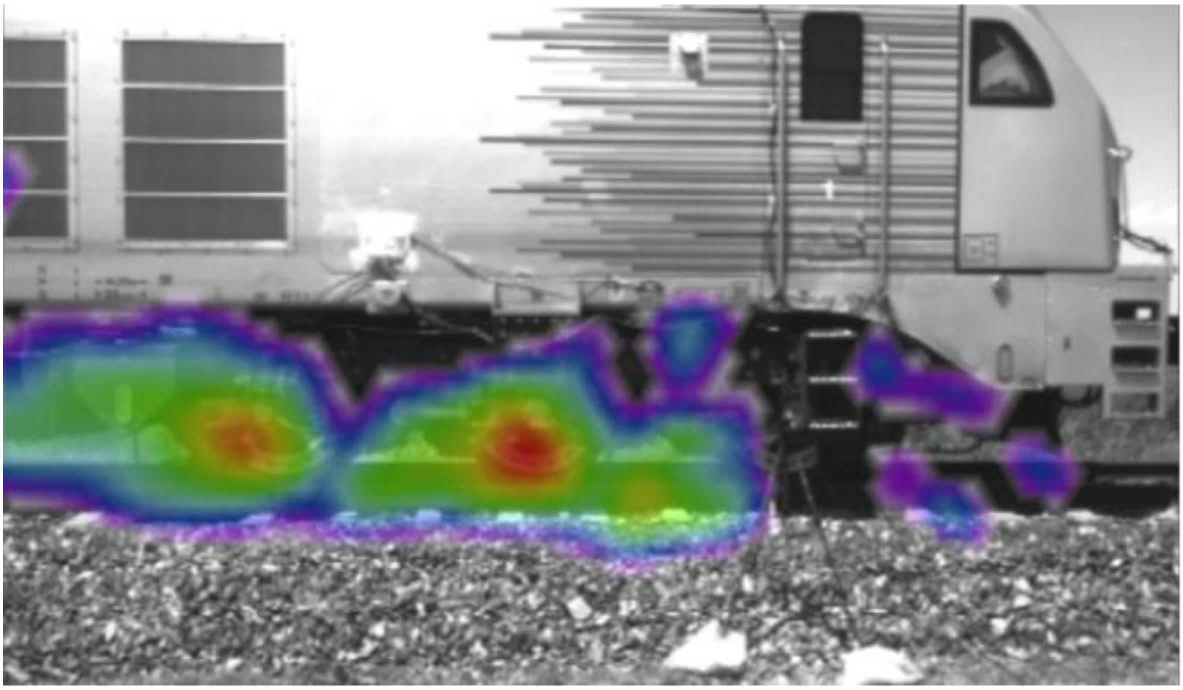

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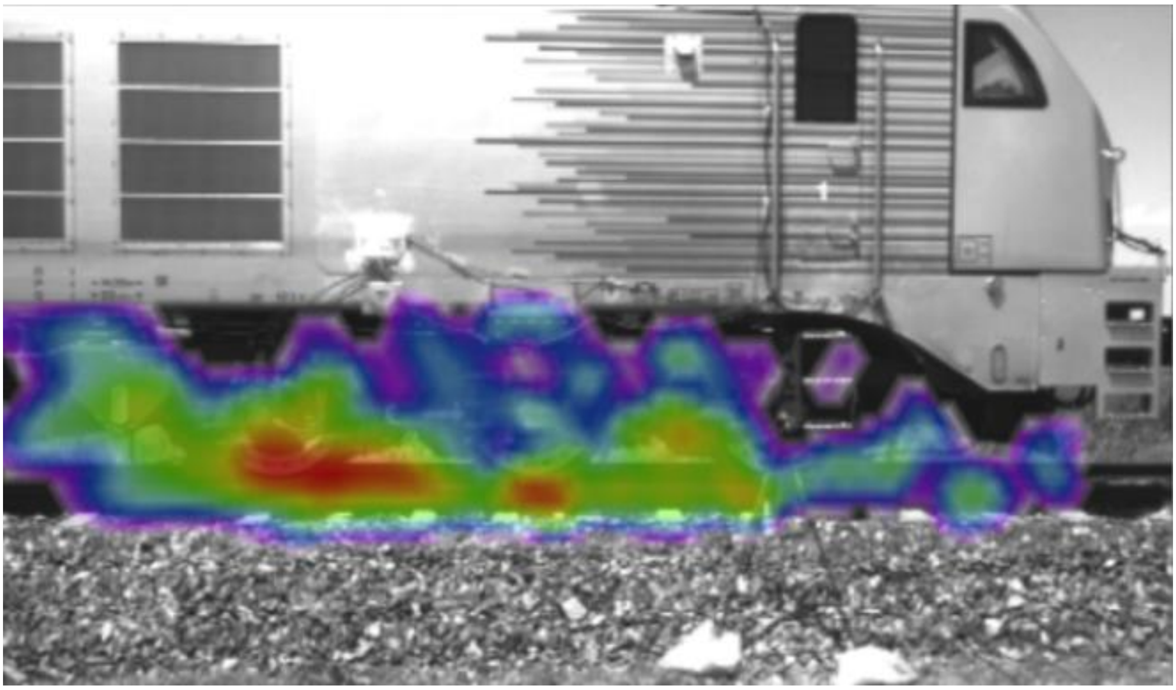

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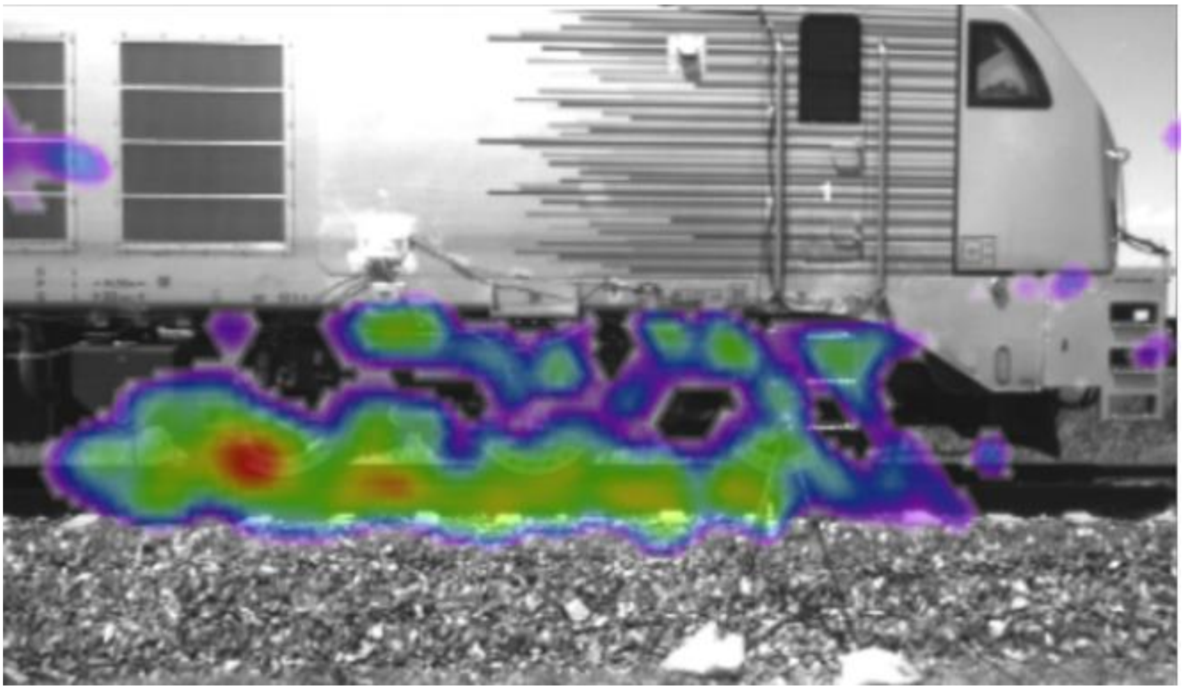

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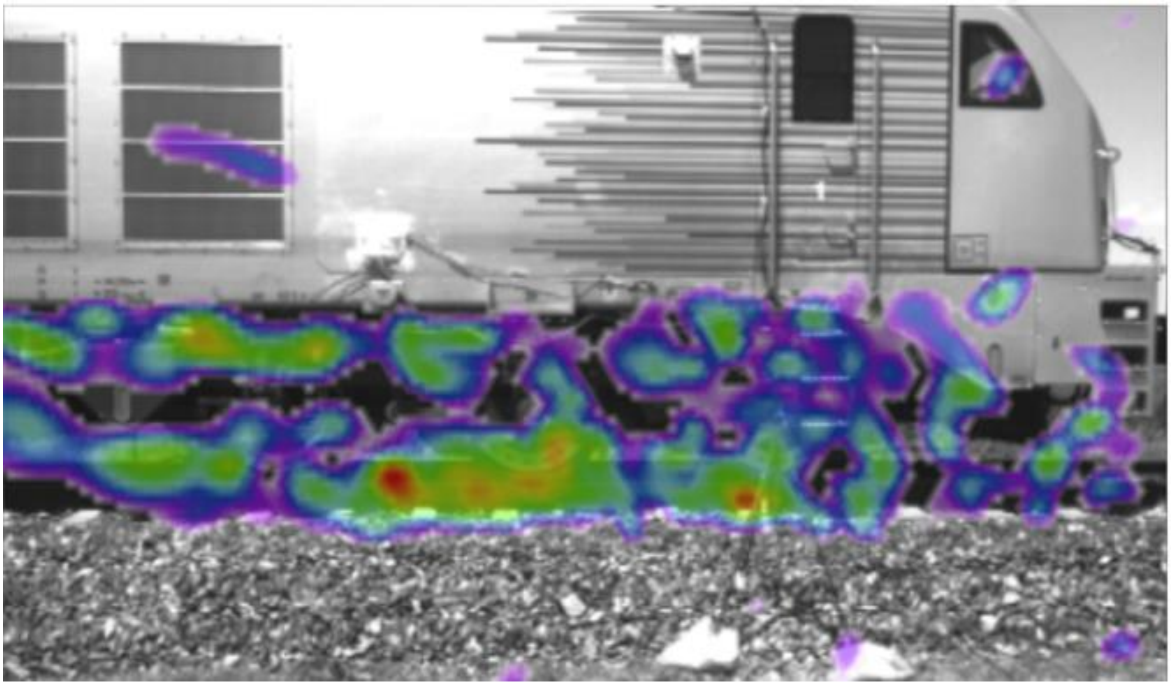

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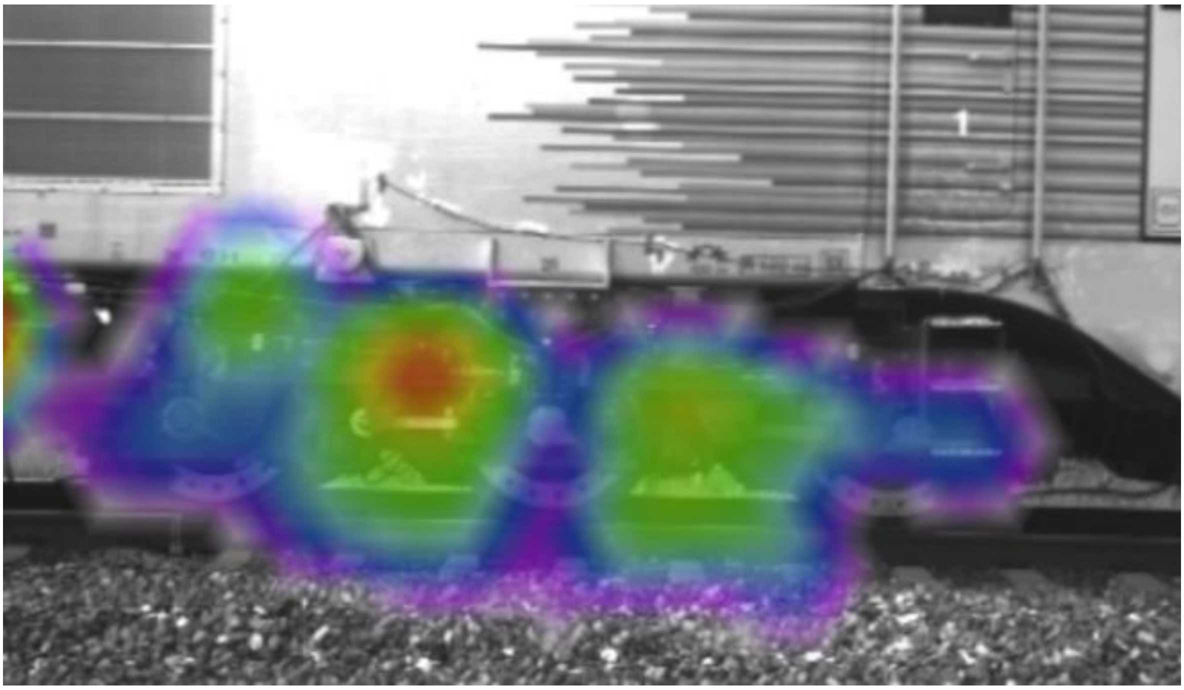

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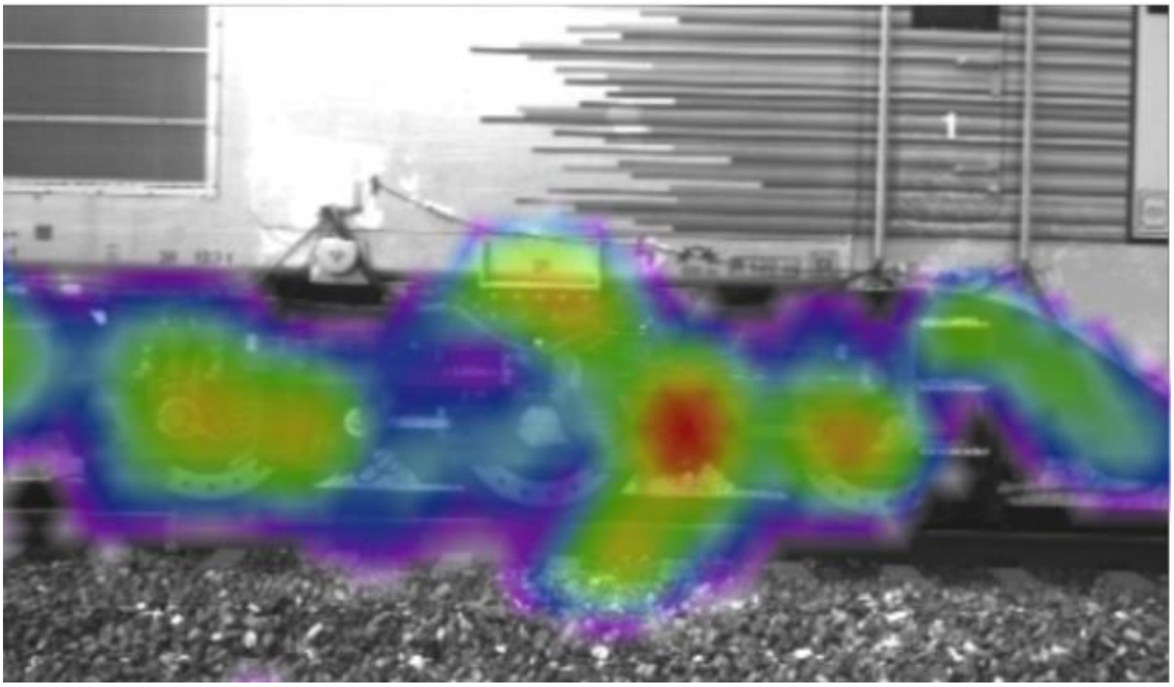

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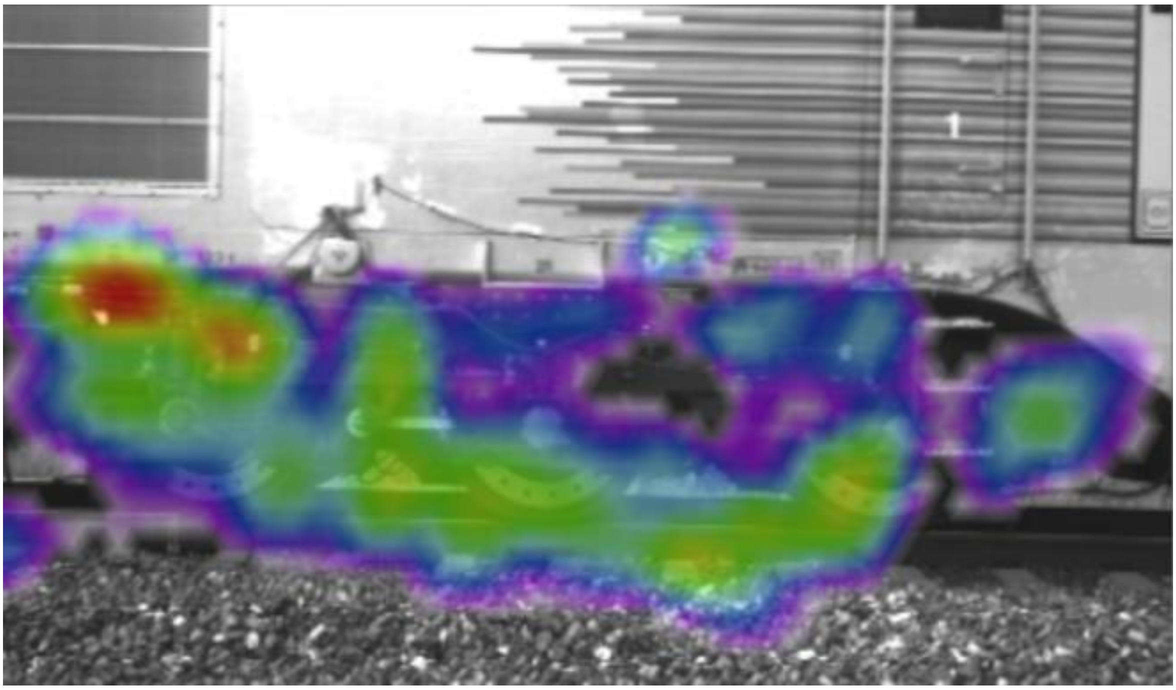

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Shift2Rail
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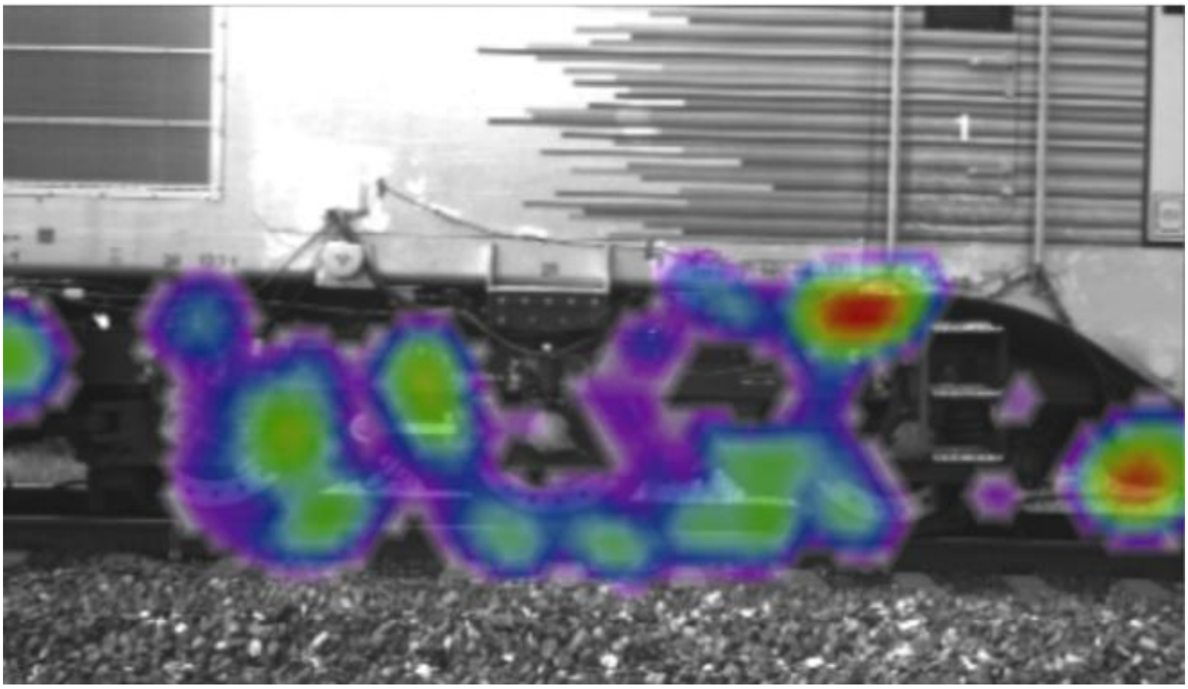

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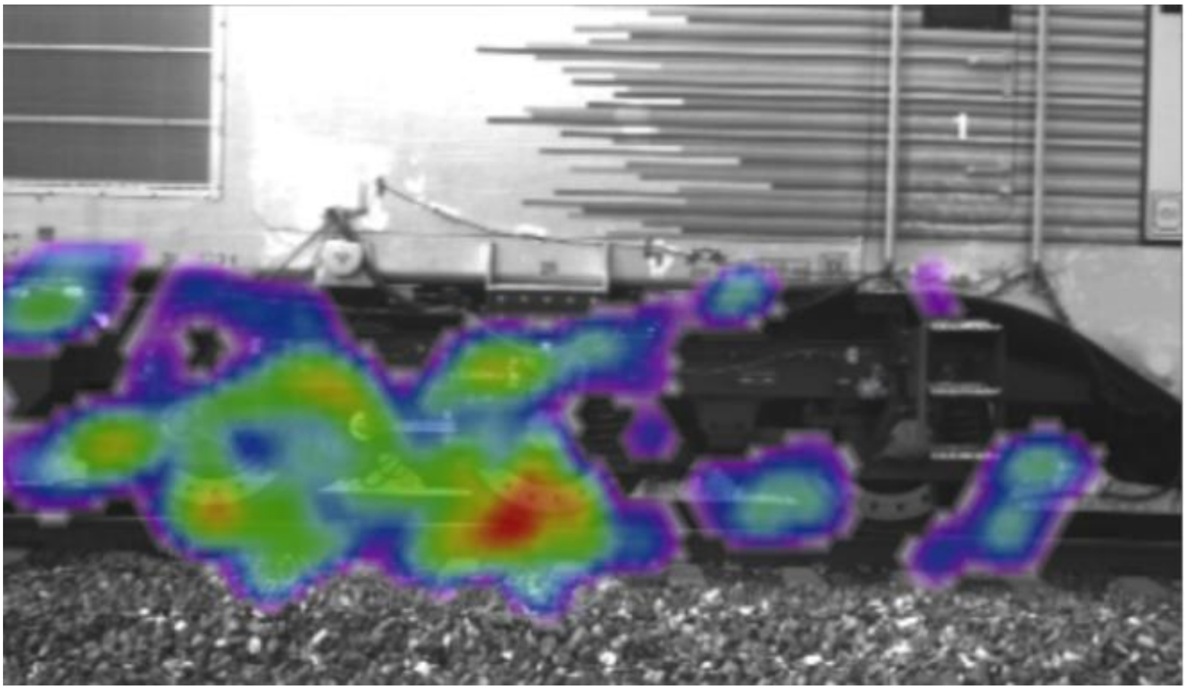

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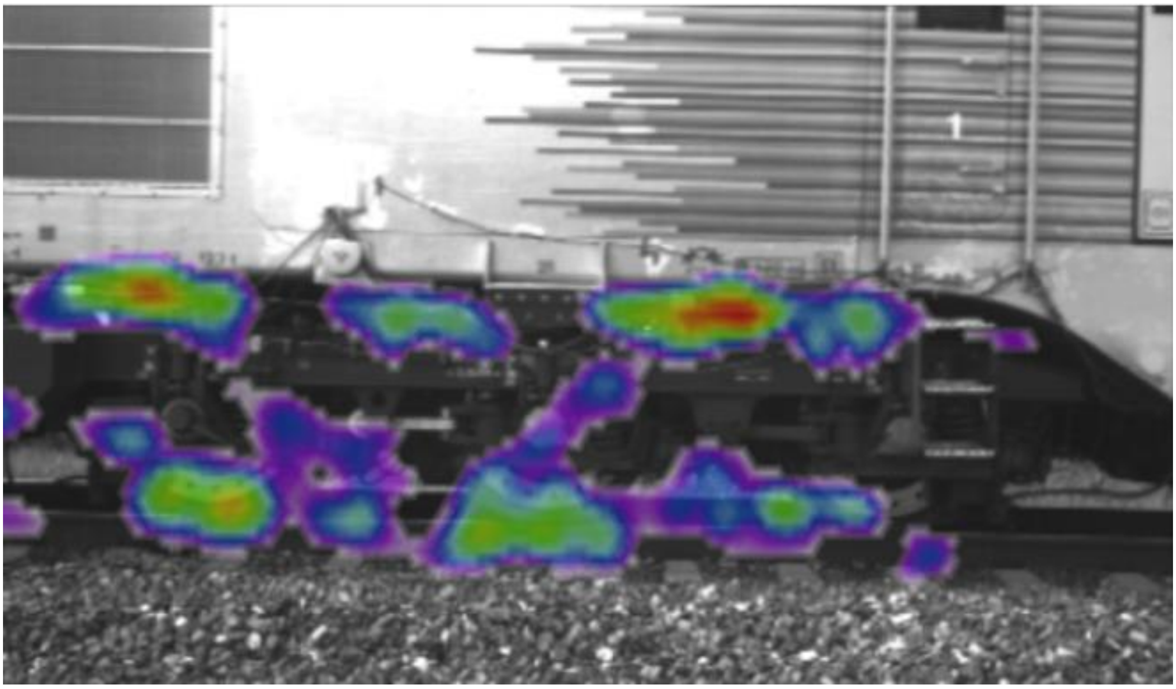

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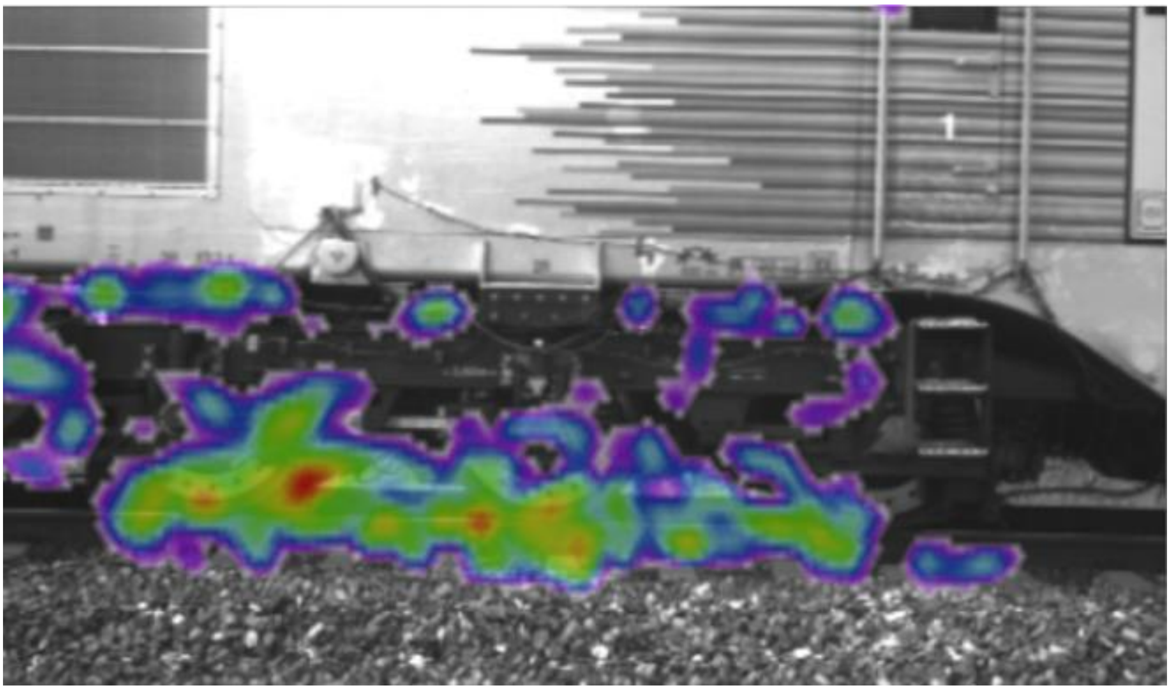

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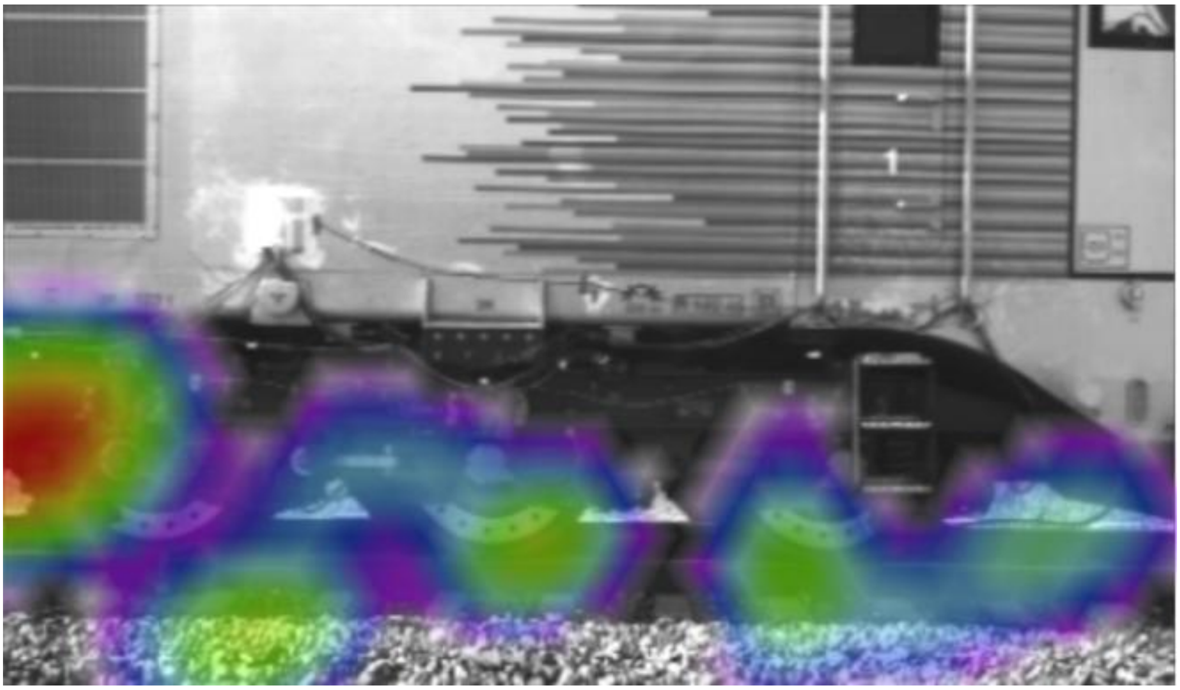

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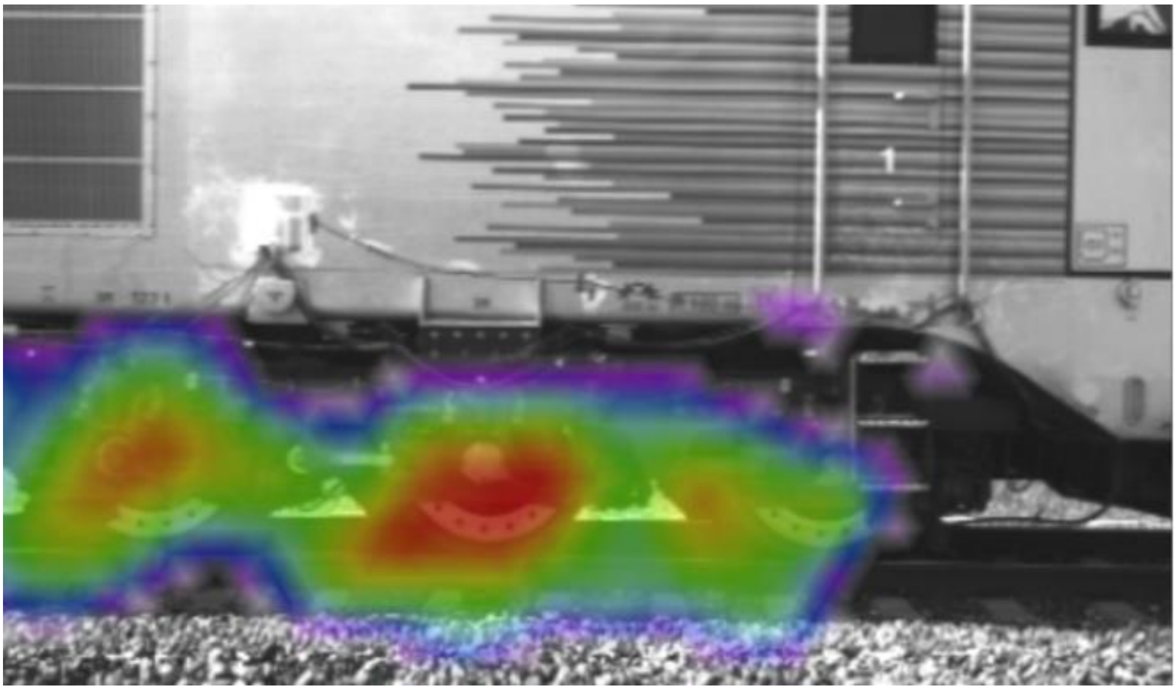

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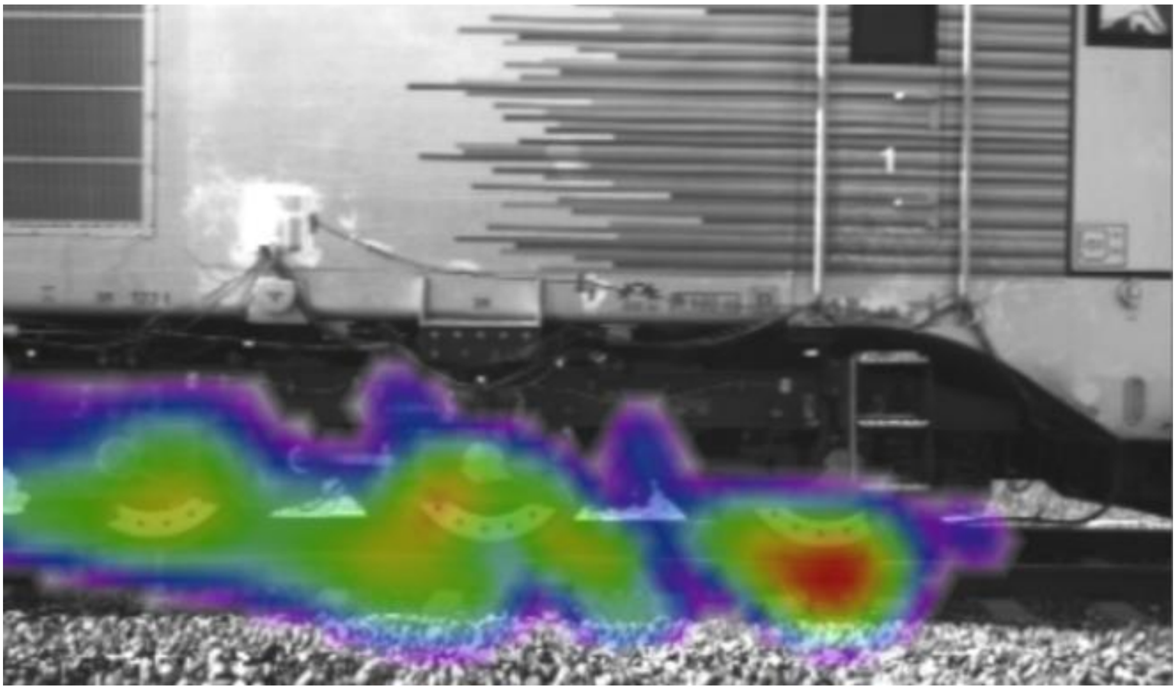

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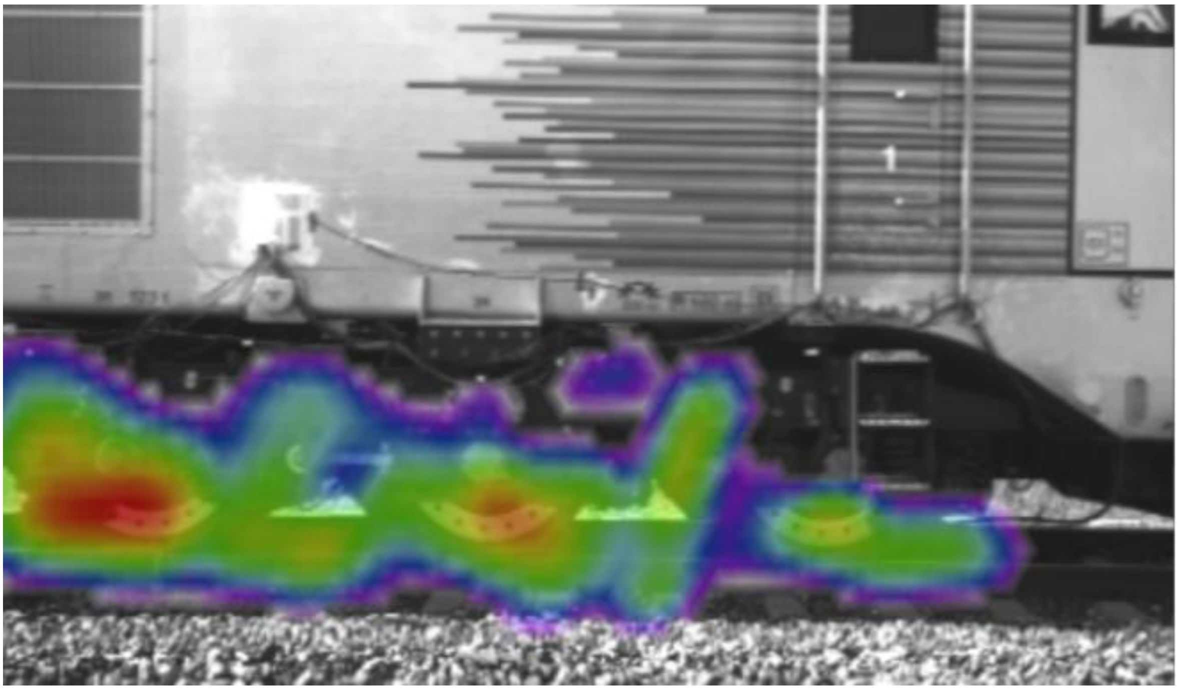

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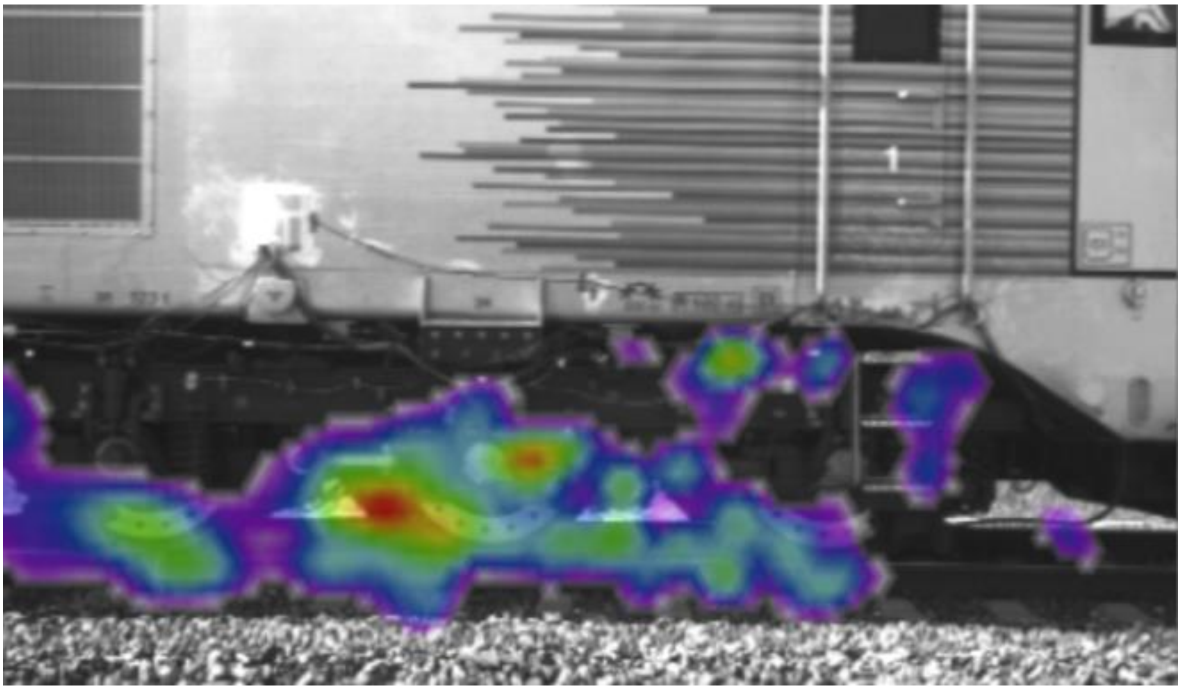

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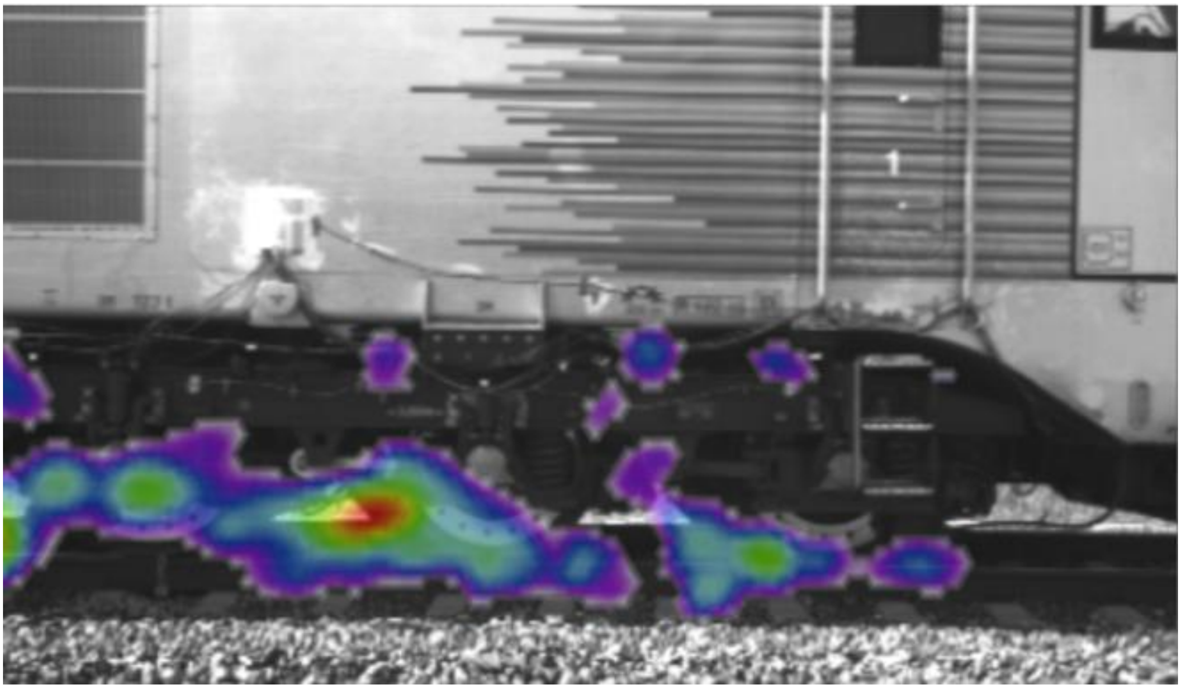

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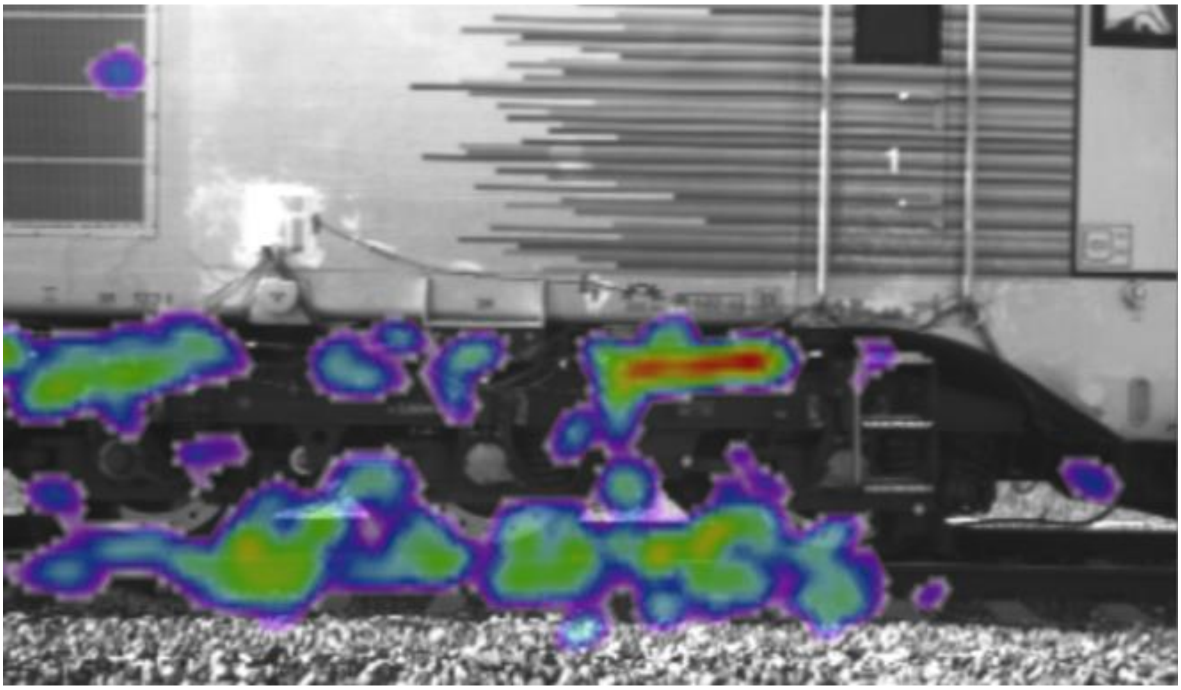

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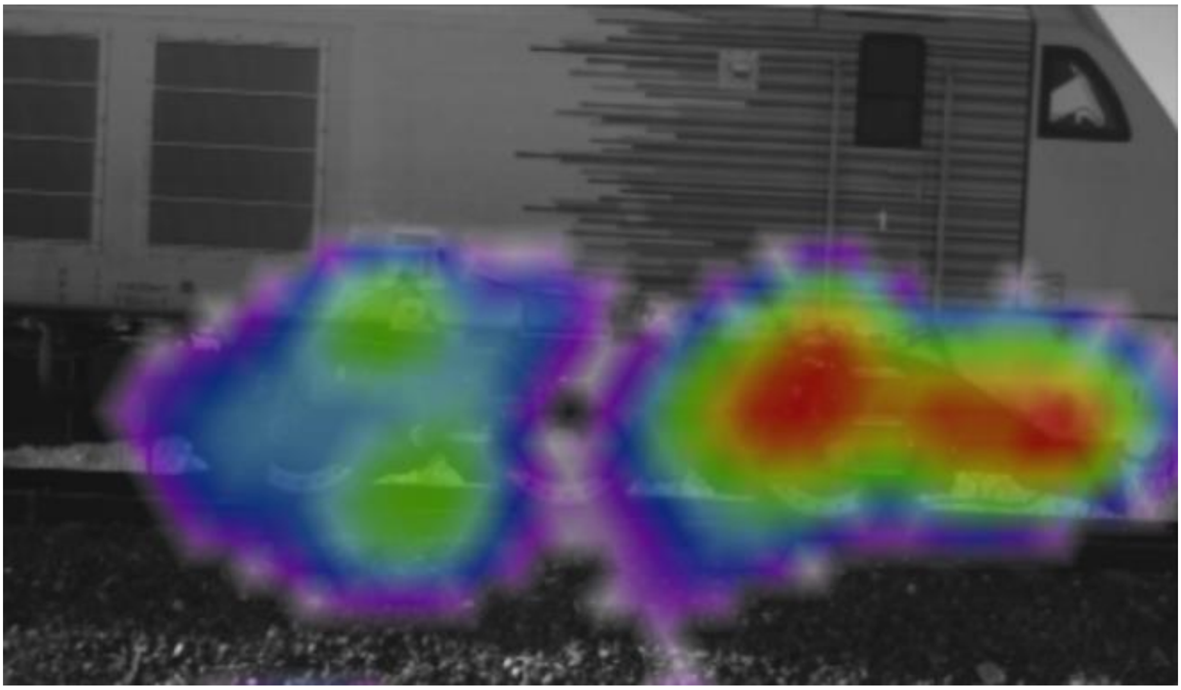

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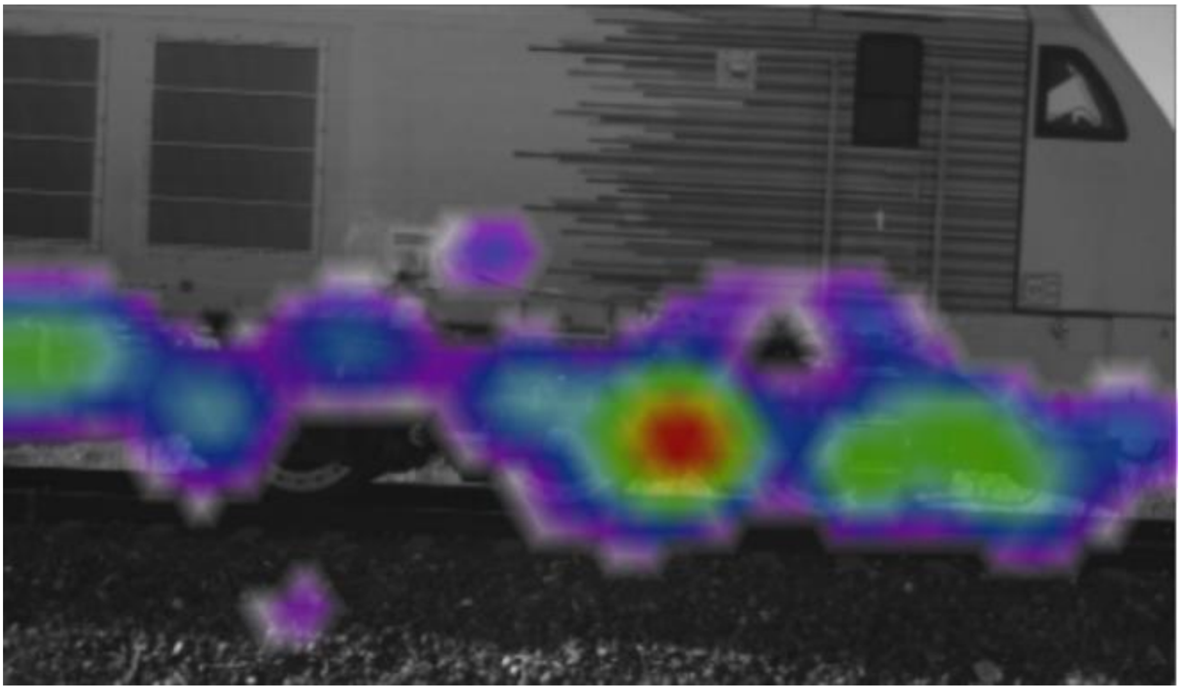
Shift2Rail
Grant Agreement Number: 730811



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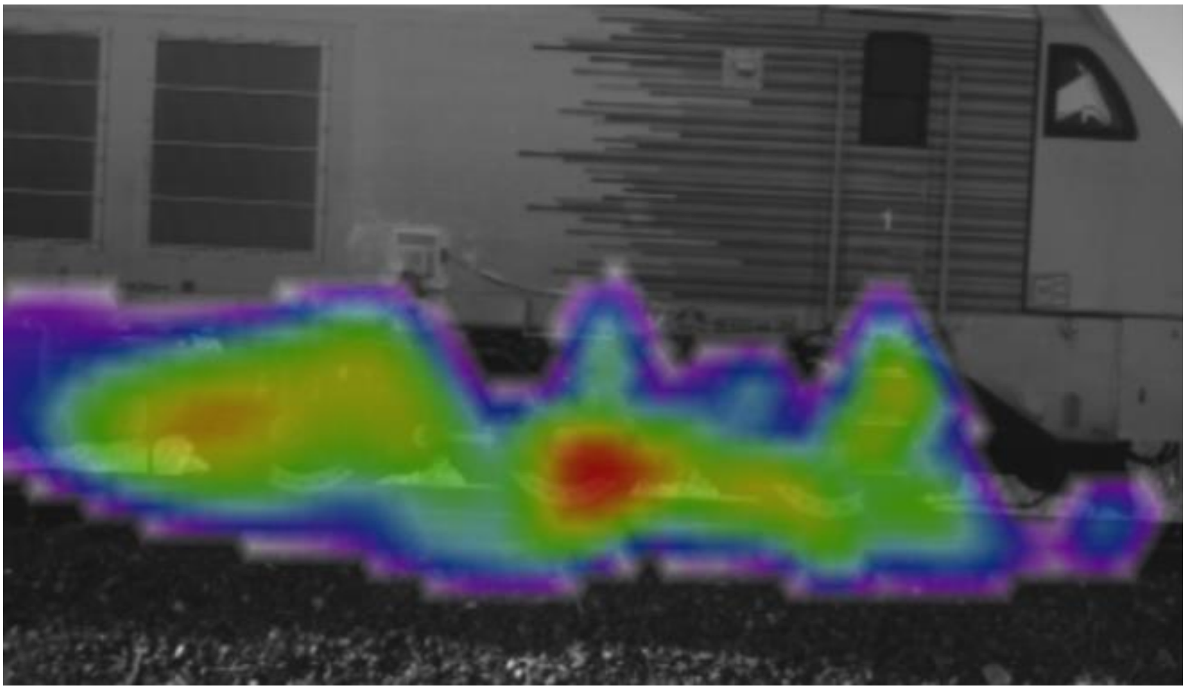

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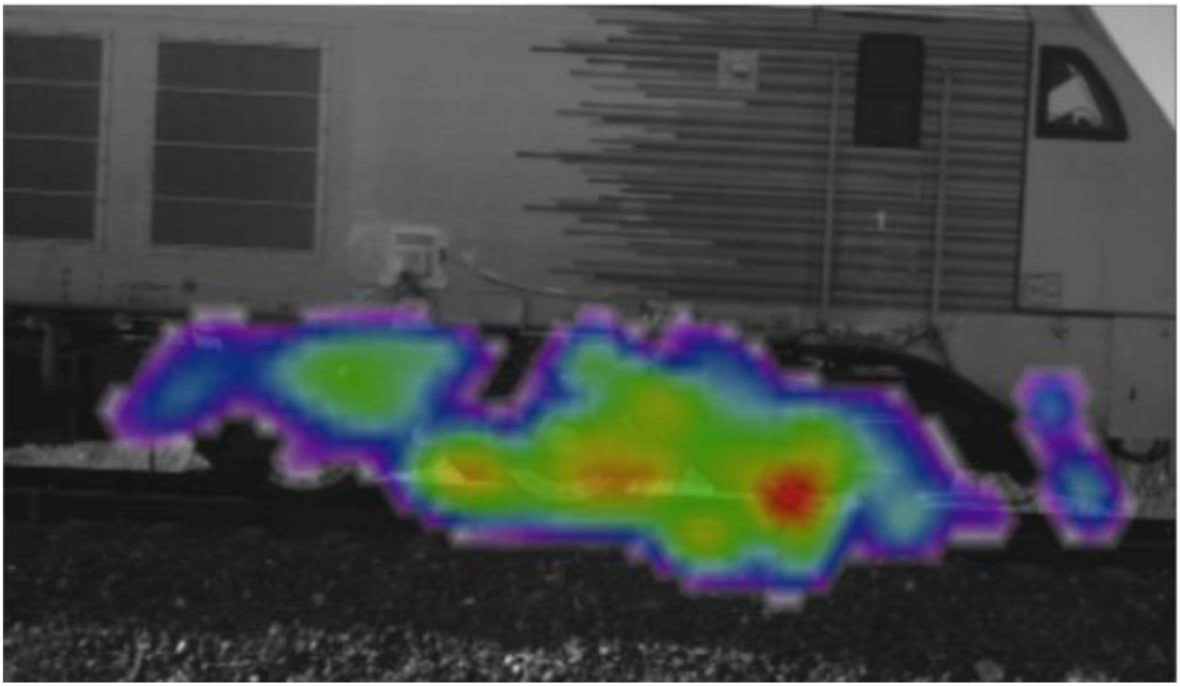

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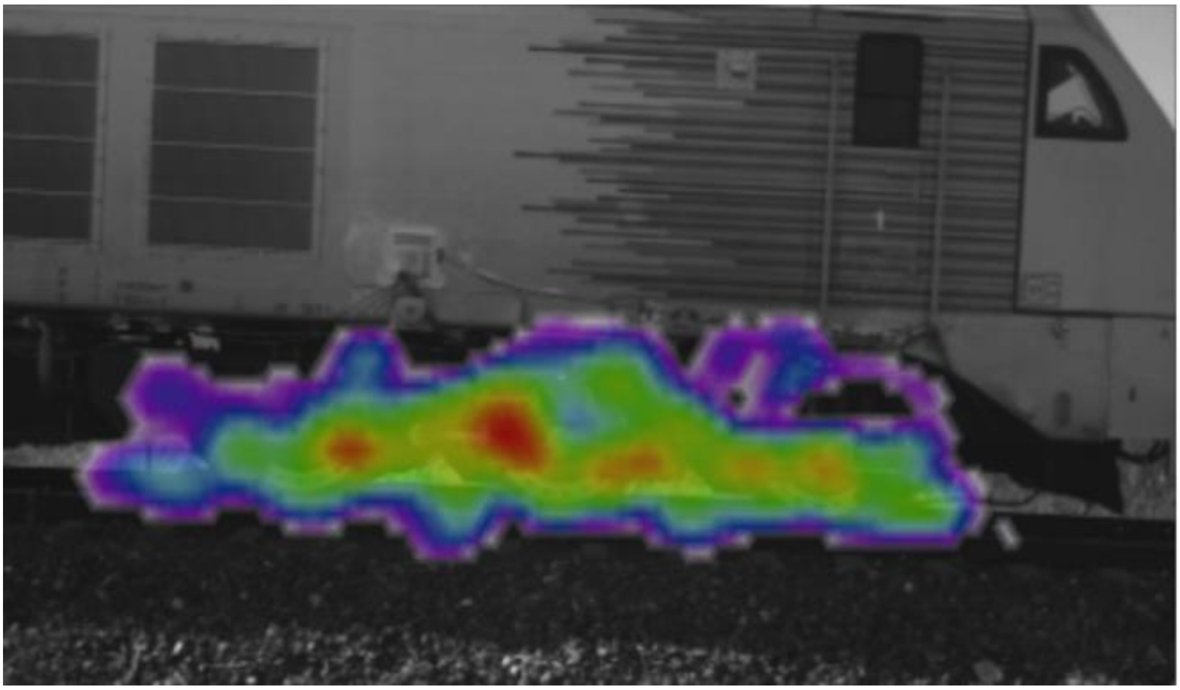

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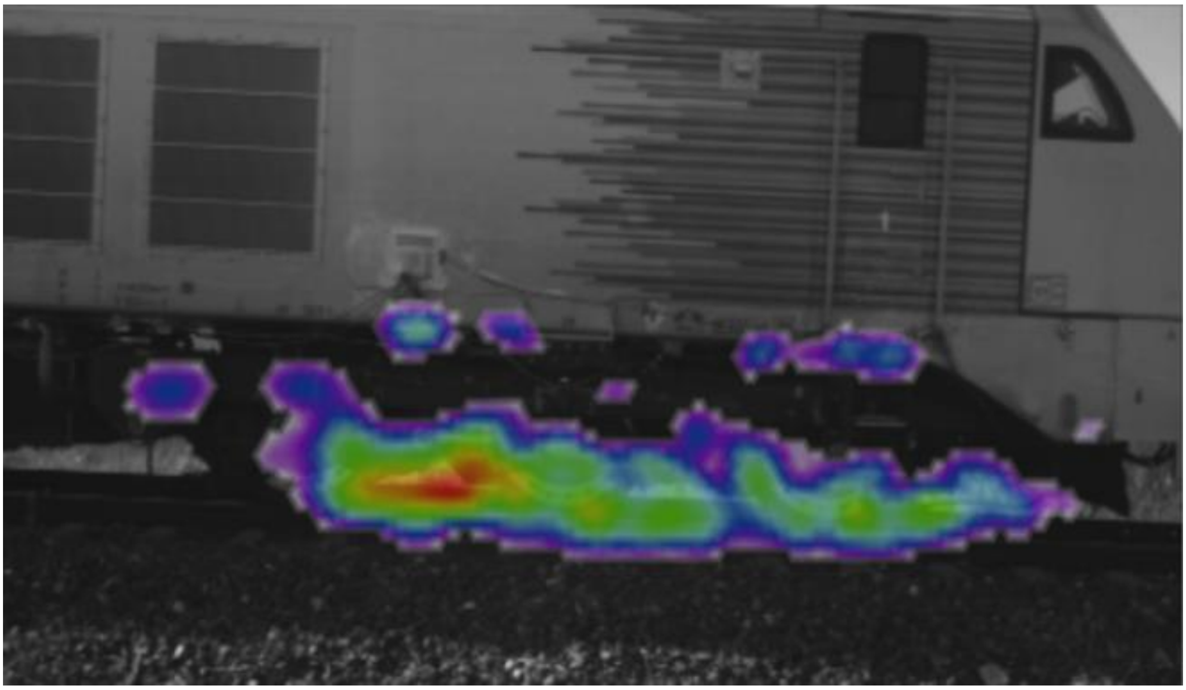

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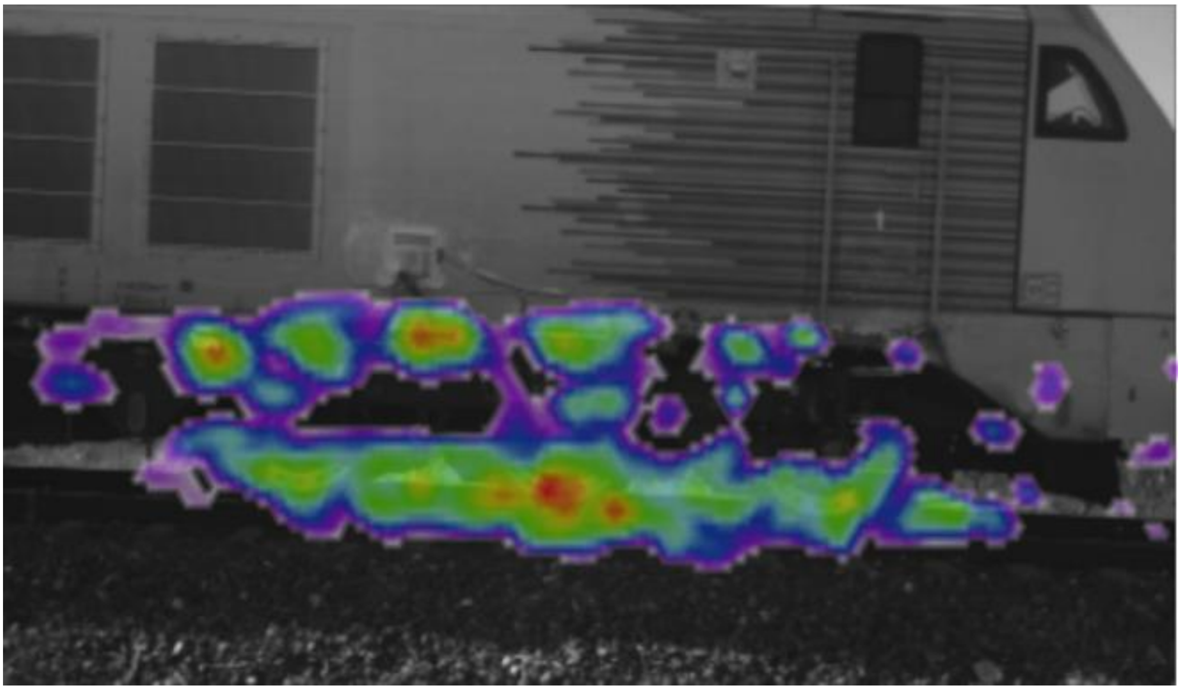

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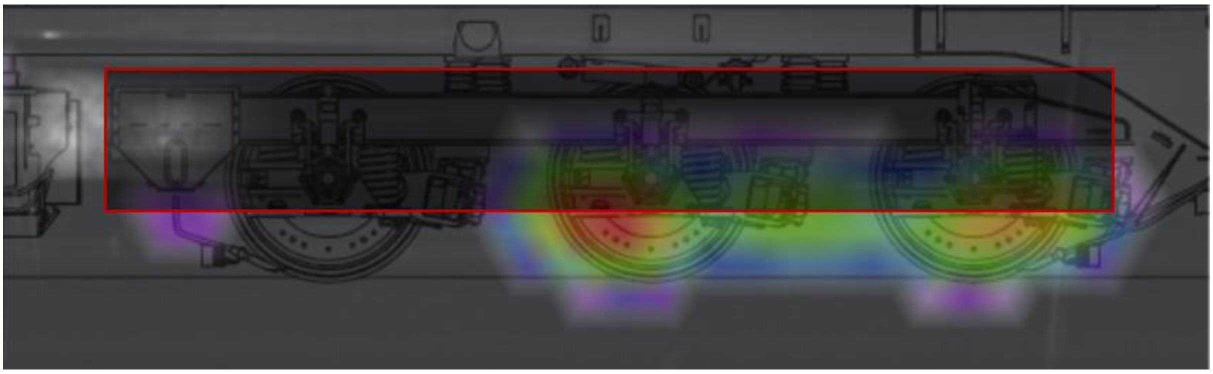
Grant Agreement Number: 730811

2. Lateral skirts measurements results



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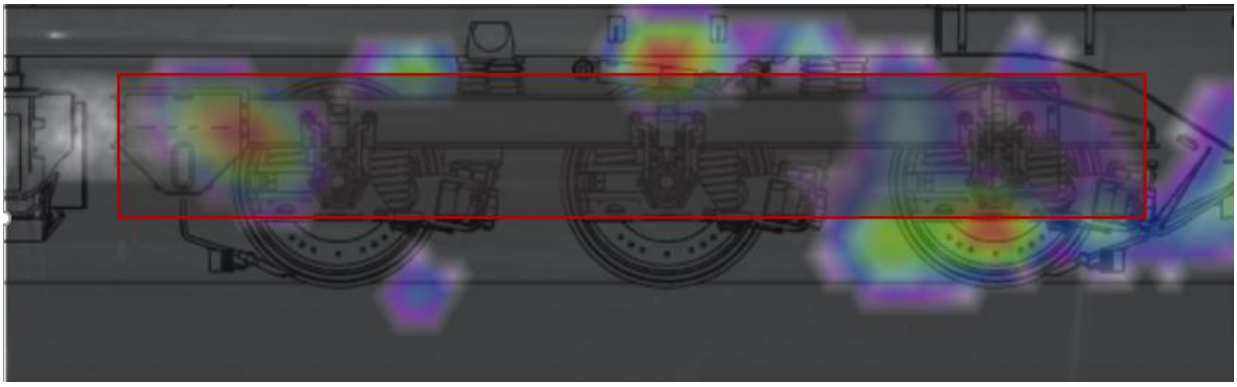

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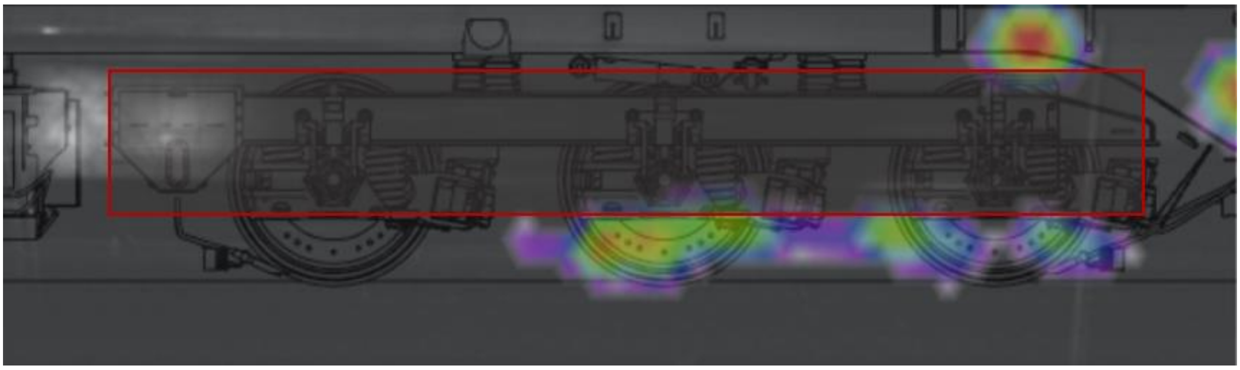

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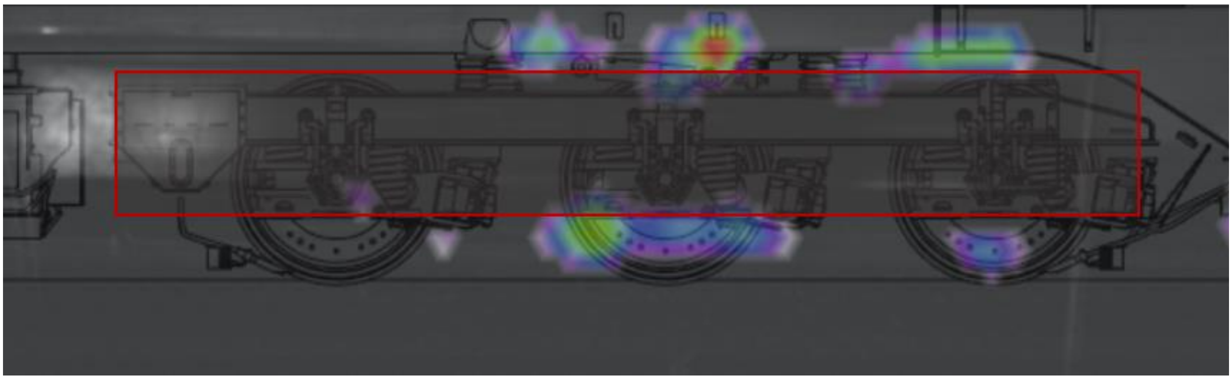

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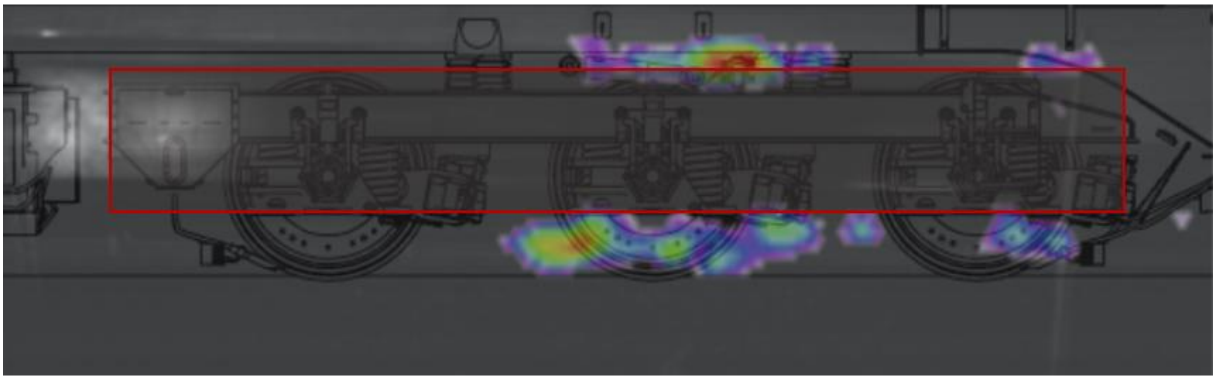

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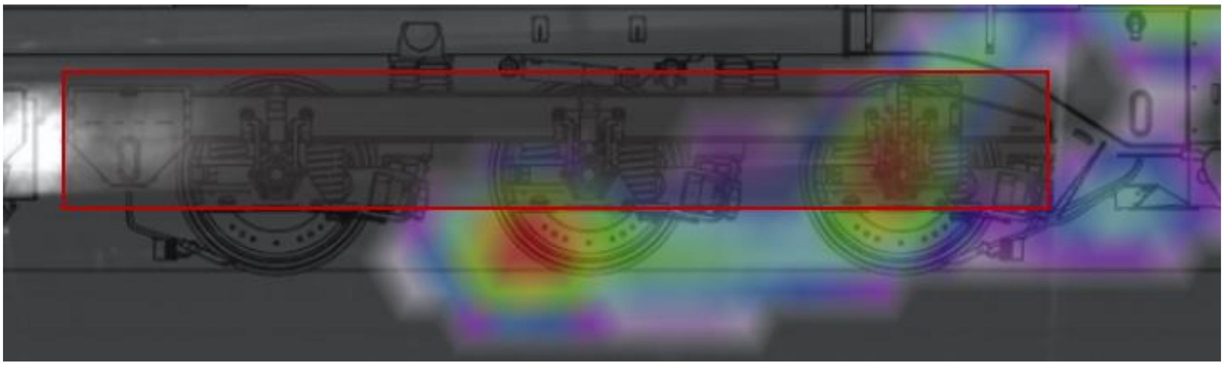

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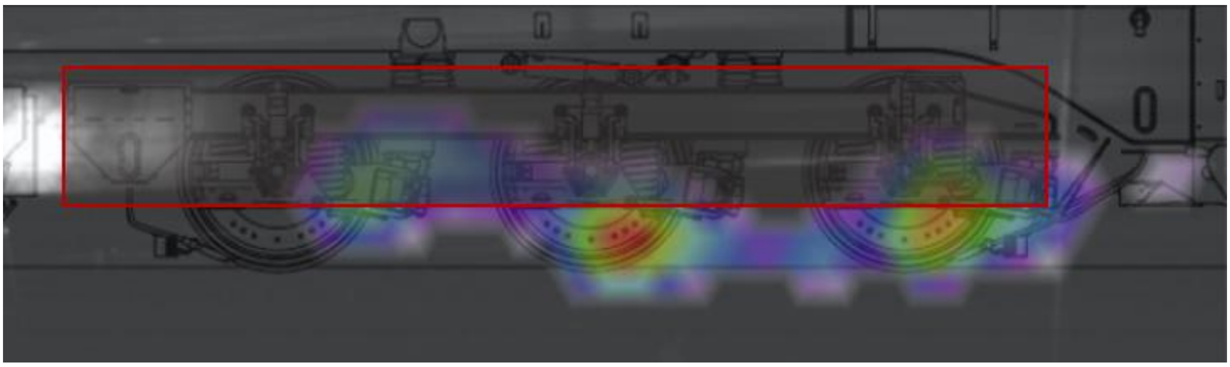

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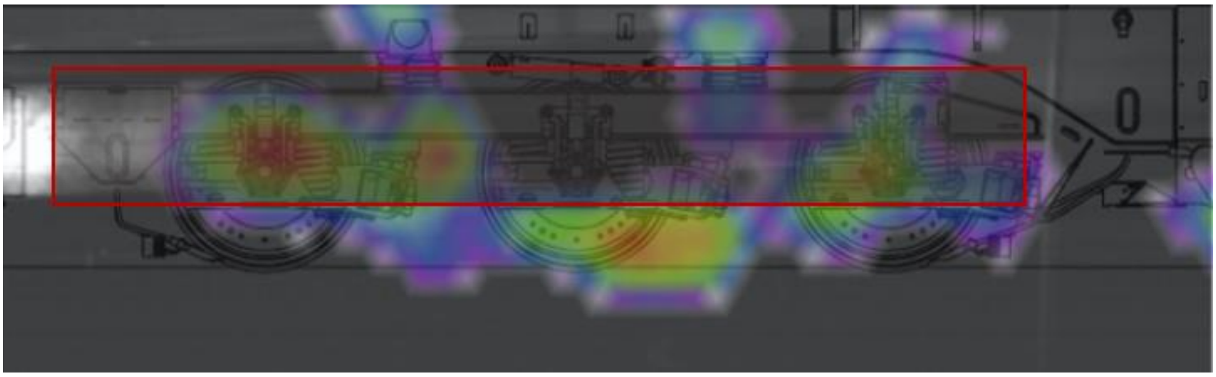

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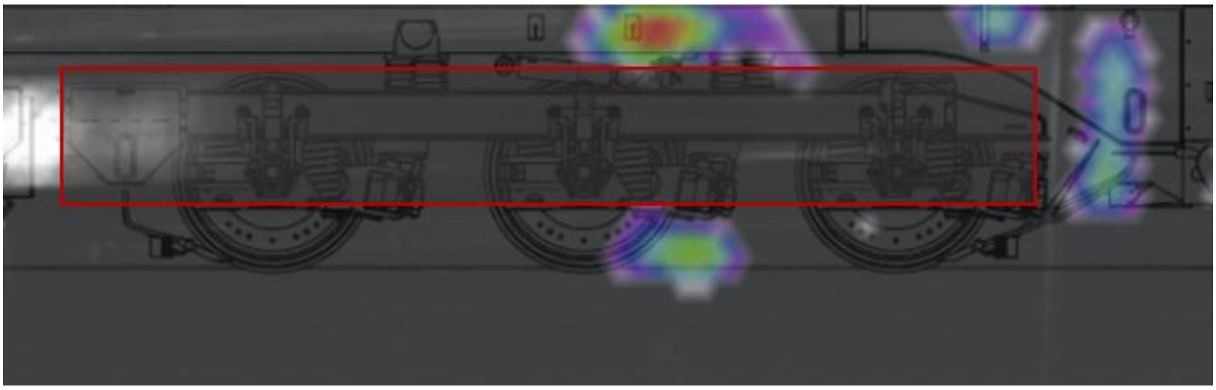

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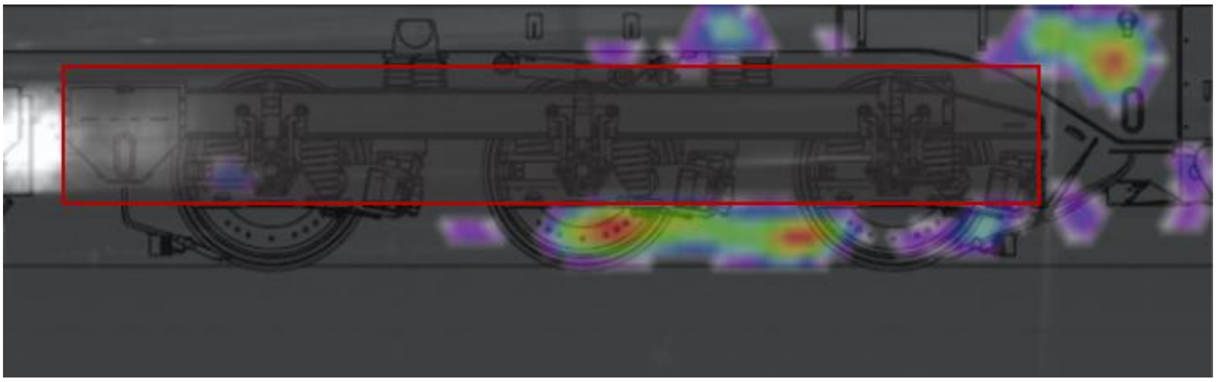

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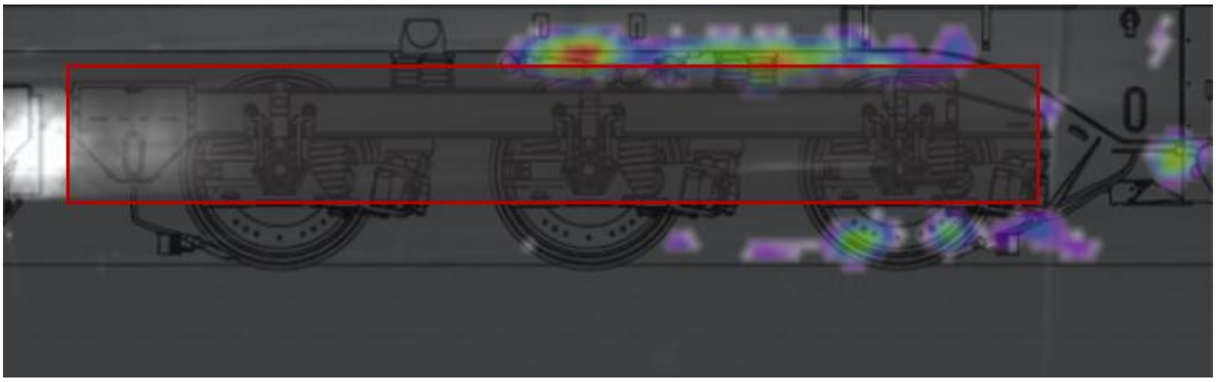
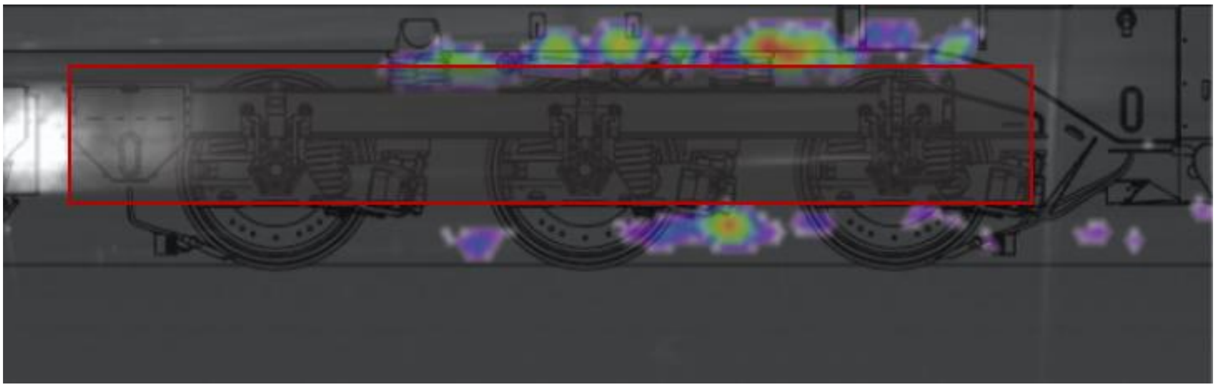


Electric Mode_120kph_3rdOctave3150Hz



Electric Mode_120kph_3rdOctave4000Hz

Grant Agreement
Number: 730811