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Local control of noise and vibration with KELTRACK™ friction modifier and Protector® trackside application: an integrated solution

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Abstract

Wheel squeal is a source of continuing concern for many railroads and transits, as well as for their neighbours. The underlying mechanism for squeal noise has been well understood in the literature for some time. However an integrated abatement method addressing the underlying cause of the problem has not previously been reported.

This paper describes practical experience using a water-based liquid Friction Modifier (KELTRACK™) applied using a top of rail trackside applicator (Portec Protector®). The Friction Modifier and delivery equipment have been co-developed to provide an optimized product/delivery system that gives significant reduction of wheel squeal in curves.

Wheels experiencing lateral creep in curves are subject to roll–slip oscillations as a result of the frictional characteristics of the interface layer between the wheel and rail. These roll–slip oscillations are amplified in the wheel web leading to the familiar squeal. Providing a thin film of material between the wheel and rail with positive friction characteristics can both in theory and practice greatly reduce the magnitude of these oscillations. The controlled intermediate friction characteristics of KELTRACK™ allow the material to be delivered to the top of both rails without compromising traction or braking.

The positive friction aspects of the friction modifier are illustrated by published laboratory studies. Delivery of KELTRACK™ to the contact patch is achieved with a proprietary top of rail electric trackside applicator, the Portec Protector®. The material is delivered to the top of both rails for optimum friction control.

The integrated product/equipment technology is now successfully controlling noise at more than twenty transit sites. Typical sound level reduction is 10–15 dB, in some cases as high as 20 dB, depending on the initial sound level. Two case studies are presented illustrating the technology.

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1. Introduction

1.1. Mechanism of wheel squeal

The literature on the mechanism of wheel squeal is extensive [1–9]. The accepted model involves top of rail frictional instability under lateral creep conditions leading to excitation of out of plane wheel bending oscillations. These are radiated and heard as squeal. The most recent developments involve more rigorous mathematical modelling by Heckl [1]. The starting point for squeal is the lateral creep forces that occur as a bogie goes through a curve. As the creep force increases, the friction decreases, leading to frictional instability.

A critical component in all the modelling work is the requirement that beyond the point of creep saturation, further increases in creep levels lead to a progressively lower coefficient of friction. This is known as negative friction, referring to the slope of the friction creep curve at saturated creep conditions. In more general tribological terms, this would be equated to changes in sliding velocity, rather than the railroad term creep. This leads to stick–slip (or more accurately roll–slip) oscillations between the wheel and the rail, which are amplified in the wheel web.

Despite the clear scientific consensus that wheel squeal is predominantly a result of top of rail frictional conditions, a perception remains within railroads that squeal is due to contact between the wheel flange and the gauge face of the rail. While flange (and back of flange) contact is clearly *not* the major mechanism of wheel squeal, the wide variation in squeal under nominally similar conditions may reflect the varying contribution of this effect under different circumstances.

1.2. Means of mitigating wheel squeal

Although there is extensive literature on the mechanism of wheel squeal, there is less objective comparative data on alternatives for squeal abatement. Remington [8] categorized mitigation methods into three general categories:

- Change the wheel/rail surface conditions to avoid conditions of decreasing coefficient of friction with increasing creep.
- Change track layout and bogie design such that the steady lateral creep is always small enough to ensure that the coefficient of friction will be increasing monotonically with increasing creep.
- Damp the wheels so as to overcome the negative damping introduced by the lateral friction forces at the wheel–rail interface. In the scientific literature most attention has been given to wheel damping. In practical terms however, this solution does not appear to have been widely adopted.

Friction modifiers represent an available practical option to modify the friction characteristics according to Remington's first category. A comparison of a range of on-board, trackwork, and wayside options for mitigating wheel squeal has also been published [10].

2. Friction modifier characteristics

Friction modifiers (high positive friction) materials are available as solid sticks and more recently as water-based liquids. The latter contain a suspension of active friction modifier materials, together with a number of other functional additives. Two key characteristics of top of rail friction modifiers (whether liquid or solid) are:

- Control of top of rail friction at an intermediate value. This is typically around 0.35 but occasionally can be somewhat higher or lower depending on characteristics of the rail, and of the amount and type of contaminants present.
- Positive friction attributes, meaning a thin film of the material shows *increasing* coefficient of friction with either creep or sliding velocity. The term positive refers to the slope of the friction/creep beyond the point of creep saturation [11].

Liquid friction modifiers can be clearly distinguished from lubricants, which reduce top of rail coefficient of friction to a level <0.2 on application [12]. Numerous measurements in the laboratory and the field have confirmed the intermediate coefficient of friction provided by the thin dry film of friction modifier. This intermediate coefficient of friction is key to allowing the material to be used on top of the rail. At this coefficient of friction braking and traction systems are not impaired.

The positive friction attributes of HPF friction modifiers have been established in a number of laboratory studies [12,13]. An excellent illustration of this for liquid HPF (KELTRACK™) has been published [14]. Figures from this paper are reproduced in Figs. 1 and 2 for lateral creep. The data were generated using a two-roller stand with a 1/5-scaled truck.

For the dry, or clean contact condition (steel-on-steel, with iron oxides likely to be present), the small but distinct negative slope can be seen clearly beyond the point of creep saturation. With liquid friction modifier applied, the friction–creep curve shows a clearly positive slope throughout the creep range examined.

The role of the friction modifier in terms of wheel squeal mitigation is to provide material with positive friction characteristics between the wheel and the rail. To be effective there must be sufficient friction modifier present to overcome negative friction characteristics of all the materials present in the interfacial layer (third body). As modelling and experimental work by Hou et al. [15] has shown, many of the materials present in the interfacial layer (including iron oxides and sand) have negative friction attributes.

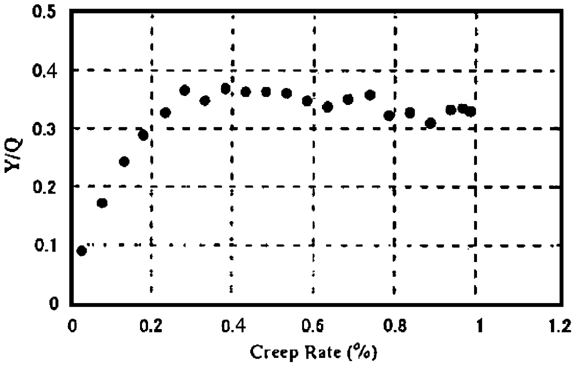

Fig. 1. Clean contact (dry). (Reproduced from [14] with permission.)

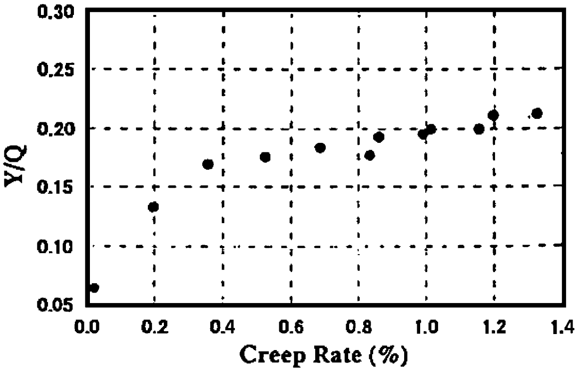

Fig. 2. Liquid HPF film applied. (Reproduced from [14] with permission.)

3. Trackside applicator for top of rail friction modifier delivery

To take full advantage of the benefits of top of rail friction modifiers a key requirement is a reliable and effective means of delivering the material along the top of the railhead.

The “wayside” or “trackside” approach offers several advantages:

1. Application is site specific. The product is only distributed where needed—for example, throughout a noisy curve.
2. Railroads and transits have long been accustomed to using a trackside applicator for gauge-face lubrication. Trackside top of rail equipment is similar to gauge face equipment in many regards, and thus is already familiar.
3. Implementation is relatively easy compared to alternative approaches.

The fundamental challenges of developing a trackside machine for distributing liquid friction modifiers are:

1. Depositing a liquid material on the top of the railhead in a way that will allow it to stay in place until train wheels can pick up the material and carry it down into the curve.
2. Depositing this liquid such that it can be effectively transferred to the treads of the wheels without significant wastage.
3. Distributing the friction modifier in a way that will optimize the process of carry-down.
4. Precisely controlling the rate of distribution so that the friction modifier can achieve the desired objectives with the least amount of material required.

3.1. *“Protector® ” series equipment*

Introduced in the mid-1980s, the Protector® had been developed as a simple but very effective trackside delivery system. A wheel sensor detects each wheel of a passing train without physical contact. The sensor signal is connected to electrical controls, which are easily adjusted for maximum control over distribution.

The control system provides two adjustments for directing distribution:

1. *Wheel count setting*. This control feature adjusts the frequency of distribution. Distribution is activated after the chosen number of wheels has been counted.
2. *Pump duration*. This control feature adjusts the duration of the distribution (pumping time).

Using these two simple controls, the distribution rate can be fine-tuned to different track conditions and other local variables.

The control system is used to direct a simple but positive pump arrangement. An electric motor is directly coupled through a reduction box to a gear pump.

The Protector® units incorporate a dual-compartment tank. One section houses all of the system components, while the other holds the friction modifier material. In the component section, the motor/pump is mounted to the separation wall, near the bottom, such that material is drawn directly into the pump inlet. Sloped walls in the material section help guide the material effectively into the pump. The Protector® is available with either AC or DC solar power sources.

From the tank, material is pumped through a large diameter (for minimal pressure drop) supply hose to a central distribution point located between the rails. From this point, material is directed uniformly to each of the distribution manifolds. Each supply hose incorporates a valve so that distribution can be further controlled as required. Check valves in the supply hoses ensure that the distribution system stays full and will not drain back (Figs. 3 and 4).

The distribution bars are mounted using clamps that mount on the base of the rails. The bars are mounted so that distribution flows onto the railhead from the field side of the rail. The internal passageways in each bar are balanced so that material flows uniformly along the length of the bar.

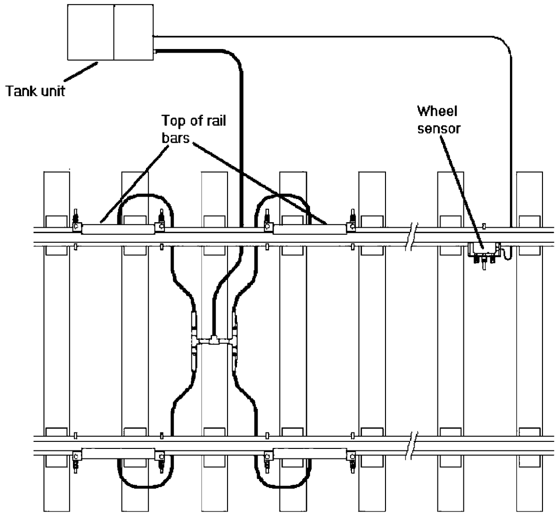

Fig. 3. Schematic of trackside application layout.

Fig. 4. Tank assembly.

Fig. 5. Friction modifier flowing from bars.

3.2. Application mechanism

A view of the liquid friction modifier emerging from the top of rail bars is shown in Fig. 5. Once the material has reached the contact patch on the top of rail, it is picked up by the passing wheel treads. The heat of the wheel–rail interface rapidly evaporates the water content of the friction modifier. Generally, no liquid material is observed further than about 6 m downstream of the top of rail bars. Beyond this point a dry thin film with controlled frictional attributes is deposited. Carry down of the dry film continues through the length of the curve. Product carry-down up to

400 m has been observed based on push tribometer measurements. Generally, one trackside unit is required for each curve.

4. Experimental results

Results from two separate installations are described. Similar results have been obtained on a variety of other transit systems.

4.1. Case 1—West Coast North American Light Rail

This system operates U2a double-ended articulated vehicles manufactured by Siemens Transportation Systems (Fig. 6). These vehicles have lightweight steel trucks with DUEWAG type mono-motors. The suspension systems comprise rubber chevron springs (primary) and coil spring system (secondary). The length of the vehicle over the couplers is 24.3 m. The unloaded car weight is 35 000 kg.

The test was carried out on the inside (inbound) track leaving a station on the system's mainline (Fig. 6). The site consisted of a 76 m long curve with a 25 m radius. The track at this location is zero gradient and is constructed of 52 kg worn rail embedded in the roadway. The trains tested travelled from a stationary position at a nearby station in-bound through the curve towards downtown. Top of rail bars were installed in imbedded track (Fig. 7).

Prior to installation of the trackside applicator, this transit system had been applying friction modifier manually using a modified paint roller. Hence the trial protocol included comparison between dry top of rail conditions, manual application results, and results with trackside application.

Sound levels were measured using a Bruel & Kjaer model 2260, set at A-weighting. The microphone was fitted with a foam windscreen and set on a tripod on the outside of the curve

Fig. 6. Vehicles traversing test curve, Case 1.

Fig. 7. Top of rail bars installed on imbedded track.

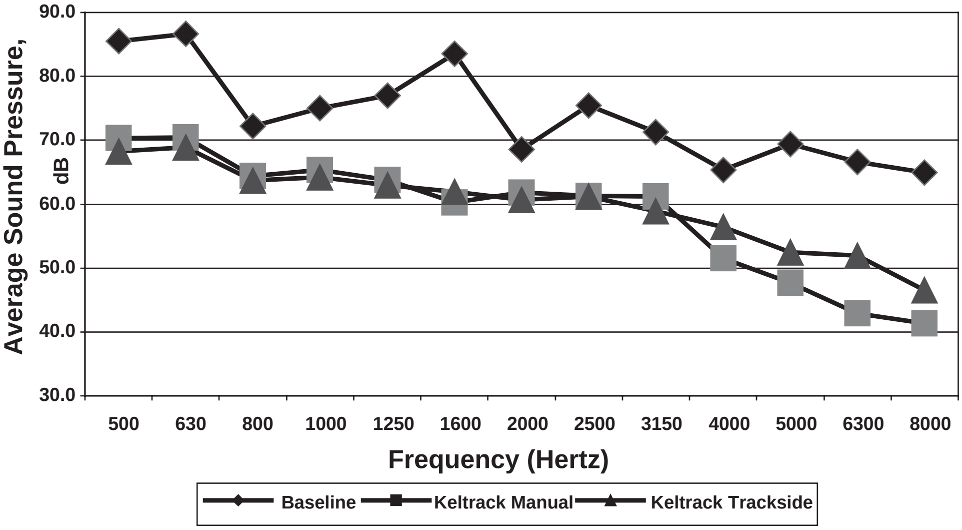

Fig. 8. Average sound pressure versus frequency.

up-track from the nearby station in the centre of the curve. The sound level range was pre-set to between 50 and 130 dB. The tripod was placed 7.5 m from the centre of the track, with the microphone 1.2 m above the height of the rail. The sound level meter was programmed for event recording, enabling the instrument to automatically measure and store the event data. Average sound pressure versus frequency for the dry, manual and trackside application appear in Fig. 8.

The baseline (dry) condition shows a peak at 1600 Hz, consistent with wheel resonance from top of rail stick–slip. Both manual and trackside applicator results show a dramatic reduction in

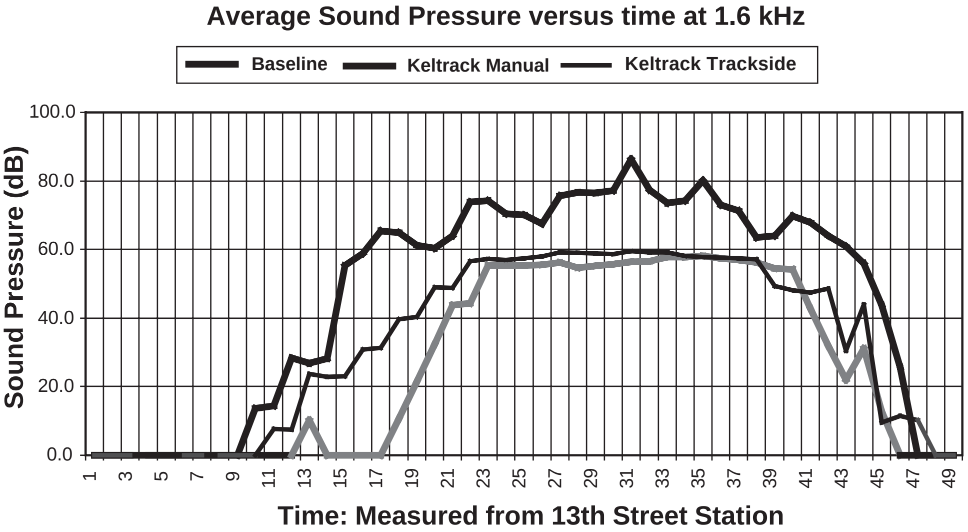

Fig. 9. Time-based squeal frequency response: 1.6 kHz.

average sound pressures across the full frequency range (Fig. 8). The average dry (baseline) maximum A-weighted sound level was 102.6 dB for data containing both two- and four-car tram baseline runs. Following manual application of friction modifier the average peak sound level was 88.3 dB, a reduction of 14.3 dB for the same trams. For automatic trackside application, an average maximum A-weighted sound level of 89.8 dB was recorded after start-up of the Protector® units, a reduction of 12.8 dB compared to the baseline value.

Time-based analysis further illustrates the effectiveness of this system. Figs. 9 and 10 show average sound pressures at two different squeal range frequencies. The starting (zero) point for these graphs is the departure of the vehicle from the nearby station. Fig. 9 shows the average for all four-vehicle trams at the 1.6 kHz, and Fig 10 shows equivalent data at 2.5 kHz. The measurements begin at the same location at the nearby station, and continue until the tram has passed through the curve. There are slight differences in the duration of the runs due to small variations in the train speeds.

Apart from the much lower sound levels, another noteworthy feature is the longer time with no measurable sound level at the frequencies of interest with the friction modifier applied.

4.2. Case 2—Japanese tram system

This system employs light trams in a high frequency service, where wheel squeal levels were an annoyance to local residents. KELTRACK™ friction modifier was applied to the top of rail at the entrance to the test curve spiral, using a Protector-III™ wayside applicator. The controls were set to apply the product every eight axle passes for a 0.75 s duration. The test used single car trams during regular revenue service, which comprised two types of four axle transit vehicles. The trams operated in equivalent conditions, during control (baseline) and test runs.

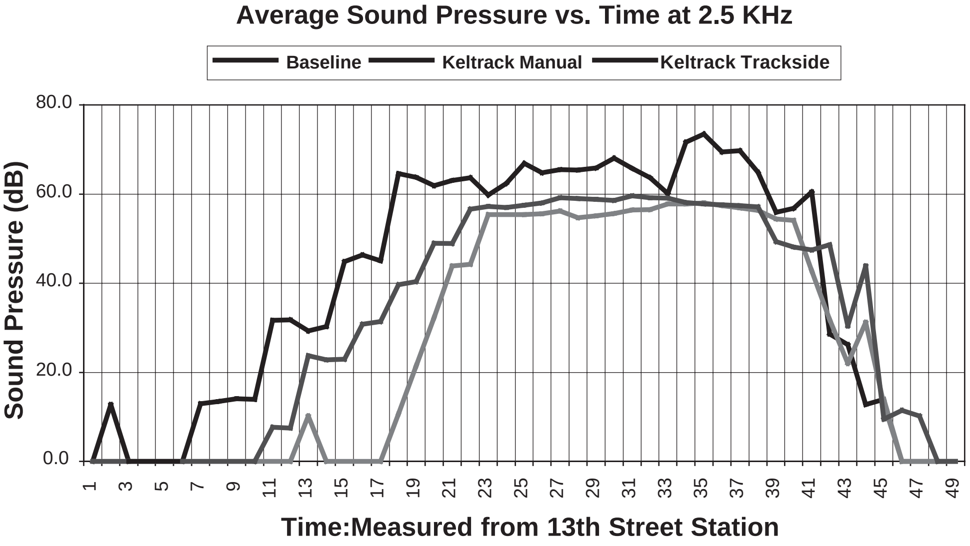

Fig. 10. Time-based squeal frequency response: 2.5 kHz.

Fig. 11. Test cars traversing test curve, Case 2.

The test area was located on the inside, inbound track approaching a station on the mainline of this system (Fig. 11). The site consists of a 160 m radius curve with a 3% downgrade. A restraining rail is present on the low rail throughout the curve. The trams tested travelled downhill through the test area curve and stopped at the station shortly after the curve.

The trial included evaluation of friction modifier application to the low rail only as well as to the top of both rails. Sound level measurements were performed using a Rion NA-24 instrument, set for A-weighting. The microphone was fitted with a foam windscreen and set on a tripod on the inside of the curve up track from the station. The sound level range was pre-set to between 70 and

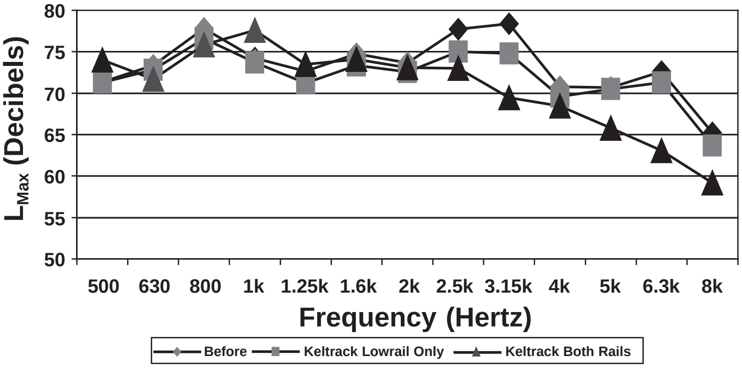

Fig. 12. Sound pressure versus frequency, Case 2.

110 dB. The tripod was placed 1.82 m from the low rail of the inner (inbound) track, with the microphone 1.05 m above the height of the rail. The sound level meter was programmed for event recording, enabling the instrument to automatically measure and store the event data. The logged data was then downloaded via a serial interface to a chart recorder.

Results are shown in Fig 12, illustrating the noise frequency spectrum for baseline (control) and liquid friction modifier application, on low rail only as well as on both rails.

A significant reduction is observed for sound levels in the squeal region. The control spectrum does not show any particular dominant frequency, suggesting that a large portion of this noise may be due to flange or back of flange contact.

Larger reductions in noise are achieved when the friction modifier is applied to both rails compared to just the low rail. This may be because application to both rails provides a controlled coefficient of friction on both rails. Reducing the coefficient of friction on the top of the high rail is expected to reduce the flanging force contact. The result will be to reduce the noise originating by this mechanism. The average maximum sound levels for the baseline runs were 90.8 dB. Three distinct sounds were audible during the baseline runs: top of rail contact, flange contact sand back of flange contact. After the application of the friction modifier to the low rail only, the average peak sound level was reduced to 87.3 dB. With application to the top of both rails, the average peak noise level was reduced to 85.7 dB. Maximum sound levels were reduced by 5.1 dB, relative to the average baseline values.

The average measured baseline maximum noise level is lower than is typical for this system because of slower than normal speeds on some of the measured trams. With more typical vehicle speeds, an average reduction of 10 dB would be expected. Car ♯7016 for example was recorded for all three controls. It showed a 10.1 dB reduction in peak noise level for the baseline conditions compared to the case of application of the friction modifier on the top of both rails.

5. Conclusions

KELTRACK™ liquid friction modifier applied through a Protector® top of rail trackside application system is an effective and practical means of substantially reducing wheel squeal. The system is suitable for noise control on specific curves. Application of the friction modifier greatly reduces wheel squeal by changing the fundamental friction characteristics (negative to positive friction), and provides an intermediate sliding coefficient of friction so that traction and braking are not affected.

This new technology now provides a cost-effective solution to squeal on some of the world's largest and most sophisticated transit systems.

Acknowledgements

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