Field verification of curving noise mechanisms
This report sets out observations of the curving noise mechanism as evidenced by field testing at Beecroft; and following from that, establishes the direction for curving noise mitigation trials.
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Executive summary

Curving noise, including squeal and flanging noise, is one of the main causes of community concern and pollution complaints from rail operations, particularly from freight train operations, in developed areas in Australia. Curving noise is the least understood railway noise source. It behaves like a ‘random’ phenomenon owing to its complex origin. As a result, it is difficult to predict its occurrence, intensity and frequency, and take corresponding effective mitigation measures. This report focuses on the verification of the mechanisms which drive curving noise. The verification is based on field measurements with unique access to an operating rail system, rather than relying on laboratory-based test rigs and scale model systems. In this way, this research has an excellent opportunity to advance the understanding of curve squeal and deliver positive outcomes in terms of control measures for the industry.

Specific field measurements were conducted at Beecroft in Sydney to assist in understanding the mechanisms of curving noise generation. The field measurements involved noise, angle of attack, rail forces, and rail vibration measurements. In total, 131 squeal noise events with the noise level exceeding 100 dB(A) were detected from 242 freight train pass-bys (with 44,707 passing axles) in the field test. The majority (67%) of these squeal events occurred in a frequency range of 1000 to 3000 Hz. The noise level of squeal events was generally 10–20 dB(A) higher than flanging noise.

The relationship between curving noise and angle of attack was analysed and concluded as below:

1. Curve squeal noise only occurs when the lateral creepage (or angle of attack) exceeds a certain level, around 7~10 mrad.
2. Not only the magnitude, but also the likelihood, of squeal noise appears to increase with increasing creepage (or angle of attack).
3. The presence of high angles of attack (>10 mrad) may not necessarily generate squeal noise, but may induce a low-frequency oscillation in a frequency range of 100 to 240Hz, which corresponds to the typical short-wavelength corrugation. This finding helps to explain why some wheels with a high angle of attack are not observed to create audible squeal noise. It also suggests that high lateral creepage (or angle of attack) can be the main contribution to the formation of corrugation at sharp curves.
4. The occurrence of flanging noise has no obvious relationship to angle of attack.

Some phenomenon that are either contradictory to, or cannot be explained by, the current curving noise theory were observed with the occurrence of curve squeal: apparent amplitude modulation in noise, rail vibration and rail forces; constant phase shift between the vertical and lateral rail motions; slight difference in the excited frequencies between the vertical and lateral rail vibrations; and more frequent squeal noise generated from the outer wheel–high rail than from the low rail. These observations appear to support the notion that squeal noise is a beating phenomenon, and can be explained by the mode-coupling mechanism.

Given the observations of force, vibration and noise oscillations in the field measurements at Beecroft, it seems that mechanisms other than negative friction-induced instability may be significant to the generation of squeal noise. As a result, the mode-coupling instability theory is introduced and used to explain those observations. Further field study conducted at other sites is suggested to test and consolidate the mode-coupling theory discovered from the Beecroft observation. In coping with the new squeal theory, corresponding squeal noise mitigation methods are proposed and need to be developed and trialled in the next stage.

Valid curving noise monitoring strategies depending on various requirements have been demonstrated.
A single microphone-based noise monitoring system can meet the requirements of general curving noise monitoring and detection.

With additional wheel sensors, the identified curving noise can be associated with a particular wheel axle.

To further separate the curving noise into inner or outer wheel events, vibrations of the low- and high rail, in either vertical or lateral direction, are required to be measured.

By adding a portable angle of attack detector, the system will be able to identify bogies with abnormal curving behaviour which lead to excessive lateral force, vibration and noise, and direct subsequent rolling stock-based maintenance efforts.
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Abbreviations and acronyms

AoA  Angle of attack
ARTC  Australia Rail Track Corporation
CRC  Cooperative Research Centre
L/V  Lateral/vertical force ratio
QR  Queensland Rail
1. **Introduction**

This report is the result of the work conducted for project R1-105, Improved Noise Management – Part A Curving Noise.

1.1. **Aims of the project**

The aims of project R1-105 Part A were to:

- analyse extensive field data collected by rail agencies to assist in understanding and validating the complex mechanism of curving noise generation
- develop validated strategies for monitoring and control of curving noise for existing networks.

This report sets out observations of the curving noise mechanism as evidenced by field testing at Beecroft; and following from that, establishes the direction for curving noise mitigation trials.

1.2. **Background**

Rail has good environmental credentials, but noise remains a significant environmental weakness. It is the dominant cause of community concern and the most frequent source of pollution complaints for existing rail networks, forcing rail agencies to take a reactive approach to noise management. At present, noise pollution significantly undermines rail’s efficiency and public image. It is also the primary trigger for mitigation treatments for rail expansion projects.

Many aspects of rail noise management can be improved without the need for research. Indeed, the last 15 years have seen significant improvement in noise performance as a result of development guidelines, maintenance standards and management policies.

However, certain noise issues remain a significant issue for the industry, despite substantial effort to date. Curving noise is an acute noise issue at certain locations on existing networks. This is due to its complex origin, the unintegrated or unbalanced approach to its control between above and below rail parties, and the absence of a clear understanding behind the benefits to both parties.

This project is aimed at understanding the apparent random nature of its generation.

1.3. **Structure of the report**

The project aim and background is introduced in Chapter 1. Chapter 2 gives a brief review of current knowledge on curving noise, and the research questions addressed in this study. Chapter 3 describes the experimental design, instrumentation and field data collection. The detailed data analysis and results are presented in Chapters 4 to 8, which includes the curving noise monitoring and identification, the relationship between curving noise and angle of attack (AoA), the influence of friction on curving noise, the influence of AoA on the formation of corrugation, and the exploration of curving squeal noise as a beating phenomenon and a potential new curve squeal noise mechanism.
2. A review of current knowledge on curving noise

A brief review of the current knowledge on curving noise is given below, followed by a discussion and potential research questions. For more details of the technical background and research questions, please refer to the previous project report (Dwight & Jiang, 2009).

2.1. Curving noise mechanisms

Curving noise is the least understood railway noise, and has been the subject of research for several decades.

In the 1970s, Rudd (1976) identified three possible mechanisms of the pure tonal squeal noise owing to the stick-slip behaviour at the wheel and rail contact region: lateral creepage between the wheel tread and the top of rail head; wheel flange rubbing on the rail gauge face; and longitudinal creepage on the wheel tread due to differential slip between inner and outer wheels of an axle. The longitudinal differential slip between inner and outer wheels was discounted by Rudd, considering the elastic deformations of inner and outer wheel tread when a torque is applied can compensate for the differential velocity, and independently driven wheels still squeal in tight curves. Flange rubbing has also been discounted as a sufficient source mechanism for squeal on account of some experimental observations: lubricating only the outer flange did not eliminate squeal; laboratory experiments show that flange contact reduces the level of squeal noise (Remington, 1985); and the inner wheels were found by von Stappenbeck to generate squeal (1954). Therefore, lateral creepage was suggested by Rudd and others (Remington, 1985; Finburg, 1990; Heckl, 1999; Nelson, 1997) to be the main cause of squeal noise, particularly from the leading inner wheel of a bogie.

Various models have been developed since Rudd (Finburg, 1990; Périard, 1998; de Beer, Janssens & Kooijman, 2003; Heckl, 1999; Thompson et al., 2003), based on the assumption that the friction force decreases after saturation with increasing creep velocity, or the friction creep curve has a negative slope when the creep exceeds a certain threshold. Based on this theory, there are three prerequisites for the occurrence of curve squeal noise:

1. high lateral creepage (or AoA equivalently) greater than 10 mrad or so
2. characteristic friction condition with falling slope of the creepage–creeps force curve, or negative friction damping
3. low wheel internal damping relative to the negative friction damping.

However, this theory has not been fully validated, and has been accepted with surprisingly little evidence to support it. In fact, there are other mechanisms existing that may lead to friction instability, such as the mode-coupling mechanism (Johnson, 2002). The negative slope of the friction–creep curve is necessary in order to generate instability in the case of a single degree of freedom system. In the case of curve squeal, the assumption of a single degree of freedom system may not be reasonable. If a multiple degree of freedom system is assumed, there is no need for a negative slope friction–creep curve in order to create instability leading to squeal. In fact, squeal noise has been observed without a falling friction characteristic in a laboratory test rig (Koch et al., 2006), which suggests that other mechanisms may play a part in curve squeal noise generation.

Apart from the lateral creepage between the wheel tread and rail top surface, the longitudinal creepage may also play a part in squeal noise generation. For example, it has been found that a 2% longitudinal creepage results in the elimination of squeal on a 1:3 scale twin-roller test rig (Monk-Steel et al., 2006). This may explain why some locomotive wheels do not squeal in the presence of
both high lateral and longitudinal creepage. It should be noted that the leading axle of a locomotive bogie has a higher AoA than a normal wagon bogie due to its longer wheelbase.

Although flange contact has been discounted as a squeal noise mechanism (Rudd, 1976), it appears to be responsible for another form of curving noise, a more broadband ‘flanging noise’ (Thompson, 2009; Eadie & Santoro, 2003; Jiang & Dwight, 2006). It is unclear why the flange contact generates the more broadband curving noise. Furthermore, the contact between the back of the wheel flange and check rails at the leading inner rail wheel can generate squeal-like noise (radial modes of the wheel) at extremely high levels (130–145 dB reported close to the wheel) (Thompson, 2009).

Some mitigation treatments for curving noise have been implemented with varying success. Wheel damping treatments are commonly used to reduce the occurrence of squeal noise (Nelson, 1997). Rail damping treatments appear to be successful in some installations; however, the mechanism is not understood (Thompson, 2009; p. 338). Top-of-rail friction modification is reported to be effective in some locations, but only partially effective (if at all) in other locations (Anderson & Wheatley, 2007).

The difficulty in studying curving noise not only arises from understanding and modelling its mechanism, but also lies in the difficulty in controlling laboratory tests and field observations. The occurrence of friction-induced vibration is often very sporadic in apparently identical conditions in laboratory tests, relative to currently accepted mechanism parameter values, and the frequency of vibration can change for no apparent reason (Johnson, 2002; Thompson et al., 2003). The observation of curve squeal in the field is more problematic, owing to its apparent sensitivity to a range of parameters, such as normal load, train speed, humidity, temperature, track geometry, and wheel and rail wear.

2.2. Australian experiences in dealing with curving noise

The Australian rail industry is active in wheel squeal and curve noise research and mitigation. For example:

- extensive applied research and development, including the trialling of numerous mitigation techniques over the last 10 to 15 years such as the application of friction modifier to the top of the rail (Anderson & Wheatley, 2007; Kerr et al., 1998; Kerr & Lak, 1999; Powell, 1998), water spray (Powell, 1998), improved freight bogie steering through centre-bowl lubrication (Anderson & Wheatley, 2007; Powell, 1998), passenger train wheels damping (fin and ring damps) (Anderson & Wheatley, 2007; Powell, 1998)
- pioneering noise monitoring and detection techniques for curve noise research (including via project 36 in the previous CRC) (Dwight & Jiang, 2009)
- pioneering use of wayside systems to monitor curve noise, including RailSquad noise telescope on a curve in the Adelaide Hills, and TBOGI AoA detector on a curve in Sydney (Cowley et al., 2006)
- promotion of the definition and categorisation of curve noise (Dwight & Jiang, 2009).

But, despite all the above progress, the noise problem remains a significant one. Techniques such as top-of-rail friction modification are effective in some locations, but only partially effective (if at all) in other locations (Anderson & Wheatley, 2007).

2.3. Summary of research questions

Following the literature review conducted at the beginning of the project, and in order to better understand the mechanism for squeal noise generation, a series of research questions was proposed. It is recognised that current industry knowledge allows particular mitigating actions to be devised.
With a better understanding of the mechanism, it is held that more effective, efficient and reliable actions can be devised for specific sites in a cost-effective manner.

Those research questions addressed in this report are:

1. Apart from the generally accepted lateral stick-slip mono-tonal squeal noise generation mechanism, are there any other mechanisms involved under some specific conditions? How are they to be identified? What will be the corresponding treatment methods?
2. What is the nature of the correlation between AoA and curving noise, particularly mono-tonal squeal noise?
3. What is the nature of the correlation between AoA and curving noise, particularly mono-tonal squeal noise?
4. Why is mono-tonal squeal noise apparently inconsistently related to AoA? What are other factors (excluding AoA and friction) that may play a role, e.g. train speed, wheel lateral position, tread and rail profile, longitudinal creepage, flange contact, vehicle weight distribution, wheel diameters and relative diameters, vehicle suspension condition?
5. Is there field evidence for the existence of negative damping at the contact patch when mono-tonal squeal occurs?
6. Under what situations does a top-of-rail friction modifier mitigate the occurrence of mono-tonal squeal noise?
7. What is the optimum procedure for the application of top-of-rail friction modifier (in terms of how much and how frequently the product should be applied, for example)?
8. Does rolling stock maintenance work improve the AoAs (reduced to the normal range), and will that eliminate curving noise?

As mentioned earlier, the relationship between rail damping and curve squeal noise is not known. However, the study of the rail contribution to curve squeal noise is beyond the current project scope and is not included as a research question.
3. Experimental design and field data collection

This study concerns the field verification of the generally accepted mechanisms proposed by Rudd (1976), as resulting in curving noise, through trackside measurements and observations. As discussed in the previous section, there are three prerequisites for the occurrence of mono-tonal curve squeal noise: high AoA; characteristics friction condition with falling slope of the creepage–creeps force curve; and low wheel internal damping relative to the negative friction damping. This theory has been accepted since the work of Rudd, but without comprehensive field tests and validation. Therefore, complete and accurate measurements of those three parameters and noise emission in the field are necessary in order to validate the current curving noise mechanisms.

The measurement plan consists of:
1. selection of a monitoring site
2. instrumentation at the monitoring site
3. data collection
4. data analysis.

3.1. Monitoring site

A curve site located on the main north rail line at Beecroft, Sydney, was selected as the monitoring site. This line carries a substantial proportion of the freight traffic traversing RailCorp’s urban network. The main reason for choosing this site was the fact that a permanent AoA detector was installed at this site in August 2007, specifically for the purpose of a curve squeal noise study. The curve radius is 284 m, with a 130 mm superelevation and 1:52 downhill grade.

The track components of the measurement site include 60 kg rails, concrete sleepers, e-Clip fastening system, and ballast founded on a dry reactive soil formation.

Curve squeal noise from freight trains is the main source of noise complaints from people living close to this site, although the track in question mainly carries passenger trains, including EMUs, DMUs, and locomotive-hauled cars.

3.2. Instrumentation

- **Field test 1**: AoA data were collected over a period of one year (from August 2007 to July 2008), encompassing around 183,000 passenger bogie and 33,000 freight bogie pass-bys. During the same year, trackside noise monitoring was conducted at the same site intermittently for an accumulated period of about 83 days.

- **Field test 2**: Later on, additional data, including rail forces and vibrations, were measured simultaneously with noise and AoA data over a shorter period of nine days, capturing 32 freight train pass-by events. The measurements conducted in field test 2 were conducted in the daytime only.

**Noise**

Noise data was collected using a B&K Type 2260 noise monitoring microphone. The microphone was positioned adjacent to the AoA detector, about 2 m laterally from the inner rail, and around 1 m above the top of rail. The microphone position varied for different measurement periods. The noise data was sampled at 22 kHz per second using commercial data acquisition system DEWESoft™.
AoA
The AoA detector was installed at about halfway, on the inner side of the curve (see Figure 1). The system picks up traffic in one direction only (up direction, and on a downhill). AoA detected from passing wheel axles from both passenger and freight trains are recorded and stored in a database, and can be downloaded for further analysis. Other parameters such as wheel set lateral position, train speed, rail and ambient temperature and dew point are also available from the AoA detector system.

![AoA Detector](image1)

Figure 1: The location (a) and the AoA detector (b) installed at Beecroft, Sydney

Strain gauges
Strain gauges were installed on both the low and high rail near the AoA detector in field test 2. The vertical rail force measurement pattern with excellent linearity and minimal cross-talk, and the base chevron pattern as suggested in Harrison and Ahlbeck (1981), were used to measure the vertical and lateral rail forces respectively. One sleeper was displaced to allow a span of 600 mm to be achieved between tie edges in order to install the strain gauges. The spacing between two regions (a–b and c–d) of strain gauges is 450 mm.

Accelerometers
Tri-axial accelerometers were mounted on both rail webs in the vicinity of the strain gauges in field test 2. Three-dimensional accelerations of both rails were measured simultaneously with rail forces, noise and AoA using a commercial data acquisition system, DEWESoft™. Both the rail forces and accelerations were sampled at 10 K samples per second, while the sound was sampled at 20 K samples per second.

3.3. Field data collection
Given that it was an observational study, no control is available on various additional factors, such as rolling stock operation conditions, weather conditions, and the occurrence of curving noise. It is therefore desirable to conduct a longer observation period at some point.

In field test 1, noise and AoA data were collected over a relatively long monitoring period. AoA data were collected from the permanent AoA detector over a one-year period (from August 2007 to July
2008). During the same period, noise data were collected over some isolated periods from August 2007 to February 2008. A portable noise data logger was used to record noise.

The data collection for field test 2 was conducted only during the daytime over nine days from 22 March 2010 to 6 May 2010. In total, 32 freight train pass-bys were recorded during field test 2.

The data analysis methods and results are presented in the following sections.
4. Curving noise identification and analysis

In order to verify the curving noise mechanisms, the type of curving noise and its origin have to be identified in a verified and rigorous manner. Curving noise identification involves three aspects: the identification of the type of noise, i.e. the mono-tonal squeal or the broadband flanging noise; the identification of the origin wheel set; and the identification of the origin wheels (the inner or outer wheel).

The curving noise identification method and results are discussed in the following sections.

4.1. What type of curving noise?

Automated squeal and flanging noise identification has previously been developed by the authors (Dwight & Jiang, 2009). The method is based on the different spectrum characteristics of these noises. For example, typical 1/3 octave spectra of squeal noise and flanging noise is shown in Figure 2. In this example, the squeal noise has a dominant frequency component at approximately 2000 Hz and some harmonics; while flanging noise exhibits multiple tonal components or broadband frequency components ranging from ~300 to over 20,000 Hz.

The raw noise data recorded in field test 1 were analysed using the automatic curving noise identification method mentioned above. In total, 131 squeal noise events were detected from 242 freight train passes (44,707 axles), with the noise level exceeding 100 dB(A) recorded between August 2007 and February 2008. The frequency of the first harmonic of the detected squeal noise was analysed. The distribution of the squeal noise frequency is shown in Figure 3. It was observed that the majority (67%) of the squeal events detected occurred in a frequency range from 1000 to 3000 Hz.

Squeal noise differs from flanging noise, not only in its frequency spectrum, but also in its intensity. The maximum squeal and flanging noise levels over the same monitoring duration were calculated. The result is shown in Table 1. It is evident that the squeal noise level was generally higher than the flanging noise emitted from both freight and passenger trains at this particular site. It was also noted...
that the squeal noise generated from freight trains was generally higher than that generated by passenger trains.

![Figure 3: The distribution of squeal noise frequency](image)

(Note: the squeal events were recorded in field test 1)

<table>
<thead>
<tr>
<th>Monitoring periods</th>
<th>LAFmax (dB(A))</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Passenger</td>
</tr>
<tr>
<td></td>
<td>Squeal</td>
</tr>
<tr>
<td>1 (2007-06-15 to 2007-06-22)</td>
<td>98</td>
</tr>
<tr>
<td>2 (2007-10-18 to 2007-10-23)</td>
<td>94</td>
</tr>
<tr>
<td>3 (2007-10-23 to 2007-10-26)</td>
<td>91</td>
</tr>
<tr>
<td>4 (2007-11-08 to 2007-11-12)</td>
<td>102</td>
</tr>
<tr>
<td>6 (2007-12-21 to 2008-01-01)</td>
<td>102</td>
</tr>
<tr>
<td>7 (2008-01-03 to 2008-01-10)</td>
<td>93</td>
</tr>
<tr>
<td>8 (2008-01-10 to 2008-01-14)</td>
<td>102</td>
</tr>
<tr>
<td>9 (2008-01-24 to 2008-02-04)</td>
<td>105</td>
</tr>
<tr>
<td>10 (2008-02-04 to 2008-02-11)</td>
<td>103</td>
</tr>
<tr>
<td>11 (2008-02-15 to 2008-02-19)</td>
<td>89</td>
</tr>
<tr>
<td>13 (2008-07-03 to 2008-07-08)</td>
<td>111</td>
</tr>
<tr>
<td>Max</td>
<td>117</td>
</tr>
</tbody>
</table>
4.2. Identification of axle-generating curving noise

Another key aspect of noise identification is to associate the identified noise event with a specific axle. For a single microphone-based system, the identification of the axle that is generating squeal or flanging noise may be based on the temporal characteristics of that noise event. The noise level increases as the wheel emitting squeal noise approaches, and then reduces as the wheel departs from the microphone, if constant source strength is assumed. It would be expected, and must be assumed, that the maximum noise level is observed when the wheel is adjacent to the microphone.

In a simple monitoring system, the wheel responsible for the noise may be identified as the one closest when the noise event reaches a peak level. This method leads to a false detection of the noise generating wheel if the noise is intermittent and stops before that wheel arrives at the position opposite to the measurement microphone. This is more of an issue for flanging noise. It will also be imprecise where adjacent wheel sets are generating curving noise. The allocation of curving noise to a particular wheel set may sometimes be verified by the point of frequency shift on account of the Doppler effect.

In order to test the accuracy of this approach to axle identification, a comparison was made between this single microphone-based trackside noise (TN) system Dwight & Jiang, 2009) and a microphone array-based system, RailSQAD (Jiang, 2010). This comparison was conducted at a different location to the field trials owing to the availability of the RailSQAD system. It was found that:

- the number of squeal noise events detected by RailSQAD is significantly higher than that detected by the TN system
- the RailSQAD system is more likely to falsely identify some wheel sets as emitting squeal noise in the case of a sustained severe squeal noise occurring or a ‘squeal-like’ flanging noise occurring.

This may indicate that an axle is squealing when the noise may, in fact, be generated by an adjacent axle. Where these are on adjacent cars on a one-off basis, or where those cars are coupled together in successive consists, the car may be labelled as ‘poorly performing’ from a noise perspective, thereby resulting in unnecessary inspection.

The TN system is likely to fail to identify axles generating squeal noise in the case when two or more squealing axles are closely spaced, although this rarely occurred at Beecroft. It may also overestimate squeal events when ‘squeal-like’ flanging noise occurs, which is more likely to occur when the noise level of identified squeal events is relatively low.

The noise level measured by the TN system was, in the configuration of this test, on average about 2.9 dB(A) higher than that measured by the RailSQAD system.

In this report, only those severe squeal events detected by TN — >100 dB(A) — were used for analysis. Therefore, the false detection of squeal events is minimised.

4.3. Squeal generation from inner or outer wheel

A further step to identify the source of squeal noise is to detect which wheel, i.e. the outer or inner wheel, is generating the noise. In the literature, the pure tonal squeal noise is generally thought to be generated from the inner wheel, as has been reported from a number of field observations (von Stappenbeck, 1954; Vincent et al., 2006). The identification of the exact squeal noise source, i.e. the inner or outer wheel, is not only important in verifying the squeal noise mechanisms, but also important in practice to take appropriate noise mitigation measures. Accelerometers and strain gauges were installed on both rails in the field test 2 measurements at Beecroft to allow verification of the rail and wheel involved in the generation of the vibration.
Examples of low and high rail squeal events are shown in Figure 4 (a) and (b) respectively. In the case of inner wheel squeal, the lateral rail vibration at the low rail is about 20 dB higher than the corresponding vibrations at the high rail. Similarly, the high rail vibration is approximately 20 dB higher than the corresponding low rail vibration when the outer wheel generates squeal.

The identification of inner or outer wheel generated squeal noise can also be done based on the rail force measurements. This will be explored later in this report.

**Figure 4:** Raw lateral accelerations indicate the origin of squeal noise:
(a) an inner wheel squeal recorded from a freight train passing the site (48 km/h) at 11:22 am on 22 April 2010
(b) an outer wheel squeal from a freight train passing the site (41 km/h) at 1:50 pm on 22 April 2010.
More squeal events were observed from the outer wheel than the inner wheel during the field tests at Beecroft. During the short monitoring period (nine days) in field test 2, 28 squeal noise events were recorded from 32 freight train passes. Interestingly, only four of those 28 squeals were found to be generated from the low rail–inner wheel interface. This was not expected, and is contradictory to the current theory and some other authors’ field findings. It is believed that flange contact at high rail prevents squeal noise generation. Squeal noise is therefore normally generated from inner wheel–low rail, as observed from field tests (von Stappenbeck, 1954; Vincent et al., 2006).

The role of flange contact

For this unexpected observation, the immediate questions are:

- Is the outer wheel in flange contact with the high rail?
- If yes, what is the role of flange contact in squeal noise generation?
- Why does the flange promote squeal noise generation instead of suppressing its generation?
- Is this site specific?

The wheel lateral positions are examined to answer the question as to whether the outer wheel (generating squeal) is in flange contact or not.

Figure 5 shows a scatter plot of wheel AoA vs lateral position. The data were recorded from a freight train pass-by (2010-3-24, 14:48 pm). In this figure, positive AoA means that the wheel is attacking the high rail, and positive lateral position means the wheel shifts toward the high rail. It is seen that the majority of wheels have a positive AoA and lateral position, which implies that a wheel set tends to steer and shift toward the high rail when there is a positive AoA. Similarly, when the wheel set has a large negative AoA, it tends to steer and shift to the low rail. For wheels with a positive AoA (>10 mrad), the lateral positions are all in the positive 20~30 mm range, whereas for wheels with large negative AoAs (< -10 mrad), the lateral positions are all in the negative 20~30 mm range. If a wheel set lateral position is around positive 20~30 mm, it is thought that the outer wheel is in flange contact against the high rail; while a negative 20~30 mm means that the inner wheel is in flange contact against the gauge side of the low rail.

![Wheel AoA vs. Lateral Position](image)

Figure 5: Scatter plot of wheel AoAs vs lateral positions recorded from a freight train pass-by.
For this particular train, there were five squeal events identified from the outer wheel–high rail interface. For all five, squealing wheel sets are in the high rail–flange contact position, i.e. the lateral position is in the range of positive 20–30 mm.

This observation appears to show that flange contact may not only generate flanging noise, but also may play a role in the pure-tonal squeal noise. It clearly contradicts the theory that wheel flange normally prevents squeal noise generation (Remington, 1985), in addition to some laboratory and field measurements (Remington, 1985; von Stappenbeck, 1954; Vincent et al., 2006). At this stage, it is not clear what role wheel flange contact plays in the generation of squeal noise.

4.4. Curving noise monitoring strategies

On the basis of previous discussions, valid curving noise monitoring strategies can be developed depending on various requirements:

- A single microphone-based noise monitoring system can meet the requirements of general curving noise monitoring and detection.
- With additional wheel sensors, the identified curving noise can be associated with a particular wheel axle.
- To further separate the curving noise into inner or outer wheel events, vibrations of the low and high rail, in either vertical or lateral direction, need to be measured.
5. The influence of angle of attack on curving noise

5.1. Introduction

It is generally thought that the wheel AoA is the key factor for the occurrence of curve squeal. Squeal noise only occurs if the wheel AoA exceeds a certain threshold, after which the wheel–rail interface creep force becomes saturated. The threshold of AoA is identified as 7–10 mrad by some authors (Remington, 1985; Vincent et al., 2006). It is noted that high AoA (>10 mrad) is a necessary, but not sufficient condition for curve squeal noise generation (Dwight & Jiang, 2009). Curve squeal noise may not occur with the presence of high AoA when the other conditions, such as the wheel–rail interface friction condition and light wheel damping, are not present. The reverse is true based on the same theory, i.e. that curve squeal will not occur, irrespective of other factors, with low AoA.

The field data collected at field test 1 can be used to verify the current knowledge on the correlation between AoA and curving noise. Research questions to be answered include:

1. In relation to the nature of the correlation between AoA and curve squeal noise, what is the critical AoA (or creepage) for squeal noise occurrence?

2. How is flanging noise related to AoA?

5.2. AoA results

AoA distributions

The average and standard deviations of AoAs over a one-year period (from August 2007 to July 2008) have been calculated and are shown in Table 2.

Table 2. The average and standard deviation of measured AoAs

<table>
<thead>
<tr>
<th>Axle type and position</th>
<th>Total number of bogies</th>
<th>Average AoA (mrad)</th>
<th>Standard deviation (mrad)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Freight, leading axle of leading bogie</td>
<td>33324</td>
<td>6.28</td>
<td>5.06</td>
</tr>
<tr>
<td>Freight, leading axle of trailing bogie</td>
<td>33394</td>
<td>4.20</td>
<td>3.67</td>
</tr>
<tr>
<td>Freight, trailing axle of leading bogie</td>
<td>33397</td>
<td>0.93</td>
<td>4.78</td>
</tr>
<tr>
<td>Freight, trailing axle of trailing bogie</td>
<td>33393</td>
<td>-0.88</td>
<td>3.41</td>
</tr>
<tr>
<td>Passenger, leading axle of leading bogie</td>
<td>183253</td>
<td>7.07</td>
<td>1.48</td>
</tr>
<tr>
<td>Passenger, leading axle of trailing bogie</td>
<td>183339</td>
<td>6.34</td>
<td>1.50</td>
</tr>
<tr>
<td>Passenger, trailing axle of leading bogie</td>
<td>183341</td>
<td>0.03</td>
<td>0.70</td>
</tr>
<tr>
<td>Passenger, trailing axle of trailing bogie</td>
<td>183341</td>
<td>-0.73</td>
<td>1.01</td>
</tr>
</tbody>
</table>

(Note: AoAs measured at Beecroft from Aug 2007 to July 2008)
The average AoA of freight train leading axles of the leading bogie is 6.28 mrad, which is slightly less than the corresponding passenger train AoA (7.07 mrad). This is expected, as the wheelbase of a passenger train (around 2.4 m) is normally longer than the freight train wheelbase (around 1.8 m). The average AoA of the trailing axle is close to 0 mrad for both passenger and freight trains, which means that the trailing axles tend to align well with the curve, as predicted by rolling stock theory (Boocock, 1969; Elkins & Eickhoff, 1982).

The observed standard deviation for freight train AoAs is several times higher than that of passenger trains. For example, the standard deviation of the freight train AoA of the leading axle of leading bogie is 5 mrad, and the corresponding passenger train is only 1.48 mrad.

However, the wide variation of AoAs is not seen from all freight wagons. For example, one particular wagon class (see Class A in Figure 6) has the most passes at Beecroft site (3498 out of 33,324, or more than 10% of the total freight wagon passes), 100% of its 3498 passes are detected within a narrow band (0–10 mrad). In comparison, the worst-performing freight wagon class in terms of AoA variations (see Class B in Figure 6) has more than 70% of leading AoAs exceeding 10 mrad, and 158 out of 632 (or 25%) leading AoAs exceeding 30 mrad. This wagon class occupies more than 40% of overall freight wagons passes with AoAs exceeding 30 mrad.

It is notable that the Class B has a bi-modal distribution: the first mode is centred on 6 mrad, which is the normal average AoA; the second mode is centred on 30 mrad, which corresponds approximately to the AoA values of those bogies without rotation. Bogies with excessive rotation resistance may not turn when they enter into a curve, and thus result in excessive AoA values. The excessive resistance may be caused by a worn centre-plate, improperly constant contact side-bearer settings, and other malfunctions of the bogies. Bogies with such behaviour do not steer well in curves, generate excessive forces between wheel and rail, and contribute to wheel and rail wear. The influence on curving noise emission and the formation of corrugation are discussed in the following sections. Bogies with excessive rotation resistance are likely to function properly on tangent track. Therefore, the detection of bogies with large AoAs due to excessive rotation resistance should be carried out at curves instead of tangent tracks.
Comparing the measured AoAs with calculated values

The upper limit of the leading wheel AoA can be estimated using a simple geometry based equation (Remington, 1985):

\[
\text{Equation 1}
\]

i.e. the wheelbase ‘a’ divided by the curve radius ‘R’, assuming that the trailing wheel is aligned to the curve.

For example, the leading wheel AoA is around 6.3 mrad (i.e. 1.8 m/284 m) for 1.8 m wheelbase bogies, which is common to freight bogies. The calculated value (6.3 mrad) is very close to the measured average AoA (6.28 mrad). Similarly, the calculated AoA of the leading axle is around 8.5 mrad for a passenger train with a 2.4m wheelbase; while the measured average AoA is around 7.07 mrad.

It should be noted that this formula is valid for the majority of bogies being monitored at Beecroft. Therefore, it can be defined as a ‘normal’ curving behaviour if a leading wheel’s AoA falls between the limits defined by Equation 1. If a leading wheel’s AoA is beyond those calculated limits, attention should be paid to those wagons. This abnormal behaviour is likely to be due to some faulty parts of the bogie, such as worn central bowl-plate or improperly set side-bearers, both of which may lead to excessive bogie rotating resistance and high AoAs.

Expected curving noise behaviour at Beecroft

It is seen that either the measured or calculated AoAs from the leading axles of both passenger and freight trains are normally less than the critical threshold (~ 10 mrad). That means both freight and passenger trains passing Beecroft normally will not generate squeal noise if the theory underpinning the 10 mrad threshold is valid. Only those trains with abnormal AoA behaviour are expected to generate squeal. On account of the much wider variation of AoA, freight wagons passing Beecroft are more likely to exceed the 10 mrad AoA threshold and generate squeal noise.

5.3. Correlation between AoA and curving noise

131 squeal noise events were detected from pass-by freight trains (see section 4.1). These were matched to the corresponding AoA measurements. Some examples are shown in Figure 7 to illustrate the typical behaviour of curving noise with wheel AoA.

A freight train with all bogies having normal AoAs is correspondingly generally quiet, and no curving noise is detected from this train pass-by (see Figure 7 a). Leading wheels of locomotive bogies have higher AoAs, around 12 mrad, owing to the longer wheelbase than the general wagon bogies. Therefore, they may or may not generate squeal noise subject to the presence of other factors such as friction condition. Another unique factor involving locomotive wheels is the longitudinal creepage resulting from the traction effort. As indicated in Monk-Steel et al. (2006), the presence of longitudinal creepage may help to prevent squeal noise generation.

Figure 7 b shows a freight train with all normal AoA bogies that generates a significant amount of flanging noise detected during its pass-by.

Figure 7 c shows a freight train with one only bogie exhibiting an abnormally high AoA and coincident squeal noise emission.

Figure 7 d shows a freight train with several bogies exhibiting a high AoA, but with only one of these associated with a significant level squeal noise event.
A summary of the squeal events and their corresponding AoAs, measured from August 2007 to February 2008, is shown in Table 3. This result tends to suggest that squeal noise rarely occurs when the wheel’s AoA is small (<10 mrad). Squeal noise occurrence tends to increase with AoA, and the percentage of squeal noise occurrence reaches more than 50 to 60% with very high AoA (>40 mrad). In total, about 7.5% of wheels with AoA greater than 15 mrad were observed to generate squeal noise above 100 dB(A). These observations are consistent with those made in Report A1 (Dwight & Jiang, 2009).
Table 3: Summary of squeal events by the category of AoA and noise level

<table>
<thead>
<tr>
<th>AoA (mrad)</th>
<th>Noise category</th>
<th>Subtotal</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>num</td>
<td>100-105</td>
</tr>
<tr>
<td>&lt;=5</td>
<td>27382</td>
<td>1</td>
</tr>
<tr>
<td>5-10</td>
<td>13923</td>
<td>6</td>
</tr>
<tr>
<td>10-15</td>
<td>1694</td>
<td>0</td>
</tr>
<tr>
<td>15-20</td>
<td>575</td>
<td>1</td>
</tr>
<tr>
<td>20-25</td>
<td>459</td>
<td>2</td>
</tr>
<tr>
<td>25-26</td>
<td>360</td>
<td>3</td>
</tr>
<tr>
<td>30-35</td>
<td>201</td>
<td>1</td>
</tr>
<tr>
<td>35-40</td>
<td>84</td>
<td>0</td>
</tr>
<tr>
<td>40-45</td>
<td>26</td>
<td>0</td>
</tr>
<tr>
<td>&gt;45</td>
<td>3</td>
<td>0</td>
</tr>
<tr>
<td>Total</td>
<td>44707</td>
<td>14</td>
</tr>
</tbody>
</table>

(Note: microphone positioned 2 m from the inner rail)

It should be noted that the results presented in Table 3 only represent those squeal events with a noise level exceeding 100 dB(A), and with a primary frequency within a specific range from 500 to 10000 Hz. Squeal noise can occur at higher frequency range (above 10 kHz), or even in the non-audible ultrasonic range (greater than 20 KHz), such as reported by Rudd (1976). It is expected that the percentage (7.5%) will be higher if all squeal events with lower amplitude and frequencies out of the specific range (500~10000Hz) are taken into account. Nevertheless, it is expected that some wheels with a high AoA will not generate audible squeal noise. This may be due to the absence of other conditions required, such as friction conditions which, according to Rudd’s theory, influence the occurrence of squeal noise; or else it may be due to some other parameters not included by the accepted theory. The investigation of the influence of friction condition on curving noise will be discussed in the next section.

The correlation study between curving noise and AoA based on the Beecroft data can be concluded as follows:

- It tends to prove that curve squeal noise only occurs when the lateral creepage (or AoA) exceeds a certain level — around 7~10 mrad.
- Squeal noise may or may not occur in the presence of high AoAs (>10 mrad). For example, the Beecroft monitoring result shows that only 7.5% of axles with AoAs >15 mrad generated squeal noise over 100 dB(A) (measured 2 m away from the inner rail).
- Not only the magnitude, but also the likelihood, of squeal noise appears to increase with increasing creepage (or AoA).
- The occurrence of flanging noise has no obvious relationship to AoA.
6. The influence of friction

6.1. Introduction

Previous observations at the Beecroft site suggest that almost all squeal noise events are associated with high amplitude AoAs (>7~10 mrad), while many axles with high AoAs do not generate squeal noise. Such observation supports the theory that high AoA is a necessary, but not sufficient, condition for the generation of curve squeal noise. It is suspected that those wheels with high AoA (>10 mrad) that are not generating curve squeal noise lack other parameters, particularly the friction condition. Therefore, a more complete field measurement, including friction measurement, was necessary to verify the influence of the friction on curving noise.

The test conducted in field test 2 was specifically designed to investigate the influence of the friction condition on curving noise. Rail forces were measured using strain gauges to estimate the friction between the wheel–rail interfaces at the point where each wheel passes the measurement site.

6.2. Friction estimation based on the lateral and vertical rail forces

Forces acting at the wheel–rail contact are illustrated in Figure 8, which shows a typical two-point loading condition when a wheel set is negotiating a sharp curve, and which illustrates the feasibility of using lateral rail force/vertical rail force (L/V) ratio to measure the wheel–rail interface friction coefficients. Similar conclusions can be drawn in the case of single-point contact on the high rail–outer wheel contact situation.

\[ F_y = \text{High rail vertical force at tread contact} \]
\[ F_{lc} = \text{High rail lateral force due to creep} \]
\[ F_{rf} = \text{High rail vertical force at flange contact} \]
\[ F_{lr} = \text{High rail lateral force due to flange force} \]

Figure 8: Schematic diagram of wheel-rail interface forces shown acting on the rails
At the low rail, the lateral force is mainly due to the lateral creep. Therefore, the L/V ratio can be used to estimate the friction coefficient of the low rail in the case of saturated lateral creepage (AoA > 10 mrad). For AoAs less than 10 mrad, the low rail L/V ratios will approximate the rolling friction coefficient.

The measured AoA is equivalent to the lateral wheel creepage under steady running condition, while, under stick-slip state or squeal noise generation state, the dynamic creepage fluctuates around the saturation point. It is generally reasonable to assume that an axle’s AoA is stable while running through a curve unless there is a dynamic hunting instability occurring. It is expected that measured L/Vs against various AoAs will provide an estimation of the creepage–creep force curve at the low rail–wheel tread interface.

For the high rail, the net lateral force is a combination of the lateral creep force and flange force, while the lateral flange force results from the lateral creep forces from both rails. In the case of a two-point contact, the vertical forces on the high rail consist of the static vehicle weight, dynamic forces owing to surface irregularities, responses to track geometry errors, and a vertical component of the flange contact force. As a result, the high rail L/Vs cannot be used to estimate the wheel–high rail interface friction.

As the low rail is generally thought to be the origin of the tonal curve squeal noise, it seems feasible to use the measured low rail L/Vs vs AoA as an estimation of the wheel–low rail interface friction condition. It can also be used to verify the influence of friction on curve squeal noise.

6.3. L/V results from the low rail

Scatter plots of low rail L/Vs vs AoAs measured from two freight train pass-bys are shown in Figure 9. A 4th order polynomial trend-line is added to each scatter plot to illustrate the general trend of the L/V vs AoA distribution.

Wide variations of L/Vs are observed from both pass-bys. For example, the L/V values indicated vary from 0.1 to 0.4 in the AoA range 10 to 15 mrad, as measured from one train (2010-3-24), and the indicated L/Vs vary from 0.3 to 0.7 in the same AoA range for the other train (2010-4-21). This wide variation of the friction coefficient estimated from the L/Vs curiously suggests that the friction may vary significantly from one wheel to the other, even from the same train pass-by. The average friction coefficient (or average L/Vs for AoA > 10 mrad) indicated for the pass-bys is 0.2 and 0.44 respectively. This indicates that the general friction level from one train pass-by (2010-3-24) is much lower than the other one (2010-4-21).

All of this indicates that the measured curve (L/Vs vs AoAs) does not provide an accurate estimation of the actual creep force–creepage curves under running condition on account of the wide variation of the L/Vs with close AoAs. Controlled laboratory tests are required to establish the corresponding creep force–creepage curves under various conditions, such as dry, wet and oil-contaminated. The difficulty with this is recognising the relevant conditions for a particular train pass-by.
The influence of friction on curving squeal noise

As indicated in section 4.3, the majority of squeal events identified in stage 2 measurements were generated from the outer wheel–high rail interface, and only four squeal events were identified to be generated from the inner wheel–low rail interface. As we know, the friction estimation method based on the L/V ratio is only valid for the top of low rail–inner wheel tread interface; it is not valid for the high rail–outer wheel contact owing to the contact angle. The friction condition estimated from the low rail L/V ratios is only valid for the low rail–wheel tread interface. It therefore cannot be used to infer the high rail–outer wheel interface friction as the high rail gauge face friction may vary significantly subject to the gauge lubrication state. Since there was very few limited squeal noise samples observed from the low rail–inner wheel, the investigation of the effect of friction on squeal noise is not pursued in this study.
To investigate the influence of friction on squeal noise in a thorough fashion, some alternative measurement plans are suggested:
1. longer term monitoring at Beecroft to accumulate sufficient squeal noise samples from the inner wheel–low rail
2. monitoring at other sites
3. conducting friction squeal noise tests on a laboratory test rig.

6.5. The influence of humidity on squeal noise

The squeal events collected in field test 1 can be used to analyse the influence of humidity on the occurrence of squeal noise. In total, there were 131 events identified with noise level exceeding 100 dB(A) (measured from 2 m away). Figure 10 shows a scatter plot of the squeal noise events vs the locally measured relative humidity and the percentage of squeal noise occurrence. It seems that the occurrence of squeal noise events is evenly distributed across the relative humidity range from 20% to 100%; however, no severe squeal noise occurred in the very low humidity range (<20%).

Another study has been conducted by ARTC to investigate the relationship between the occurrence of squeal noise and relative humidity using RailSQAD data collected at Heathfield, Adelaide Hills (Vincent et al., 2006). In that study, wheel squeal 2 events (>90 dB and <105 dB) are used to study the relationship, as they are more frequent and hence are more statistically significant. The result is shown in Figure 11. There does not appear to be a compelling relationship.

No conclusive result is yet drawn about the relationship between relative humidity and squeal noise occurrence in this study.
Figure 11: Squeal events vs. relative humidity (%) (Adelaide Hills data from Vincent et al., 2006)
7. AoA and corrugation

It is observed from the field test 2 measurements that a percentage of wheels with a high AoA (>10 mrad) do not generate squeal noise, but exhibit a low-frequency fluctuation in the rail forces measured. This chapter presents the results of the observed low-frequency fluctuation, and the analysis of its relationship with AoA. The relationship between AoA and corrugation is also considered.

7.1. Low-frequency fluctuation

Many wheel sets with a large AoA can roll or slip relatively constantly across a curve, as do those with small AoA.

For example, Figure 12 shows relatively constant rail forces measured at Beecroft when a wheel set passes with high AoA (16.9 mrad). This figure records the change in load as a wheel traverses the span between the sleepers for which strain gauges were attached. The vertical rail forces (as shown in the top graph in Figure 12) rise to a relatively constant value and remain at this value for about 300 mm of wheel travel, which is about 100 mm shorter than the spacing between the strain gauges used to measure the vertical forces. In this figure, the vertical loads for both low and high rail are around 5 t. The bottom graph shows the measured lateral forces from both rails. In contrast to the vertical rail force measurements, the lateral rail forces increase progressively and then decrease, again progressively. This occurs over a longer distance than for the vertical force changes. The peak lateral rail loads are around 2 t for both rails.

Some wheels with a high AoA have been observed with large amplitude force fluctuations at frequencies below 300 Hz. For example, Figure 13 shows a wheel (from a freight train pass-by, 2010-4-16_0030) with high AoA (23 mrad) accompanied with large rail force fluctuations. The peak-to-peak amplitude of the vertical load of the low rail is over 5 t, which is equivalent to its static vertical load. The peak-to-peak amplitude of the low rail lateral force is around 3 t. The high rail lateral forces...
also show the low-frequency fluctuation, but at a lower amplitude. The frequency of the fluctuation is 146 Hz. The corresponding wavelength of the low-frequency fluctuation is calculated as 102 mm.

![Figure 13: Example of rail forces with low-frequency (146Hz at 54.2km/h) fluctuation](image)

All the low-frequency fluctuation events with an RMS magnitude exceeding 0.3 t and 0.7 t were detected and listed in Table 4. In total, 4274 axles were detected during the field test 2 monitoring period, 24% of them with an AoA exceeding 10 mrad. There were 246 (5.6%) low-frequency fluctuation events detected among those 4274 axle passes. To break down this number, 54% of these events were associated with a high AoA (>10 mrad), which occupies 13% of the axles of that category (with AoA >10 mrad). For the low-frequency fluctuation events with higher fluctuating amplitude (RMS >0.7 t), a higher proportion (81%) were associated with wheels with a high AoA (>10 mrad). These data suggest that the majority (81%) of severe low-frequency fluctuation events generated were associated with high lateral creepage (>10 mrad).

The distribution of the frequency and wavelength of the low-frequency fluctuation is shown in Figure 14. This indicates that most of the low-frequency fluctuations fall into a frequency range of 100 to 240 Hz, and a corresponding wavelength range of 40 to 140 mm (based on vehicle speeds). The majority of the calculated wavelengths fall within the range of 40 to 100 mm, which is the typical range of short-wavelength corrugation.
Table 4: The distribution of the magnitudes of the low-frequency rail forces against AoAs

<table>
<thead>
<tr>
<th>AoA&lt;10 mrad</th>
<th>RMS &gt; 0.3 tonne</th>
<th>RMS &gt; 0.7 tonne</th>
</tr>
</thead>
<tbody>
<tr>
<td>3236 (76%)</td>
<td>113 (46%)</td>
<td>10 (19%)</td>
</tr>
<tr>
<td>AoA&gt;10 mrad</td>
<td>1039 (24%)</td>
<td>133 (54%)</td>
</tr>
<tr>
<td>Total</td>
<td>4274</td>
<td>246</td>
</tr>
</tbody>
</table>

(Note: the source data is from the field test 2 at Beecroft)

Figure 14: Distribution of the frequency (a) and wavelength (b) of the low-frequency oscillations
7.2. Discussion

The examination of the rail forces measured at Beecroft reveals that a high AoA not only can give rise to high-frequency squeal noise, but also may induce a low-frequency oscillation in a frequency range of 100 to 240 Hz. The corresponding wavelength of those low-frequency oscillations is around 40 to 140 mm, which is the typical wavelength of short-wavelength corrugation. Although no apparent corrugation was observed at Beecroft during the test, the above results show that those high-amplitude and low-frequency dynamic forces induced by high lateral creepage (or AoA) may contribute to the formation of corrugation at sharp curves. This might also be the main source of corrugation, given that the majority of the severe low-frequency oscillations are related to high AoAs. Further tests at some corrugation sites are suggested to confirm this finding.

In summary, high AoAs not only generate high-frequency squeal noise, but also are able to generate low-frequency oscillations. Here, the frequency corresponds to the typical short-wavelength corrugation observed at sharp curves. This may explain why some wheels with a high AOA are not observed to create audible squeal noise. The result is also of interest in understanding corrugation growth in curves.
8. **New squeal noise mechanism**

This chapter presents some of the most interesting phenomena observed in this study. In seeking explanations of these observations not reported in the literature, an alternative mechanism of curve squeal noise is proposed.

8.1. ‘Amplitude modulation’ of squeal noise

A close examination of squeal noise reveals that the amplitude of squeal noise waveform is always modulated. As an example, waveforms of a squeal event are shown in Figure 15. The top graph (Figure 15 a) shows the raw sound pressure of a freight train pass-by recorded in field test 1 at Beecroft. This train pass-by event lasts for about two minutes. Midway through this pass-by (about 58 seconds), a high-level noise event with the peak value exceeding 50 pa was identified as a squeal event. As zoomed in, the amplitude modulation of the waveforms can be seen clearly from Figures 15 (b) and (c). Figure 15 (d) shows the band-pass filtered waveform corresponding to the raw waveform as shown in Figure 15 (c). The bandwidth of band-pass filter is 200 Hz; the low and high cut-off frequencies are 2050 Hz and 2250 Hz respectively. It is seen that the amplitude modulation is more pronounced in the band filtered waveform.

![Figure 15: Waveforms of a squeal event look like amplitude modulation](image-url)
The spectrum of the squeal noise is calculated and is shown in Figure 16. This spectrum is calculated from a small section (0.1s) of the squeal event as shown in Figure 15 (c). Therefore, the frequency resolution is of 10Hz. From the wideband spectrum, only one narrow peak around 2000 Hz can be observed. However, when it is zoomed in, more than one peak can be observed from the spectrum: the highest peak at 2160 Hz, and the second highest peak at about 2130 Hz.

The above example shows the typical behaviour of squeal noise, i.e. the amplitude modulated waveform. Such behaviour can be observed from every single squeal event of the entire hundreds of squeal noise samples that the authors have collected so far, which includes squeal events recorded at Beecroft and from any other site.

For example, a squeal event recorded at Heathfield, Adelaide Hills, as illustrated in Figure 17, shows similar behaviour. The top graph (Figure 17 a) displays the waveform of the squeal event for a short duration of 0.3 second. The waveform has been filtered through a band-pass filter (2580~2780 Hz). ‘Amplitude modulation’ can be seen clearly from the waveform. This waveform is further divided into three consecutive and non-overlapped sections with 0.1 second duration for each section. Frequency analysis of three consecutive sections allows us to see the Doppler shift in one hand, and also the complex and dynamic nature of squeal noise. The spectrum of section I (see Figure 17 b) has a dominant component around 2710 Hz; while the spectrum of section II and III shows a slight different pattern (see Figure 17 c–d), i.e. it has two separate, but closely spaced, frequency components below 2700 Hz.
After reviewing all the hundreds of squeal noise samples, including samples recorded from Beecroft and other sites, one conclusion can be safely made — ‘amplitude modulation’ in the time domain waveform has been observed from every single squeal event. Similarly, ‘amplitude modulation’ can also be observed from rail forces and vibration measurements. This will be discussed in the following section.

8.2. Coupled vertical and lateral rail vibration

As squeal noise is the sound radiation of wheel and rail vibration, it is natural to expect similar ‘amplitude modulation’ from rail vibration measurements. More interestingly, as tri-axial accelerometers have been mounted to the rail web to measure longitudinal, vertical and lateral rail accelerations simultaneously in field test 2, the relationship of vibrations between different directions can also be explored.
Two examples of rail vibrations, one from an inner wheel squeal, and one from an outer wheel squeal, are given below.

![Waveforms](image)

**Figure 18: Low rail vertical and lateral accelerations (band-pass filtered) from an inner wheel squeal**

Figure 18 gives an example of rail vibrations from an inner wheel squeal event. Both the vertical and lateral low rail accelerations are displayed. This squeal event was recorded from a freight train pass-by at 13:50 pm, 22/4/2010 at Beecroft. The waveforms have been filtered through a band-pass filter centred around the squeal frequency with a 200 Hz bandwidth (1950~2150 Hz). Zoomed in waveforms are displayed from the top to the bottom graph.

The first thing to be noted is that the rail vibrates not only in the lateral, but also in the vertical direction at relatively high levels, with the peak levels exceeding 100 g. Although it is generally accepted that squeal noise is generated from the self-excited vibration of railway wheel, it is expected that rail vibrates at the same frequencies as the wheel, but at normally lower amplitude when squeal noise occurs. The rail vibrating strongly in both the vertical and lateral direction, or in both the normal and parallel direction, shows that the wheel is vibrating strongly in both directions simultaneously. This coupled vibration is mainly due to the non-symmetric shape of the wheel, which
leads to coupled wheel and rail vibrations in both the normal (or radial) and tangential (axial) directions.

Secondly, the waveforms of both the vertical and lateral rail vibrations seem to be modulated. The modulated vibration waveforms are similar to the squeal noise waveforms.

As zoomed in further, it can be seen that the motion between the vertical and lateral direction are not in phase, i.e. they do not reach the trough and peak simultaneously. A closer look at the waveforms (see the bottom graph in Figure 19) reveals that the instantaneous phase shifts between the vertical and lateral vibration, i.e., the phase difference between the vertical and lateral vibration varies continually with time, from in-phase to out-of-phase, and to in-phase repeatedly. The possible cause and implication of these observations will be discussed later in this chapter.

![Graph showing frequency analysis of rail accelerations](image)

**Figure 19**: Frequency analysis of the rail accelerations (inner wheel squeal) as shown in Figure 18

Frequency analysis of rail vibrations is carried out and the resulting spectra are shown in Figure 19. The spectra are calculated from rail accelerations with 0.1s duration, as shown in the middle graph in Figure 18. It is seen that the spectra of rail vibrations show similar pattern: two or more closely spaced peaks can be observed from the zoomed in spectrum, while the wideband spectrum may only show one peak.

Similarly, an example of rail vibrations measured with an outer wheel squeal event is shown in Figure 20, and the frequency analysis results are shown in Figure 21. This outer wheel squeal event was recorded from a freight train pass-by on 24/3/2010 at Beecroft. Behaviours such as ‘amplitude modulation’ and continuously varying phase difference between the vertical and lateral can also be
observed from rail vibrations associated with an outer wheel squeal event. Frequency analysis of the rail vibrations shows that the rail vibrates at slightly different frequencies in the vertical and lateral direction.

Figure 20: High rail vertical and lateral accelerations (band-pass filtered) from an outer wheel squeal
Figure 21: Frequency analysis of the rail accelerations (outer wheel squeal) as shown in Figure 20

8.3. Beats or amplitude modulation

In the previous sections, we have shown the varying amplitude or ‘amplitude modulation’ phenomenon, which can be observed from all the squeal noise and vibration samples received so far. This tends to suggest that this observed phenomenon is not a unique one, but a general one associated with curve squeal noise. It is interesting to ask: why and how are the amplitudes of squeal noise and vibration modulated? Is squeal noise a beating phenomenon? To answer these questions, the terms of beats and amplitude modulation and their relationship are introduced at first.

If two simple harmonic motions with close frequencies are combined, the resultant motion exhibits periodic amplitude variation and is called beats. For example, if two sinusoidal motions with close frequencies, \(\sin(\omega t)\) and \(\sin(\omega + \Delta \omega)t\), are added together, the resulting motion is:

\[
\sin(\omega t) + \sin(\omega + \Delta \omega)t.
\] . Equation 2

Therefore, the resulting vibration could be regarded as approximately simple harmonic motion with angular frequency \(\omega\), and with amplitude and phase varying periodically at a frequency of \(\Delta \omega/2\pi\). The addition leading to beats is purely mathematical result, and can be carried out for any values of \(\omega\) and \(\Delta \omega\). But its description as a beat phenomenon is physically meaningful only if \(\Delta \omega \ll \omega\). Human perception of beats also depends on the frequency difference (Rossing, 1982).
The superposition of two simple harmonic motions is graphically illustrated in Figure 22. Two sinusoidal vibrations with close frequencies — 520 Hz and 570 Hz — are superposed. The resulting beats waveform resembles the waveform obtained by modulating the amplitude of the vibration at a frequency $\Delta \omega / 4\pi$ (half the beats frequency), but they are not the same. Unlike the linear superposition leading to beats, amplitude modulation results from nonlinear behaviour in a system. Although the waveforms of beats and amplitude modulation look alike, they can be differentiated from the frequency domain (Rossing, 1995): amplitude modulation generates spectral components having frequencies of the modulated frequency $\omega$ and various sidebands, $\omega \pm \Delta \omega$, while the spectrum of beats has spectral components $\omega$ and $\omega + \Delta \omega$ only.

Figure 22: Coupled pendulums (a) and the corresponding beats vibration (b)

Beats commonly occurs in coupled systems with close natural frequencies. Such behaviour can be exhibited by two simple pendulums connected by a spring. As illustrated in Figure , two identical pendulums are connected with a soft spring. The coupled system has two normal modes of vibration given by:

\[ \text{Equation 3} \]

where $\omega$ is the natural angular frequency of the simple pendulum, $\Delta \omega$ is the coupling frequency, $L$ is the length of the pendulum, $K$ is the spring constant and $m$ is the mass. The softer the spring, the closer two normal modes become. Draw one pendulum aside while holding the other fixed, the motions of two coupled oscillators are beats between two simple harmonic motions of the same amplitude, but with slight different frequencies (see Figure 22 b) (French, 1971).

Therefore, the observations with squeal noise and vibration, such as the amplitude modulated squeal noise and vibration waveforms, and the coupled vertical and lateral rail vibration with close frequencies, strongly indicate that squeal noise is a special beating phenomenon, where the amplitude of beats vibration is transiently amplified through the interface friction. This is true at least for those already collected squeal noise samples, although there might be some other squeal noise events that do not belong to the same category. More importantly, the mode-coupling instability theory is available for the explanation of the mechanism squeal noise, and the role of interface friction in the transition of beating to instability. This will be discussed in the next section.
8.4. Mode-coupling instability: from beating to instability

In addition to the negative damping mechanism, the mode-coupling instability has been acknowledged as one of the most prominent mechanisms leading to friction excited vibration (Ibrahim, 1994). The mode-coupling instability mechanism, under which brake squeal even occur with a constant coefficient of friction, has been believed by some authors as a more realistic cause of automotive brake squeal (von Wagner, Hochlenert & Hagedorn, 2007). However, to the authors’ knowledge, no curve squeal noise model has been developed based on the mode-coupling instability mechanism. On the contrary, the negative damping mechanism has been believed as the main cause of railway wheel curve squeal noise since Rudd (1976), and many models have been developed based on this mechanism (Finberg, 1990; Périard, 1998; Heckl, 1999; de Beer, Janssens & Kooijman, 2003).

For the mode-coupling instability, its main characteristic is that the vibration frequencies of two structural modes of a system come closer and closer together until they merge, as a function of the controlling friction. This results in a pair of an unstable and a stable mode. A minimal model with two degrees of freedom proposed by Hoffmann et al. (2002) is used to clarify the underlying mechanism of mode-coupling instability.

Hoffmann’s model is shown in Figure 23. A conveyor belt with constant velocity is pushed with a normal force $F_N$ against a block with mass $m$. The block is held by two linear springs with stiffnesses $k_1$ and $k_2$. In addition, a third linear spring with stiffness $k_3$ represents the normal contact stiffness between the mass and the moving belt. A Coulomb-type friction force $F_F$ with constant friction coefficient is assumed. When small perturbations around the steady sliding state and approximating the friction force by $\mu k_3y$ are considered, the resulting equations of motion can be written as a homogeneous system with non-symmetric stiffness matrix:

Figure 23: Minimal single mass two degrees of freedom model by Hoffmann et al. (2002)

$$\text{Equation 4}$$

with the coefficients of the stiffness matrix
The non-conservative frictional force which is expressed in the non-symmetry of the system’s stiffness matrix $A$ is the source of system instability and can be analytically analysed. A special case is used to reveal the route to instability with varying friction force

$$\text{Equation 5}$$

which is obtained for $\mu$, $\mu$, $\mu$, $\mu$ and as an abbreviation for the contribution of the friction in the stiffness matrix. The results of complex eigenvalue analysis of this equation are shown in Figure 24. This shows the well-known picture for the mode-coupling type instability. The friction clearly plays a decisive role in leading to system instability. For smaller friction force ($\mu$), there are two normal modes with different frequencies. When the friction increases and approaches 1, the frequencies come closer and closer and coalesce. For $\mu$, a pair of an unstable and a stable mode with the same frequency results.

![Figure 24: Mode coupling in terms of the merging of natural frequencies and the appearance of unstable mode (Hoffman et al., 2002)](image_url)
The eigenvalue analysis shows that the instability sets in when friction force reaches to a critical value at . This is the exactly the configuration when the off-diagonal term \((1-\ )\) in the equation vanishes. An explanation in terms of forces can be made as following: there is structural coupling between the in-plane (tangential) and out-of-plane (normal) motion, as shown in Figure 25. In addition to these structural coupling terms, the friction force also act like a coupling force, but on the in-plane only. When the friction force increases, the friction force cancels out the structural coupling force at some value \(\), and the in-plane motion is uncoupled completely from the out-of-plane motion. The in-plane motion then, in turn, acts like an external force and drives the out-of-plane motion at resonance frequency, where instability makes its first appearance.

Hoffman clarifies the mode-coupling instability from a more intuitive perspective, i.e., from the energy transfer point of view (Hoffman et al., 2002). Time series by numerical integrations of Eq. (5) are shown in Figure 25. When is slightly below 1, the system has two normal modes with close frequencies. The integrated time series with general initial conditions exhibit a beating phenomenon.
It is characteristic of the beating phenomenon that there are phase shifts between in- and out-of-plane motion. This phase shifts allows energy to be transferred back and forth periodically between in- and out-of-plane motion. When \( \gamma \) gets closer to 1 the beating frequency decreases continuously, and for \( \gamma = 1 \), a beating is stretched out to infinity. The in-plane vibration is continuously feeding energy into the out-of-plane vibration, but the out-of-plane vibration cannot return energy into the in-plane vibration. When \( \gamma \) exceeds 1 the system is unstable, both vibration components increase exponentially in time. In summary, the origin and the role of phase shifts between the in-plane and out-of-plane vibrations has been clarified with respect to the mode-coupling type instability.

### 8.5. Squeal noise explained by the mode-coupling instability

The physical mechanism underlying the mode-coupling instability of friction-induced vibration has been introduced and clarified by a simple two degrees of freedom system in the previous section. Similarly, this mechanism can be used to explain the friction-induced self-excitation vibration phenomenon occurring in systems with many degrees of freedom or continuous systems, such as curve wheel squeal noise.

A railway wheel has numerous in-plane (radial) and out-of-plane (axial) modes, of which there are some pairs of radial and axial modes with closely neighbouring frequencies. For example, some of the axial and radial modes of a UIC 920 mm freight wheel are shown in Figure 26 (Thompson & Jones, 2000). The top and bottom row show the 0- and 1-nodal circles axial modes, and the middle row shows the 0-nodal circles radial modes. Some radial modes are found with close natural frequencies to the axial modes, such as the 1935 Hz radial mode and the 1978 Hz 1-nodal circles axial, the 2582 Hz radial mode and the 2511 Hz 1-nodal circles axial mode. As the wheel is not symmetric owing to the presence of the flange and the taped wheel tread, the in-plane and out-of-plane motions become coupled. The coupling of those modes results in beats. As explained in the mode-coupling instability theory, the beating motions provide a mechanism by which simultaneous out-of-phase in- and out-of-plane motions may lead to adding, instead of dissipating, energy to the vibration system. The onset of curve squeal noise is that the friction coupling force approaches or balances the corresponding structural cross-coupling force at a particular pair of radial and axial modes with close frequencies. The selection of a particular pair of modes being excited may depend on some factors such the interface friction force, the configuration of the wheel structural modes, and initial conditions. This question cannot be readily answered by the simple model with only two degrees of freedom. A more complex model, including all the modes in the audible frequency range, is to be developed, and will be the subject of further research.
These observations, such as the obvious amplitude-modulated noise and vibration waveforms, the strongly coupled vertical and lateral rail vibrations, and squeal noise as beating phenomenon, can be readily explained by the mode-coupling instability theory. The conventional mechanism of squeal noise, which is based on the assumption of a negative damping, usually only considers the unstable motion in the tangential direction, i.e. the lateral creep between the wheel tread and top-of-rail surface. It does not explain the strongly coupled normal and tangential motion that becomes evident in both the rail force and vibration measurements. In particular, it does not explain the observation that squeal noise can be generated under apparent flange contact. The mode-coupling mechanism seems fit in well in explaining those observations we have made at Beecroft.

For the mode-coupling instability, the friction condition is relaxed: the negative damping is not required anymore for squeal noise to occur, while it is a necessary condition for the conventional theory. Nevertheless, friction still plays a critical role in squeal noise generation. As explained by the minimal model with two degrees of freedom, the onset of instability is when the friction force balances the structural cross-coupling force. This implies that the reduction of friction force (or level) is potentially an effective measure in controlling squeal noise. Having said that, the falling friction condition (or negative damping) is not excluded as one of the squeal noise mechanisms unless there is evidence that the friction does not fall with increased creepage. It is likely that different squeal noise mechanisms exist and apply to different situations.
Based on the mode-coupling instability, another requirement for squeal noise generation is that a wheel has one or more pairs of closely spaced in- and out-of-plane modes. The effect of friction, regardless of whether it adds or dissipates energy, relies on the beating motions to be generated with wheels with closely spaced natural frequencies. This may explain the observation that only a small percentage of wheels with high AoAs generate squeal noise at Beecroft. Seemingly identical wheels may differ considerably in natural frequencies as the design tolerance on a railway wheel is often several millimetres in the width of the web. Material removal resulting from wear and turning gradually reduces the wheel diameter. This will also have a large effect on the natural frequencies. Further research efforts, particularly some field testing, including squealing and non-squealing wheel modal tests and comparisons, are to be pursued.
9. Direction for curving noise mitigation trials

Previous chapters deal with the studies of the influences of AoA and friction on the curving noise. A promising new squeal noise mechanism is proposed based on field observations. The implication of these studies on curving noise, particularly squeal noise mitigation, is discussed in this chapter.

9.1. AoA focused mitigation trials

It has been shown that AoA is a necessary condition for squeal noise to occur. Squeal noise rarely occurs for AoAs less than 10 mrad (as shown in Table 3). Based on this threshold, the minimum curve radius above which curve squeal noise is unlikely to occur can be calculated as follows, based on equation (1):

- for passenger train bogies with 2.4 m wheelbases, the minimum curve radius is 240 m (2.4 m/10 mrad)
- for freight train bogies with 1.8 m wheelbases, the minimum curve radius is 180 m (1.8 m/10 mrad).

These numbers show that curve squeal noise normally only occurs at very tight curves, for example, 180 m and 240 m or less for freight and passenger trains respectively. In other words, if there are squeal events observed from curves with larger radius curve, these squeal events are most likely to be generated from trains with abnormal curving behaviour.

This has been demonstrated from the Beecroft results, i.e. most squeal events occur with abnormal AoA values higher than the normal ones (~7 mrad). The bi-modal distribution of the worst-performing wagon class (see section 5.2) shows that those abnormal AoAs are mainly due to the bogie rotation issue. If the abnormal AoA is due to the bogie rotation issue, i.e. excessive rotation resistance, so the bogie does not turn itself when negotiating a curve, the leading wheel AoA of that bogie is likely to exceed the threshold of 10 mrad at curves with relative very large curve radius, such as from 700 m to 1500 m. It follows that bogies with rotation issues are likely to generate large lateral forces and potentially form short-wave corrugation or generate curve squeal noise at curves with large curve radius (>700 m).

Squeal noise mitigation measures such as reducing wheelbase or increasing curve radius are not effective for this type of bogie with abnormal curving behaviour. The obvious, and probably the most cost-effective, treatment method is to detect, check and modify or correct bogies with abnormal curving behaviour, which only occupies a small percentage of overall rolling stocks. The monitoring of such behaviour is better to take place at a curve as indicated in section 5.2.

9.2. Friction-based mitigation trials

In the previous section, it was suggested that the assumption of a negative friction damping is not necessary for squeal noise generation. Alternatively, a mode-coupling mechanism was proposed to explain the observed phenomenon, which otherwise cannot be explained by the conventional theory. Based on the mode-coupling theory, the friction condition is relaxed: squeal noise may be generated without the presence of negative friction characteristics, if the friction force acting as a cross-coupling force approaches or balances the structural coupling force. In other words, friction modifiers designed to alter the friction characteristics may not be effective in mitigating squeal noise generated by the mode-coupling mechanism. This might explain the observation that top-of-rail friction modification is only partially effective (if not at all) in practice (Anderson & Wheatley, 2007). With the mode-coupling instability theory, the absolute friction level plays a crucial role in squeal noise generation. For any given wheel configuration, there should exist a corresponding minimal
friction level, under which squeal noise will not occur. So the reduction of interface friction force is critical in friction-based squeal noise mitigation in line with the mode-coupling instability theory.

9.3. Wheel-based mitigation trials

As mentioned before, the onset of squeal noise is the combined results of the friction and structural coupling force, i.e. when the friction coupling force balances the structural force. Innovative squeal noise mitigation techniques may be developed based on appropriate wheel design. Wheels may exist with specific size and geometry that are prone to squeal noise generation, and wheels with other configurations that are not. A new squeal noise model based on the mode-coupling instability is to be developed to guide both the friction and wheel based mitigation trials. It is envisaged that a minimal friction level required for squeal depends on the specific wheel modal properties, which vary with the wheel size, geometry and material.
10. Conclusion and future research suggestions

10.1. Conclusions

Field verification of curving noise mechanisms was conducted at a specific curve site (with 284 m curve radius). The field measurements involved noise, AoA, rail forces, and rail vibration measurements.

Noise and AoA data was collected over several months. In total, 131 squeal noise events, with the noise level exceeding 100 dB(A), were detected from 242 freight train pass-bys (with 44,707 passing axles). The majority (67%) of those squeal events occurred in a frequency range of 1000 to 3000 Hz. The noise level of squeal events was generally 10–20 dB(A) higher than flanging noise.

The relationship between curving noise and angle of attack was analysed and the following conclusions reached:

1. Curve squeal noise only occurred when the lateral creepage (or AoA) exceeded a certain level, around 7~10 mrad.
2. Squeal noise may or may not occur in the presence of high AoAs (>10 mrad). For example, the Beecroft monitoring result shows that only 7.5% of axles with AoAs >15 mrad generated squeal noise over 100 dB(A) (measured 2 m away from the inner rail).
3. Not only the magnitude, but also the likelihood, of squeal noise appears to increase with increasing creepage (or AoA).
4. The occurrence of flanging noise has no obvious relationship to AoA.

The examination of the rail forces measured at Beecroft revealed that a high AoA not only gives rise to high-frequency squeal noise, but alternatively may induce a low-frequency oscillation in a frequency range of 100 to 240Hz. The corresponding wavelength of those low-frequency oscillations is around 40 to 140 mm, which is the typical wavelength of short-wavelength corrugation. This finding helps to explain why some wheels with a high AoA are not observed to create audible squeal noise. It also suggests that high lateral creepage (or AoA) could be the main contribution to the formation of corrugation at sharp curves, owing to the fact that the majority of low-frequency oscillations corresponding to frequencies of typical short-wavelength corrugation were induced by the high AoAs.

Some very interesting phenomena, which are either contradictory to, or cannot be explained by, the current curving noise theory, were observed:

1. More frequent squeal noise generated from outer wheel–high rail interaction was observed in the apparent presence of wheel flange contact than from low rail. This observation appears to show that flange contact may not only generate flanging noise, but also may play a role in the pure-tonal squeal noise.
2. Amplitude modulation was observed from noise, vibration and force measurements when squeal noise was detected. This observation is not limited to the squeal noise samples collected at Beecroft, but to all squeal noise samples obtained from other sites.
3. Strong coupling and constant phase change between the vertical and lateral rail vibrations when squeal noise was detected.

The conventional mechanism of squeal noise, which is based on the assumption of negative damping, only takes account of the unstable motion in the tangential direction, i.e. the lateral creep between the wheel tread and top-of-rail surface. It does not explain the strongly coupled normal and
tangential motion, as we have observed from the rail vibration measurements. It does not explain the observation that squeal noise can be generated under apparent flange contact. These observations appear to support the notion that squeal noise is a beating phenomenon and can be explained by the mode-coupling instability mechanism. Nevertheless, negative damping is not excluded as one of the squeal noise mechanisms, unless there is evidence that the friction does not fall with increased creepage. It is likely that different squeal noise mechanisms exist and apply to different situations.

The ratio of the lateral to the vertical rail force is used to estimate the friction condition at the wheel tread–top of low rail interface. The estimation shows a wide variation of the friction coefficient. This suggests that the friction may vary significantly from one wheel to the other, even from the same train pass-by. However, the influence of friction on squeal noise is not pursued on account of the following unexpected observation: the majority of squeal events identified in field test 2 were generated from the outer wheel–high rail interface; and only four squeal events identified to be generated from the inner wheel–low rail interface.

Valid curving noise monitoring strategies depending on various requirements have been demonstrated.

1. A single microphone-based noise monitoring system can meet the requirements of general curving noise monitoring and detection.

2. With additional wheel sensors, the identified curving noise can be associated with a particular wheel axle.

3. To further separate the curving noise into inner or outer wheel events, vibrations of the low and high rail, in either vertical or lateral direction, are required to be measured.

4. Adding a portable AoA detector, the system will be able to identify bogies with abnormal curving behaviour which lead to excessive lateral force, vibration and noise, and direct subsequent rolling stock-based maintenance efforts.

10.2. Future research suggestions

Given the observations of force, vibration and noise oscillations in the field measurements at Beecroft, it seems that mechanisms other than the negative friction-induced instability may be significant to the generation of squeal noise. Given this situation, the theory of mode-coupling instability developed in other areas of research is worth investigating. Further field study conducted at other sites is suggested to test and consolidate the mode-coupling theory discovered from the Beecroft observation. In coping with the new squeal theory, corresponding squeal noise mitigation methods need to be developed.
11. References


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